RHEOLOGICAL EFFECTS ON FRICITION IN ELASTOHYDRODYNAMIC LUBRICATION

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This report presents an analytical and experimental investigation of the friction in a rolling and sliding elastohydrodynamic lubricated contact. The rheological behavior of the lubricant is described in terms of two viscoelastic models. These models represent the separate effects of non-Newtonian behavior and the transient response of the fluid. A unified description of the non-Newtonian shear rate dependence of the viscosity is presented as a new hyperbolic liquid model. The transient response of viscosity, following the rapid pressure rise encountered in the contact, is described by a compressional viscoelastic model of the volume response of a liquid to an applied pressure step. The resulting momentum and energy equations are solved by an iterative numerical technique, and a friction coefficient is calculated. The experimental study was performed, with two synthetic paraffinic lubricants, to verify the friction predictions of the analysis. The values of friction coefficient from theory and experiment are in close agreement. The variation of the friction coefficient with rolling speed in a rolling contact system is closely investigated. Good agreement with existing experimental results is obtained at rolling speeds above 50 in/sec. At lower rolling speeds, a very rapid change in the effective viscosity of the lubricant is predicted. This behavior, in conjunction with shear rate effects, leads to large errors when experimental data is extrapolated to zero rolling speed.
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and Kelley [3]. However, these papers were not concerned with promoting the basic understanding of the traction between elastohydrodynamic contacts.

Crook [4] used two kinds of rolling disk machines in measuring the friction in a line contact as a function of sliding speed. In the region of small sliding speeds, he used the four-disk machine, a center disk surrounded by three equally spaced outer disks, shown in Figure 1.1. The center disk is free-floating and the measured torque does not contain any extraneous torque from the supporting bearings. For this reason, the four-disk machine gives very accurate frictional torque measurements at small sliding speeds. The four-disk machine is not suitable in the region of high slips, however, since it cannot maintain a stable sliding speed. For high sliding speeds, Crook used the two-disk machine shown in Figure 1.2, where the rotations of both disks are controlled by variable speed motors. Thus, Crook was able to measure the friction characteristics throughout the entire range of sliding speeds, using the four-disk machine in the low slip region and the two-disk machine in the high sliding speed region.

Crook found a profound influence of rolling speed upon the frictional torque in the low slip region. In this region, the slope of the traction versus slip curve is equal to the "effective viscosity" divided by the oil film thickness. Therefore, the effective viscosity may be evaluated by measuring the slope of the traction curve and calculating the oil film thickness from existing elastohydrodynamic theory. If the thermal effects and the non-Newtonian effects of the lubricant were both absent in this region, the effective viscosity would not be a function of rolling speed. However, this condition was

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Figure 1.1. Crook's four-disk machine. (a) Principle, (b) construction (diagrammatic). A to D, disks; E, aerostatic thrust; F, gear train; G, band brake. Figure from Crook [4].
Figure 1.2. Crook's two-disk machine. A and B disks; C and D swinging arms; E axle; F and G loading cables; H spring beam; I dial gauge. Figure from Crook [4].
not found in Crook's experimental results. On the contrary, he found a marked influence of the rolling speed on the effective viscosity of the lubricant that does not appear to be due to thermal effects only. Crook speculated that it was the viscoelastic effect of the lubricant which prevented it from reaching the static viscosity in the short time interval as it passes through the contact zone.

Crook was able to extend the friction data in the high slip region, with his two-disk machine, for loads ranging from $7.5 \times 20 \times 10^7$ dynes/cm$^2$ and rolling speeds from 400 cm/sec to 1200 cm/sec. All the friction curves show the same basic trend which is characterized by an ascending portion at small sliding speeds and a descending friction at high sliding speeds. An increase in load does not change the basic characteristics of the friction curve, but does increase the level of the friction force. Similarly, Crook found that an increase in the rolling speed decreases the friction level.

Crook also attempted to predict the friction analytically by a simplified thermal friction theory based on the following four assumptions: the film thickness within the contact zone is uniform; the pressure distribution in the contact region is Hertzian; the heat carried away by the lubricant due to convection may be neglected; and the temperature rise on the surface of the disk may also be neglected. Using this simplified theory for a Newtonian lubricant, Crook was able to calculate the coefficient of friction or the effective viscosity as a function of sliding speed. However, he could not predict the sharp reduction of the effective viscosity at small sliding speeds. He concluded that the friction force at small sliding speeds cannot be accurately predicted by considering the thermal effects only.
Cheng [5] employed his full elastohydrodynamic theory in calculating the friction for the conditions corresponding to those used in Crook's experiments. The temperature calculations are based on the finite difference solution of the energy equation and are free from all the assumptions made earlier by Crook. It is seen in Figure 1.3 that even with this refined thermal analysis there still exists a large discrepancy in the low slip region. This strengthens Crook's argument that the thermal effects alone cannot account for the sharp reduction of effective viscosity in the low slip region.

Bell, Kannel and Allen [6] developed an approximate analysis to predict the temperature rise in the lubricant film at low sliding speeds. Their analysis included the heat due to convection and the heat generation due to the compression of the lubricant. They also concluded that the temperature effects are too small to account for the loss of effective viscosity at low sliding speeds. In addition to the thermal theory, they developed a non-Newtonian friction theory using a rheological model proposed by Ree and Eyring [7]. The results of this analysis indicate that drastic reductions of friction can exist if the lubricant viscosity is shear rate-dependent according to Ree-Eyring. However, in all their calculated data, the friction force was found to be dependent upon $1/h$ as the rolling speed is varied, whereas all the experimental data gathered thus far has shown the proportionality to be far greater than $1/h$ and in most cases more nearly proportional to $1/h^2$. Thus, the inclusion of the Ree-Eyring model alone in the friction analysis would not be able to predict a sufficient reduction of friction at low rolling speeds.

Smith [8] employed the rolling contact machine shown in Figure 1.4.
Figure 1.3. Comparison of Cheng's thermal theory with Crook's experiments. Curve from McGrew, Gu, Cheng and Murray [9].
Figure 1.4. Smith's disk machine.

a Cylindrical roller  d Pivots  g Strain gauge dynamometer
b Spherical roller  e Motor
b Bearings  f Normal load

Figure from Smith [10].
to measure the friction between two rollers whose axes are skewed at an arbitrary angle. With this skewed arrangement, he was able to measure the friction force due to the sliding velocity component. The resulting friction versus sliding speed curves show trends similar to those observed by Crook. Smith divided these curves into several regions. He believed a Newtonian isothermal friction theory is applicable in the region where the friction varies proportionally with the sliding speed. In the ascending portion of the friction curves, where the friction force increases with sliding speed in a non-linear fashion, Smith believed that the non-linearity is due to the non-Newtonian behavior of the lubricant. He postulated that there is also a region in which a shear plane exists at the center of the lubricant film, such that the lubricant behaves like two solid layers sliding over each other at the shear plane. He further stated that the resistance to sliding at the shear plane is dependent upon the shear plane temperature and the hydrostatic stress in the lubricant. Finally, he defined a region where seizure would take place.

A more comprehensive study of friction covering a wide range of loads, rolling speeds and sliding speeds was carried out more recently by Johnson and Cameron [11] with a two-disk machine. The maximum Hertzian pressure was varied from 62,000 psi to 243,000 psi; the rolling speed was varied from 8 in/sec to 260 in/sec; and the oil inlet temperature was varied from 30 °C to 90 °C. The most striking feature of Johnson and Cameron's data is that there exists a ceiling to all the experimental traction coefficients which cannot be exceeded no matter how the load and the rolling speed are varied. They also took extensive data in the low slip region, and from the slope of the
traction versus slip curve were able to calculate the effective viscosity as a function of rolling speed. Johnson and Cameron furnished more convincing evidence that the thermal effects alone cannot account for the experimentally measured friction, and that a Smith-type limiting shear stress is dependent only on the shear plane temperature and pressure.

Jeffris and Johnson [12] investigated the effect of surface roughness upon friction between two lubricated rollers. They concluded that the measured coefficient of friction showed remarkably little variation throughout the whole range of experimental conditions for Hertzian pressures in excess of 175 kpsi. At lower Hertzian pressures, the surface roughness effect can be substantial.

A rather interesting qualitative explanation of the velocity, rate of shear, viscosity and temperature variations across the film thickness of an elastohydrodynamic contact was offered by Plint [13]. He postulated that at the entrance of the contact zone, the rate of shear, viscosity and temperature are constant across the film thickness and the velocity profile is linear. As the thermal effects take over, the temperature at the mid-film increases and the viscosity is at a minimum at this position. This thermal effect causes the velocity profile to distort into a cusp such that the rate of shear becomes a maximum at the mid-film. A ceiling curve similar to that of Johnson and Cameron's was also found in Plint's experimental friction data.

Dowson and Holmes [14] modified Crook's four-disk machine and investigated the effect of surface quality upon the traction characteristics of rolling and sliding contacts. They showed that the friction initially decreases with surface roughness, reaches a minimum, and then increases steadily with surface roughness. Unlike Jeffris
and Johnson's conclusion on the effect of surface roughness on friction, Dowson and Holmes found that the influence of surface quality is quite pronounced. However, these two results may not be in direct contradiction since the loads used by Dowson and Holmes were much smaller than those used in Jefferis and Johnson's experiments.

Recently, Dyson [15] has made a pioneering study analyzing the frictional force in an elastohydrodynamic contact by considering the lubricant as a viscoelastic liquid. He simplified his analysis by dividing the friction versus sliding speed curve into three regions, as shown in Figure 1.5: the linear region, where the frictional force varies linearly with the sliding speed; the non-linear ascending region, where the slope of the friction curve reduces rapidly as the sliding speed increases; and the thermal region, where the frictional force decreases with the sliding speed. The results of this study are most encouraging and have inspired the author's investigation of the rheological effects in an elastohydrodynamic lubricated contact.

The friction analysis presented in this thesis describes the rheological behavior of the lubricant in an elastohydrodynamic concentrated contact in terms of two viscoelastic models. These models represent the separate effects of non-Newtonian behavior and the transient response of the fluid.

A unified description of the non-Newtonian shear rate dependence of the viscosity is presented in Chapter II as a new hyperbolic liquid model. The hyperbolic model is based upon a shear viscoelastic liquid model with the addition of a limiting value of shear stress. The limiting shear stress is related to the high frequency limiting shear modulus.
Figure 1.5. Friction versus sliding speed curve. Curve from Dyson [15].
of the lubricant $G_\infty$, as proposed by Dyson [15].

The transient response of the viscosity, following the rapid pressure rise encountered in the contact, is described in Chapter III by a compressional viscoelastic model of the volume response of a liquid to an applied pressure step. Kovacs [16] first investigated this non-linear model for the volume creep of polymer melts.

The governing equations, the fluid property functions and the technique used to calculate the tractive force transmitted during sliding between the two surfaces of a rolling disk machine are developed in Chapter IV. The experimental investigation is detailed in Chapter V and the analytical and experimental results are discussed and correlated in Chapter VI.
A friction analysis based upon a Newtonian lubricant having a viscosity varying with the statically measured pressure and temperature can yield a frictional coefficient far greater than those measured. There is little doubt that the fluid ceases to be Newtonian. Thus, a realistic friction analysis must consider a non-linear relationship between shear stress and shear rate.

A major difficulty in predicting the friction for elastohydrodynamic lubrication is the lack of data available for the physical properties of the lubricant at the extreme values of pressure, temperature and shear rate encountered in the concentrated contact. It is very difficult to make direct measurements of shear stress versus shear rate in continuous shear under EHD conditions. Therefore, the physical property data must come from other fields. One source is the study of supercooled liquids under oscillatory shear. A correlation between this data and the behavior under conditions of continuous shear, as well as the restrictions of such a correlation, are discussed in this chapter. A hyperbolic shear stress-strain function is then proposed as a useful non-linear model for use under elastohydrodynamic conditions.

2.1 Shear Viscoelasticity

The phenomenological theory of viscoelasticity attempts to describe the mechanical behavior of a material in terms of time-dependent, or frequency-dependent, functions which relate the stress in
the material to the deformation. Classical elasticity theory for solids is based on Hooke's Law which requires the stress to be directly proportional to the instantaneous strain but independent of the rate of strain. Classical hydrodynamic theory is based on Newton's Law. Newton's Law states that the steady-state shear stress in a liquid is directly proportional to the instantaneous rate of strain but independent of the strain itself. Many materials closely follow the behavior specified by these laws. It is often impossible, however, to characterize a material by either of the two classical types of behavior. Substances which exhibit both solid-like and liquid-like properties show viscoelastic behavior.

The term viscoelastic is used to describe the properties of any material which is able to store energy in elastic deformation and dissipate energy as heat. If the strain and strain rate are kept sufficiently small, so that in a given experiment the ratio of stress to strain is a function of time only and independent of the stress level, the material shows linear viscoelastic behavior. Most of the physical properties of viscoelastic materials have been determined by oscillatory shear experiments. Linear viscoelastic behavior is easily obtained in these experiments since the amplitude of deformation is extremely small.

The work of Gross [17] and Alfrey [18] are examples of the large literature concerning the mathematical aspects of the phenomenological theory of linear viscoelasticity. It is more appropriate here to develop the subject in terms of simple mechanical models. The model approach is easier to understand and more closely related to the physical behavior of the materials.
2.2 Viscoelastic Functions

In most of the high frequency techniques used for measuring the viscoelastic properties of liquids, a plane shear wave is propagated through the liquid. The shear stress $\tau$ and the shear strain $\gamma$ are related by a complex quantity, the shear modulus

$$ G^* = \frac{\tau}{\gamma} \quad (2.1) $$

In a Hookean solid, the shear modulus is a real quantity since the stress varies in phase with the strain. In a Newtonian liquid, the stress is $90^\circ$ out of phase with the strain. In the latter case, the shear modulus is an imaginary quantity and is determined from Newton's Law. The strain rate is represented by

$$ \dot{\gamma} = \frac{d}{dt}(\gamma_o e^{i\omega t}) = i\omega \gamma \quad (2.2) $$

and therefore the stress is calculated as

$$ \tau = \eta \dot{\gamma} = i\omega \gamma \eta \quad (2.3) $$

The shear modulus is now calculated by its definition, equation (2.1).

$$ G^* = i\omega \eta \quad (2.4) $$

For a viscoelastic material, the stress and strain differ by a phase angle between $0^\circ$ and $90^\circ$. Therefore, the frequency-dependent shear modulus is a complex quantity with both real and imaginary components, as represented by

$$ G^*(i\omega) = G'(i\omega) + i G''(i\omega) \quad (2.5) $$

The shear modulus will not have the simple form given in equation (2.4) for a Newtonian liquid except at low frequencies where sufficient time is available during each stress cycle for viscous flow to occur. At higher frequencies, the time required for molecular translation becomes comparable with the period of the stress cycle and
the liquid exhibits a shear rigidity. At sufficiently high frequencies, the behavior will be purely elastic. There is no molecular transition during each cycle and, consequently, the energy loss due to viscous flow is negligible. Under these conditions, the liquid behaves like a glass.

The real component of the complex modulus $G'$, the ratio of the stress in phase with the strain to the strain, is called the storage modulus because of its association with the storage and release of elastic energy. The imaginary component $G''$, the ratio of the stress $90^\circ$ out of phase with the strain to the strain, is called the loss modulus because of its association with the dissipation of energy as heat by viscous flow.

The modulus components for a liquid have the following limits.

At low frequencies where the behavior is purely viscous, or Newtonian:

$$\lim_{\omega \to 0} G'(\omega) = 0$$
$$\lim_{\omega \to 0} G''(\omega) = \omega \eta$$

At high frequencies where the behavior is purely elastic:

$$\lim_{\omega \to \infty} G'(\omega) = G_\infty$$
$$\lim_{\omega \to \infty} G''(\omega) = 0$$

where $G_\infty$ is the limiting elastic modulus.

The shear mechanical impedance $Z^*$, defined as the ratio of shear stress to particle velocity, is the quantity most easily measured in the oscillatory experiments. It is mathematically related to the shear modulus by the equations governing shear wave propagation through a liquid medium. Barlow and Lamb [19] show this relationship to be
where \( \rho \) is the density of the liquid.

For a Newtonian liquid, where \( G^* \) is given by equation (2.4), the real and imaginary components of the shear mechanical impedance are given by

\[
Z^* = Z' + i Z'' = (1 + i) \sqrt{\frac{\omega \eta \rho}{2}}
\]  

Equation (2.10) allows the components of the shear modulus to be calculated from the experimentally measured components of the shear mechanical impedance as follows:

\[
G'(\omega) = \frac{(Z')^2 - (Z'')^2}{\rho} 
\]

\[
G''(\omega) = \frac{2 Z' Z''}{\rho}
\]

The liquid properties may be alternatively represented by a complex viscosity defined by

\[
\eta^*(i\omega) = \eta'(\omega) - i \eta''(\omega) = \frac{G^*(i\omega)}{i\omega}
\]

The definition requires

\[
\eta'(\omega) = \frac{G''(\omega)}{\omega}
\]

and

\[
\eta''(\omega) = \frac{G'(\omega)}{\omega}
\]

The low frequency limit of the dynamic viscosity \( \eta' \) is \( \eta \), the steady flow Newtonian viscosity.

The complex compliance of \( J^* \) is the inverse of the complex modulus.

\[
J^*(i\omega) = J'(\omega) - i J''(\omega) = \frac{1}{G^*(i\omega)}
\]
It follows that

\[ J' = \frac{G'}{(G')^2 + (G'')^2} \]  

and

\[ J'' = \frac{G''}{(G')^2 + (G'')^2} \]

The real component \( J' \) is the storage compliance and \( J'' \) is the loss compliance.

The inverse of the complex viscosity is the complex fluidity \( \mu^* \).

\[ \mu^*(i\omega) = \mu'(\omega) + i\mu''(\omega) = \frac{1}{\eta^*(i\omega)} \]  

The definitions and interrelations of the viscoelastic functions are summarized in Table 2.1.

2.3 The Maxwell Model

It is often convenient to visualize the behavior of a complex material in terms of models. The basic mechanical model elements are a coiled spring to represent Hookean elastic deformation and a dashpot to represent Newtonian viscous flow. Extension of the elements is analogous to shear strain and the associated force is analogous to the shear stress.

The combination of a spring in series with a dashpot was studied by Maxwell [20]. This simple model, shown in Figure 2.1, exhibits both viscous and elastic behavior. Viscous flow in the dashpot with negligible extension of the spring takes place if the extension rate is small. If the model is rapidly extended and immediately released, the deformation is purely elastic since sufficient time is not available for flow to occur in the dashpot. Between these extremes, the
Table 2.1

VISCOELASTIC FUNCTIONS IN OSCILLATORY SHEAR

\[ \tau = \text{shear stress acting in} \ x-\text{direction on} \ x-z \ \text{plane} \]

\[ \xi = \text{particle displacement in} \ x-\text{direction} \]

\[ \gamma = \frac{\partial \xi}{\partial y} = \text{shear strain} \]

\[ \dot{\xi} = \frac{\partial \xi}{\partial t} = \text{particle velocity in} \ x-\text{direction} \]

\[ \dot{\gamma} = \frac{\partial \gamma}{\partial t} = \frac{\partial \xi}{\partial y} = \text{shear rate, rate of strain} \]

\[ \omega = \text{angular frequency} \]

DEFINITIONS:

Complex shear modulus: \[ G^*(i\omega) = \frac{\tau}{\gamma} = G'(\omega) + iG''(\omega) \]

Complex mechanical impedance: \[ Z^*(i\omega) = -\frac{\tau}{\dot{\xi}} = Z'(\omega) + iZ''(\omega) \]

Complex viscosity: \[ \eta^*(i\omega) = \frac{\tau}{\dot{\gamma}} = \eta'(\omega) - i\eta''(\omega) \]

Complex shear compliance: \[ J^*(i\omega) = \frac{\gamma}{\tau} = J'(\omega) - iJ''(\omega) \]

Complex fluidity: \[ \mu^*(i\omega) = \frac{\dot{\gamma}}{\tau} = \mu'(\omega) + i\mu''(\omega) \]

INTERRELATIONS:

\[ G^*(i\omega) = \frac{(Z^*)^2}{\rho} = i\omega \eta(i\omega) = \frac{1}{J^*(i\omega)} = \frac{i\omega}{\mu(i\omega)} \]
behavior will be a combination of both the elastic and the viscous modes.

![Maxwell element diagram](image)

**Figure 2.1.** The Maxwell element. The spring corresponds to a shear modulus $G$ and the dashpot corresponds to a viscosity $\eta$.

The basic equations of motion for the components of the model are:

\[ \tau = \eta \dot{\gamma}_N \] (2.21) for the dashpot; and \[ \tau = G \gamma_H \] (2.22) for the spring, where

- $\tau$ = the applied stress
- $\dot{\gamma}_N$ = the rate of extension of the dashpot
- $\gamma_H$ = the extension of the spring

The rate of extension of the spring is $\dot{\gamma}_H = \dot{\tau}/G$ and the total rate of extension is then

\[ \dot{\gamma} = \dot{\gamma}_N + \dot{\gamma}_H = \frac{\tau}{\eta} + \frac{\dot{\tau}}{G} \] (2.23)

or

\[ \tau + \frac{\eta}{G} \dot{\tau} = \eta \dot{\gamma} \] (2.24)

Equation (2.24) is the constitutive equation of the Maxwell element. The ratio $\eta/G$ has the dimensions of time and is called the Maxwell relation time $\lambda$.

\[ \lambda = \frac{\eta}{G} \] (2.25)
For sinusoidal variations of stress and strain of frequency $\omega$, equation (2.24) becomes

$$\tau + i\omega\gamma = i\omega\gamma$$  \hspace{1cm} (2.26)

The complex shear modulus is evaluated from equation (2.1) as

$$G^*(i\omega) = \frac{i\omega\gamma}{1 + i\omega\lambda}$$  \hspace{1cm} (2.27)

Rationalizing this expression yields

$$G^*(i\omega) = \frac{\omega\gamma + i\omega\eta}{1 + \omega^2\lambda^2}$$  \hspace{1cm} (2.28)

and substituting for $\eta$ from equation (2.25) gives the final form of the complex shear modulus

$$G^*(i\omega) = G \cdot \frac{\omega^2 + i\omega\lambda}{1 + \omega^2\lambda^2}$$  \hspace{1cm} (2.29)

The storage modulus is

$$G'(\omega) = G \cdot \frac{\omega^2}{1 + \omega^2\lambda^2}$$  \hspace{1cm} (2.30)

which reduces to $G'(\omega) = G$ in the limit as $\omega \to \infty$; but this limiting value has been defined as $G_\infty$. Thus, the spring in the Maxwell element corresponds to the instantaneous or limiting high frequency shear modulus of a liquid. The loss modulus is given by

$$G''(\omega) = G_\infty \cdot \frac{\omega\lambda}{1 + \omega^2\lambda^2}$$  \hspace{1cm} (2.31)

which in the limit as $\omega \to 0$ becomes $G''(\omega) = G_\infty \cdot \omega\lambda = \omega\eta$. The dashpot of the Maxwell element therefore corresponds to the steady flow viscosity of a liquid. In normalized form, the variation with frequency of the modulus components and the dynamic viscosity is given by equations (2.32), (2.33) and (2.34).

$$\frac{G'(\omega)}{G_\infty} = \frac{\omega^2\lambda^2}{1 + \omega^2\lambda^2}$$  \hspace{1cm} (2.32)
The complex compliance of $J^*$ for the Maxwell element is given by the simple expression

$$J^*(i\omega) = \frac{\gamma}{\tau} = \frac{1 + i\omega\lambda}{i\omega\eta} = \frac{1}{G_\infty} - i\frac{1}{\omega\eta}$$  \hspace{1cm} (2.35)$$

The frequency variation of the modulus components and the dynamic viscosity for the Maxwell element are shown in Figure 2.2.

Gruber and Litovitz \[21\] have postulated that a Maxwell element can predict the behavior of certain liquids. The viscosity of these liquids is governed primarily by the energy required for a molecule to surmount the potential barrier due to interaction with its nearest neighbors, and jump from one site in the liquid to another. The steady flow viscosity of such a liquid is given by the Arrhenius equation:

$$\ln \eta = A + B/T$$  \hspace{1cm} (2.36)$$

where $T$ is the absolute temperature.

For liquids which have viscosities above about 0.1 poise, however, the viscosity is primarily a function of the relative availability of free volume as described by Barlow, Lamb and Matheson \[22\]. Therefore, the Arrhenius viscosity-temperature relation and the Maxwell description of viscoelastic relaxation are not adequate governing equations. The lubricant in an EHD concentrated contact is in a state where the viscosity is limited by available free volume, and therefore, another viscoelastic liquid model proposed for such liquids by Barlow, Erginsav and Lamb \[23\] must be investigated.
Figure 2.2. The frequency variation of the modulus components and the dynamic viscosity for the Maxwell element.
2.4 The B. E. L. Liquid Model

Barlow, Lamb and Matheson [22] have shown that liquids having viscosities above about 0.1 poise obey the Doolittle free-volume equation:

\[ \ln \eta = A + B \frac{v_o}{v_f} \quad (2.37) \]

where

\[ \eta = \text{viscosity} \]

\[ v_o = \text{occupied volume} \]

\[ v_f = \text{free volume} \]

\[ A, B = \text{constants of a given liquid} \]

The specific volume \( v = v_o + v_f \) and the density is a linear function of temperature. Therefore, equation (2.37) becomes

\[ \ln \eta = A' + B'(T - T_o) \quad (2.38) \]

where

\[ \eta = \text{viscosity at temperature } T \]

\[ T = \text{absolute temperature} \]

\[ T_o = \text{reference temperature, at which there would be no free volume} \]

\[ A', B' = \text{constants for a given liquid} \]

Barlow, Lamb, Matheson, Padmini and Richter [25] and Barlow, Erginsav and Lamb [23] have demonstrated that the viscoelastic properties for a large number and wide variety of liquids, which obey the Doolittle viscosity-temperature relation, can be represented by two standard curves: \( Z'/(\rho G)_{\infty}^{\frac{1}{2}} \) and \( Z''/(\rho G)_{\infty}^{\frac{1}{2}} \) versus \( \log_{10}(\omega \eta / G_{\infty}) \).

Figure 2.3 shows that the experimental results for many liquids are indistinguishable when plotted in this manner. This suggests a simple underlying phenomenological explanation. Barlow, Erginsav and Lamb propose a new liquid model consisting of the parallel combination of
Figure 2.3. Normalized plots of $\frac{Z'}{(\rho G_\infty)^{1/2}}$ and $\frac{Z''}{(\rho G_\infty)^{1/2}}$ versus $\log_{10} \left( \frac{\eta}{G_\infty} \right)$.

Curve from Lamb [26].
the shear mechanical impedances for a Newtonian liquid and a Hookean solid. The shear mechanical impedances result from equations (2.8), (2.10) and (2.11). Thus,

\[ Z_N = (1 + i) \left( \frac{\omega \eta}{2} \right)^{\frac{1}{2}} \]  

(2.39)

for a Newtonian liquid and

\[ Z_H = (\rho G) \]  

(2.40)

for a Hookean solid.

Accordingly, the components of the shear mechanical impedance are given by:

\[ Z' = \frac{(\rho G)^{\frac{1}{2}}(\omega \eta/2G)_{\infty}^{\frac{1}{2}}[1 + (2\omega \eta/G)_{\infty}^{\frac{1}{2}}]}{[1 + (\omega \eta/2G)_{\infty}^{\frac{1}{2}}]^{2} + (\omega \eta/2G)_{\infty}} \]  

(2.41)

\[ Z'' = \frac{(\rho G)^{\frac{1}{2}}(\omega \eta/2G)_{\infty}^{\frac{1}{2}}}{[1 + (\omega \eta/2G)_{\infty}^{\frac{1}{2}}]^{2} + (\omega \eta/2G)_{\infty}} \]  

(2.42)

The components of the shear modulus \( G^* \) and the compliance \( J^* \) for the B. E. L. model are given by:

\[ G' = \frac{4G(\omega \eta/2G)_{\infty}^{3/2}[1 + (\omega \eta/2G)_{\infty}^{\frac{1}{2}}]}{\left\{ [1 + (\omega \eta/2G)_{\infty}^{\frac{1}{2}}]^{2} + (\omega \eta/2G)_{\infty}\right\}^{2}} \]  

(2.43)

\[ G'' = \frac{2G(\omega \eta/2G)_{\infty}\left[1 + (\omega \eta/2G)_{\infty}^{\frac{1}{2}}\right]}{\left\{ [1 + (\omega \eta/2G)_{\infty}^{\frac{1}{2}}]^{2} + (\omega \eta/2G)_{\infty}\right\}^{2}} \]  

(2.44)

\[ J' = \frac{1}{G_{\infty}} + \frac{1}{(\omega \eta G_{\infty}/2)^{\frac{1}{2}}} \]  

(2.45)

\[ J'' = \frac{1}{\omega \eta} + \frac{1}{(\omega \eta G_{\infty}/2)^{\frac{1}{2}}} \]  

(2.46)
Finally, the dynamic viscosity is given by

\[ \eta' = \frac{G''}{\omega} = \frac{\eta \left[ 1 + \left( \omega \eta / 2G_\infty \right)^{\frac{1}{2}} \right]}{\left[ \left( 1 + \left( \omega \eta / 2G_\infty \right)^{\frac{1}{2}} \right)^2 + \left( \omega \eta / 2G_\infty \right)^2 \right]^{\frac{1}{2}}} \]  

(2.47)

The variations of the storage modulus, the loss modulus and the dynamic viscosity with frequency, calculated according to equations (2.43), (2.44) and (2.47), are graphically displayed in Figure 2.4. As compared with the results for a Maxwell element, displayed in Figure 2.2, the B. E. L. liquid model has a longer relaxation time. This is consistent with the results of previous correlations based upon distributions of Maxwell elements.

The curves plotted through the data points of Figure 2.3 are calculated according to the B. E. L. liquid model from equations (2.41) and (2.42). There is excellent agreement with the experimental results.

2.5 Relationship of Continuous and Oscillatory Shear

Dyson \[ \text{[15]} \] has had considerable success in correlating the results of elastohydrodynamic lubrication experiments with the properties of fluids experimentally determined in oscillatory shear. Dyson bases his comparison on a simplification of Oldroyd's \[ \text{[39]} \] theory of the steady motion of an idealized liquid.

The analysis postulates that a simple continuous shear deformation includes a rotation of the liquid elements. It is therefore necessary to refer all equations that describe its viscoelastic behavior in continuous shear to reference axes which rotate with the element of fluid. The rotating axes yield additional time derivative terms in the equation of motion of the fluid and thus additional deformations. These equations are solved subject to the velocity boundary conditions,
Figure 2.4. The frequency variation of the modulus components and the dynamic viscosity for the B. E. L. liquid model.
to determine the stresses in the fluid. Finally, the normal stresses are described with reference to the fixed axes.

Dyson's [27] simplification of the Oldroyd parameters permits the normal stresses for a fluid with relaxation time \( \lambda = \eta / G_\infty \), in simple laminar shear, to be expressed in terms of one parameter \( K \):

\[
\begin{align*}
\frac{P_{xx}}{G_\infty} &= \frac{2}{K^2} \cdot \frac{K^2 D^2 \lambda^2}{1 + K^2 D^2 \lambda^2} \\
\frac{P_{xy}}{G_\infty} &= \frac{1}{K} \cdot \frac{K D \lambda}{1 + K^2 D^2 \lambda^2} \\
\end{align*}
\]

(2.48)

(2.49)

\[P_{yy} = P_{zz} = 0\]  

(2.50)

where \( P_{--} \) = normal stresses

\( G_\infty \) = limiting shear modulus

\( K \) = parameter of the analysis

\( \lambda \) = Maxwell relaxation time

\( D \) = shear rate

\( x \) = direction of flow

\( y \) = direction of velocity gradient

\( z \) = direction normal to both \( x \) and \( y \)

Equations (2.32) and (2.33), repeated below, have been derived for a Maxwell fluid subject to oscillatory shear.

\[
\frac{G'}{G_\infty} = \frac{\omega^2 \lambda^2}{1 + \omega^2 \lambda^2} \\
\frac{G''}{G_\infty} = \frac{\omega \lambda}{1 + \omega^2 \lambda^2}
\]

(2.51)

(2.52)

Dyson observed, as a result of the comparison of equations (2.48) and (2.49) with (2.51) and (2.52), that the shear stress \( P_{xy} \) is equal
to $1/K$ of the value of $G''$ at an angular frequency $\omega = KD$. Furthermore, one half of the normal stress difference, $\frac{1}{2}(P_{xx} - P_{yy})$, should be $1/K^2$ of the value of $G'$ at an angular frequency $\omega = KD$. A comparison of the dynamic viscosity for continuous shear

$$\eta' = \frac{P_{xy}}{D} = \frac{\eta}{1 + K^2 D^2 \lambda^2}$$  \hspace{1cm} (2.53)$$

with equation (2.34), the dynamic viscosity in oscillatory shear,

$$\eta' = \frac{G''}{\omega} = \frac{\eta}{1 + \omega^2 \lambda^2}$$  \hspace{1cm} (2.54)$$

shows the variation with shear rate $D$ is the same as with angular frequency $\omega$, with $\omega$ replaced by $KD$.

The hypothesis above is checked against the results of Russel [28] in Figure 2.5. The variation of apparent viscosity is shown for the same three fluids in both oscillatory and continuous shear. Note that the two curves begin to diverge at an abscissa value between 1 and 10. This corresponds to the conditions where $G''$ reaches its maximum.

Whatever model is employed to represent the viscoelastic properties of the liquid, its application to continuous shear must be made in the rotating coordinate system. Therefore, the generalization of this analysis is stated as

$$\tau(D) = \frac{G''(\omega)}{K}$$  \hspace{1cm} (2.55)$$

Dyson's application of equation (2.55) to the B. E. L. liquid model is compared with the experimental results of Smith [29] at low shear rates in Figure 2.6. Dyson [15] reports that a constant value of $K = 7.5$ shows good correlation over all Smith's experimental conditions.
Figure 2.5. Comparison of variation of apparent viscosity in oscillatory and in continuous shear.

Oscillatory shear, $\eta/\eta_0$ against $\eta_0\omega_0/E_0$
Continuous shear, $\eta/\eta_0$ against $\eta_0D/E_0$

$E_0$ is a parameter of the order of $G_\infty$. Curve from Dyson [27].
Comparison of results of Barlow & Lamb in oscillatory shear with those of Smith in continuous shear—mineral oils, steel surfaces.

Oscillatory shear $G''$ against $\eta_0 \omega$ results of Barlow & Lamb [19]: ......, LVI mineral oil; -- --, MVI mineral oil; ———, HVI mineral oil.

Continuous shear $\dot{K_T}$ against $\eta_0 K_D$ results of Smith [29]: (figure number in original reference):

<table>
<thead>
<tr>
<th>fig. 8</th>
<th>fig. 9</th>
<th>fig. 10</th>
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<tr>
<td>$\times$ 23°C</td>
<td>$\triangledown$ 23°C</td>
<td>$\square$ 25°C</td>
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<tr>
<td>$\odot$ 100°C</td>
<td>$+$ 100°C</td>
<td>$\diamond$ 100°C</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$\triangle$ 190°C</td>
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Curve from Dyson [27].
As a result of equation (2.55), \((G_\infty/K)\), and not \(G_\infty\), will appear in the equations of motion. A new limiting shear modulus for continuous shear is now defined to include the Oldroyd-Dyson parameter \(K\):

\[
\bar{G}_\infty = \left(\frac{G_\infty}{K}\right)
\]  

(2.56)

2.6 Limiting Shear Stress

The Maxwell or B. E. L. liquid model, when applied to continuous shear, predicts a shear stress that rises to a maximum and then falls with increasing shear rate independent of thermal effects. This behavior is intuitively doubtful and Dyson [27] reviews the mathematical objections. It is suggested that this behavior would give rise to an unstable flow pattern. The correlation shown in Figure 2.5 suggests a transition to another mechanism of flow as the shear rate approaches the value which corresponds to a maximum shear stress. At this shear rate, the correlation between experimental and predicted values weakens.

As an alternative to the falling portion of the shear stress-deformation relation, the possibility of a limiting shear stress is suggested. The limiting shear stress is the maximum stress a fluid can transmit; an increase in the rate of shear can no longer cause an increase in the shearing stress. Smith [29] first suggested this behavior of a fluid analogous to plastic deformation of a solid. Plint's [13] results further suggest the existence of a limiting shear stress in an EHD fluid film. He interpreted the limiting shear stress to be the result of a discontinuous shear failure. Dyson [15] suggested the limiting shear stress be a function of the limiting shear modulus \(\bar{G}_\infty\). Figure 2.7 shows this results in a good correlation with the experimental data of Johnson and Cameron [11].
Figure 2.7. Dyson's correlation with the experimental data of Johnson and Cameron [11] at high shear rates. Curve from Dyson [15].
Consequently, it is surmised that there are two mechanisms of flow for a liquid under the conditions of continuous shear. The material properties of the liquid, as well as the transition between these two mechanisms of flow, are continuous. The liquid model for continuous shear is, therefore, a composite non-linear shear stress-strain relation. It is comprised of a viscoelastic relation for shear rates up to the value predicting the maximum shear stress, and a limiting shear stress equal to this maximum at higher shear rates.

2.7 Hyperbolic Liquid Model

Barlow and Lamb [19] investigated the viscoelastic relaxation in three mineral oils of different viscosity index and composition. The experimental results, shown in Figure 2.8, show slight deviations from the B. E. L. liquid model. The experimental results are displaced to higher values of frequency and lower values of the loss modulus.

To add flexibility in the analysis, a "hyperbolic" shear stress-strain relation is used which allows easy changes in the limiting shear stress or the rate of rise to this limit. The relation has the additional feature of providing a smooth transition to the flow dominated by a limiting shear stress. The model is mathematically represented by

\[
\Omega = \frac{c^2}{1 + a} \cdot \left(1 + \frac{c}{\tau - G_\infty}\right)^2 - c \quad (2.57)
\]

where
- \(\tau\) = shear stress
- \(G_\infty\) = limiting shear modulus

\[
\Omega = \frac{n}{c} \cdot \frac{\partial u}{\partial y}, \text{ dimensionless shear rate} \quad (2.58)
\]
Figure 2.8. Components of the shear modulus calculated from measured values of $Z'$ and $Z''$: $\cdots\cdots\cdots$, h.v.i.; $\cdots\cdots\cdots$, m.v.i.; $\cdots\cdots\cdots$, l.v.i. Curve from Barlow and Lamb [19].
c = limiting shear stress/limiting shear modulus ratio
a = rise parameter, rate of rise to limiting shear stress
decreases as a increases

Four models of interest, the hyperbolic model for c = .25 and c = .20, and the Maxwell and B. E. L. - limiting shear models are illustrated in Figure 2.9.

The hyperbolic liquid model has the following limiting values:

\[
\lim \frac{\tau}{G_\infty} = 0 \quad \Omega \to 0 \quad (2.59)
\]

\[
\lim \left[ \frac{d}{dD} \frac{\tau}{G_\infty} \right] \cdot \frac{G_\infty}{G} = \eta \quad \Omega \to 0 \quad (2.60)
\]

\[
\lim \frac{\tau}{G_\infty} = c \quad \Omega \to \infty \quad (2.61)
\]

\[
\lim \left[ \frac{d}{dD} \frac{\tau}{G_\infty} \right] \quad \Omega \to \infty \quad (2.62)
\]

For the case of a = 0, equation (2.57) reduces to the true hyperbola:

\[
\Omega = \frac{\frac{c \cdot \tau}{G_\infty}}{c - \frac{\tau}{G_\infty}} \quad (2.63)
\]
LIQUID MODELS

Figure 2.9. Comparison of liquid models.
CHAPATER III
TRANSIENT VISCOSITY

The maximum pressure in the lubricant film between highly loaded contacts may be as high as 250,000 psi. The lubricant film is therefore subjected to a large pressure transient as it passes through the contact, and the equilibrium viscosity at the maximum pressure is several orders of magnitude greater than the atmospheric pressure value.

Measurement of the tractional force between the two contact surfaces at low values of slip enables an "effective viscosity" of the lubricant to be calculated. At high rolling speeds, this effective viscosity is found to be lower than the value calculated from the equilibrium value of the viscosity as a function of pressure. The effective viscosity also decreases with increasing rolling speed, in a manner which is not adequately explained by either viscous heating of the lubricant film or by variation of the viscosity as a function of shear rate.

Fein [30] has suggested the failure of the lubricant viscosity to respond to the rapid pressure changes encountered in the contact area could be an explanation for the low values of effective viscosity which are observed. His analysis shows that under certain conditions the time of transit of the lubricant through the contact zone could be small compared with the time required for the lubricant to reach a state of equilibrium following an applied pressure step. Consequently, the compression of the lubricant never reaches the equilibrium state corresponding to the peak pressure, and the viscosity has a lower value than that measured under equilibrium conditions. An increase in the rolling speed reduces the residence time of the lubricant in the contact
zone. This results in an even lower value for the viscosity attained by the lubricant, and a consequent decrease in the effective viscosity.

Chapter III is an analysis of the effect of compressional visco-elasticity on the pressure-induced viscosity changes that occur in concentrated contact lubrication. The variation of viscosity with time, following an applied step in pressure, is described by a non-linear model proposed by Kovacs [16] for the volume creep of polymer melts.

3.1 Compressional Viscoelasticity

The response of a liquid to a rapid change in pressure consists of an instantaneous volume change, followed by a time-dependent volume change. The instantaneous change is attributed to the elastic compression of the liquid "lattice", while the time-dependent response is attributed to molecular rearrangements. The instantaneous response of the liquid, when the experimental time scale is small compared with the time required for molecular rearrangements, is characterized by a bulk modulus $K_\infty$. When the experimental time scale is large compared with the molecular rearrangement time, the bulk modulus has a lower value, the equilibrium value $K_e$. This behavior may be represented by the simple models shown in Figure 3.1.

Model A is widely used when volume relaxation is investigated as a function of frequency. The overall modulus then rises from a low frequency value $K_o$ to a high frequency limiting value $K_\infty = K_o + K_2$. $K_2$ is the high frequency value of the real part of the complex relaxational modulus, $K_\Re(j\omega) = K'_\Re(\omega) + i K''_\Re(\omega)$. For model A, the total bulk modulus is given by the expression
Figure 3.1. Models for compressional viscoelasticity.
and the relaxation time

\[ \lambda_v = \frac{\eta_v}{\kappa_2} \]  

(3.2)

where \( \eta_v = \) volume viscosity
\( \omega = \) angular frequency

Model B is more suited to a description of the change in volume following a sudden increase in pressure, volume creep, since the instantaneous and time-dependent parts of the response are easily separated. The response is more simply expressed in terms of the overall compressibility, the reciprocal of the bulk modulus, given by equation (3.3) as a function of frequency.

\[ \frac{1}{K} = \frac{1}{K_\infty} + \frac{1}{K_f (1 + i\omega \lambda_f)} \]  

(3.3)

\( K_f \) is a modulus associated with molecular rearrangements corresponding to changes in the free volume and \( \lambda_f \) is the retardation time given by

\[ \lambda_f = \frac{\eta_f}{K_f} \]  

(3.4)

The viscosity \( \eta_f \) is associated with the changes in the free volume.

The low frequency or equilibrium modulus \( K_0 \) is obtained from equation (3.3) for \( \omega = 0 \).

\[ \frac{1}{K_0} = \frac{1}{K_\infty} + \frac{1}{K_f} \]  

(3.5)

or

\[ K_0 = \frac{K K_f}{K_\infty + K_f} \]  

(3.6)

Models A and B describe the same behavior and a comparison of
equations (3.1) and (3.3) yields the following additional relations between the parameters of the two models:

\[ K_\infty = K_0 + K_2 \]  

(3.7)

\[ \eta_f = \eta_v \left( \frac{K_\infty}{K_2} \right)^2 \]  

(3.8)

It follows from equations (3.5) and (3.7) that

\[ \frac{K_f}{K_0} = \frac{K_\infty}{K_2} \]  

(3.9)

The behavior of liquids is generally found to be more complex than that described by these simple models and a combination of several models, each with different time constant and moduli, is necessary. Alternatively, a continuous distribution of relaxation, or retardation, times may be used to characterize the liquid behavior. The introduction of a distributed spectrum causes considerable complication in the analysis and is not warranted in the present study. Model B of Figure 3.1 will be used to characterize the behavior of the lubricant.

3.2 Viscosity Response to a Pressure Step

The overall change in volume from an initial volume \( v_1 \) to a final volume \( v_2 \), caused by a pressure change \( P \), is given by the definition of the secant bulk modulus \( K_0 \):

\[ v_1 - v_2 = -\frac{v_1 P}{K_0} \]  

(3.10)

The volume change corresponding to the purely elastic deformation \((v_1 - v_i)\) is given by

\[ v_1 - v_i = -\frac{v_1 P}{K_\infty} \]  

(3.11)
Equation (3.10), with the aid of equations (3.5), (3.10) and (3.11), may be written as

\[(v_1 - v_i) + (v_i - v_2) = \frac{v_1 P}{K_\infty} + \frac{v_1 P}{K_f}\]  

(3.12)

Therefore,

\[\frac{v_i - v_2}{v_1} = \frac{P}{K_f}\]  

(3.13)

Equation (3.13) may be taken as a definition of \(K_f\).

The time dependence of this volume change is given by the parallel spring and dashpot combination of model B. The response is governed by

\[P = \frac{\eta_f}{v_1} \frac{dv}{dt} + K_f \frac{v_i - v}{v_1}\]  

(3.14)

Combining equations (3.13) and (3.14) yields

\[\frac{\eta_f}{K_f} \frac{dv}{dt} = v_2 - v\]  

(3.15)

where \(v\) varies between \(v_1\) and \(v_2\). For small changes in pressure, when \(\eta_f\) and \(K_f\) can be regarded as constants, equation (3.15) has the solution

\[v - v_2 = (v_i - v_2) \exp(-t/\lambda_f)\]  

(3.16)

and the total response to the pressure step is

\[\frac{v_1 - v}{v_1} = P \left\{ \frac{1}{K_\infty} + \frac{1}{K_f} \left[ 1 - \exp(-t/\lambda_f) \right] \right\}\]  

(3.17)

where the retardation time \(\lambda_f = \frac{\eta_f}{K_f}\).

However, for large pressure changes the parameters \(\eta_f\), \(K_\infty\) and \(K_f\) can no longer be regarded as constants. In particular, the viscosity \(\eta_f\) may be expected to change by many orders of magnitude under the pressures occurring in the contact zone. There is considerable
evidence from ultrasonic studies of liquids that the volume viscosity \( \eta_v \), and hence \( \eta_f \), is closely related to the shear viscosity \( \eta \) and has the same temperature dependence. Litovitz and Davis [31] and Tasköprülü, Barlow and Lamb [45] offer such evidence for liquids including lubricating oils. It is assumed here that \( \eta_f \) has the same dependence on the free volume as the shear viscosity. Then \( \eta_f \) is related to the free volume \( v_f \) by the Doolittle [24] equation:

\[
\ln \eta_f = A + B/f
\]  

(3.18)

where \( f = \frac{v-v_o}{v_o} \), fractional free volume  

(3.19)

\( v_o \) = specific occupied volume  

\( A, B \) = constants

The value of \( A \) is characteristic of the liquid; the value of \( B \) is usually close to unity. The occupied volume, a function of pressure, is assumed to be independent of time and thus is associated with the instantaneous bulk modulus \( K_\infty \). The variation of \( \eta_f \) with pressure is described by the parameter \( s \), defined by

\[
s = \ln \left( \frac{\eta_{f_2}}{\eta_f} \right) = B \left[ \frac{1}{f_2} - \frac{1}{f} \right]
\]  

(3.20)

where \( f_2 \) is the final volume of the fractional free volume, and \( \eta_{f_2} \) is the final value of \( \eta_f \), at pressure \( P \). If no change in the fractional free volume occurs during the instantaneous compression, the initial value of \( s \) is

\[
s_1 = \ln \left( \frac{\eta_{f_2}}{\eta_{f_1}} \right) = B \left[ \frac{1}{f_2} - \frac{1}{f_1} \right]
\]  

(3.21)
where $\eta_f$ and $f_1$ are the values of $\eta_f$ and $f$ at the initial equilibrium state when $P = 0$. The shear viscosity is also described by equations (3.18) and (3.20), so that:

$$s_1 = \ln \left( \frac{\eta_2}{\eta_1} \right)$$

(2.22)

where $\eta_1$ and $\eta_2$ are the initial and final equilibrium values, respectively, of the shear viscosity $\eta$.

Following Kovacs [16], if it is assumed that the occupied volume $v_o$ remains constant after the initial compression, the parameter $s$ is evaluated from its definition:

$$s = \frac{(v-v_2)v_o}{(v-v_o)(v_2-v_o)}$$

(3.23)

Therefore, the differential $dv$ is

$$dv = \frac{(v-v_o)^2}{Bv_o} \, ds$$

(3.24)

Equation (3.15) is written in terms of the parameter $s$ with substitutions from equations (3.20) and (3.24):

$$- \frac{\exp(-s)}{s} \frac{(v-v_o)}{(v_2-v_o)} \, ds = \frac{dt}{\lambda_2}$$

(3.25)

Noting that

$$1 - \frac{sf_2}{B} = \frac{(v_2-v_o)}{(v-v_o)}$$

(3.26)

equation (3.25) becomes

$$\frac{\exp(-s)}{s} \frac{1}{1-\left(\frac{sf_2}{B}\right)} \, ds = - \frac{dt}{\lambda_2}$$

(3.27)

where $\lambda_2$ is a retardation time characteristic of the final equilibrium.
state, given by
\[ \lambda_2 K_f = \eta_{f_2} = \eta_{f_1} \exp(s_1) \tag{3.28} \]

The term \((sf_2/B)\) in equation (3.27) is typically much less than unity, so that the expression \((1-sf_2/B)^{-1}\) may be expanded to give
\[ \frac{\exp(-s)}{s} \, ds + \exp(-s) \frac{f_2}{B} \, ds = -\frac{dt}{\lambda_2} \tag{3.29} \]
as the differential equation describing the time-dependent compression of the liquid. This may then be integrated from the initial value \(s_1\) at \(t = 0\) to an intermediate value \(s\) at time \(t\):
\[ \text{Ei}(-s) - \text{Ei}(-s_1) + \frac{f_2}{B} \left[ \exp(-s) - \exp(-s_1) \right] = \frac{t}{\lambda_2} \tag{3.30} \]
where \(\text{Ei}\) is the exponential integral. The viscosity at time \(t\) is given by
\[ \eta = \eta_2 \exp(-s) \tag{3.31} \]

Figure 3.2 shows the variation of viscosity with time predicted by equation (3.30) for the following parameters: \(P = 200,000\) psi; \(\eta_1 = 10^{-5}\) lbf-sec/in\(^2\); \(\eta_2 = 10^3\) lbf-sec/in\(^2\); \(f_2 = 0.05\); \(B = 1\). These values are typical of those experienced by a lubricant in the contact zone of a heavily loaded rolling contact. For values of \(t\) of the order of \(\lambda_2\) or less, the viscosity is seen to be significantly less than the equilibrium value.

### 3.3 Viscosity Response of a Lubricant to a Pressure Step

To determine whether the behavior described by equation (3.30) has a significant effect on the properties of the lubricant, values of the residence time of the lubricant in the contact and the retardation
Figure 3.2. Variation of viscosity with time, following an applied pressure step. Initial viscosity = $10^{-5}$ lbf·sec·in$^{-2}$; final viscosity = $10^3$ lbf·sec·in$^{-2}$. 
time $\lambda_2$ must be determined. The lubricant in a rolling contact is subjected to high pressure for a time equal to $2b/U$ where $b$ is the half-width of the contact zone and $U$ is the rolling speed. For typical values of $b = 10^{-2}$ in and $U = 100$ in/sec, the residence time is of the order of $10^{-4}$ sec.

The time constant $\lambda_2$ is characteristic of the final equilibrium state of the liquid. Values of $\eta_f$ and $K_f$ for lubricants are not available, but reasonable estimates may be made from ultrasonic data on other liquids. The viscosity $\eta_f$ is related to the volume viscosity $\eta_v$ by equation (3.8), $\eta_f = \eta_v(K_\infty/K_2)^2$. Litovitz and Davis [31] report that the ratio $K_\infty/K_2$ is of the order of 3 for many liquids. Therefore, a value for $(K_\infty/K_2)^2$ of 10 may be used. Ultrasonic studies also indicate that $\eta_v$ is closely related to the shear viscosity; a ratio of $\eta_v/\eta = 5$ has recently been reported by Barlow, Lamb and Taskoprulu [32]. The value of $\eta_f$ is then given by $50\eta_1$, where $\eta_1$ is the atmospheric pressure shear viscosity. The time constant $\lambda_2$ is given by $\exp(s_1)\eta_f/K_f = 50\eta_2/K_f$. The bulk modulus $K_f$ is related to the relaxational modulus $K_2$ by equations (3.7) and (3.9); for a ratio $K_f/K_\infty = K_\infty/K_2 = 3$, then $K_f = 6K_2$. But $K_2$ is experimentally found to be approximately equal to $4/3G_\infty$, where $G_\infty$ is the high frequency limiting shear modulus of the liquid. A value for $G_\infty$ of $4.35 \times 10^4$ psi $(3 \times 10^9$ dyn. cm$^{-2}$) has been reported by Hutton [33] for a H.V.I. lubricating oil at 30 °C giving a value of $K_f$ of $3.5 \times 10^5$ psi. This modulus will change significantly with pressure. Dyson [15] reports measurements of $G_\infty$ as a function of pressure give a typical value for $\partial G_\infty/\partial P$ of 3, and $K_\infty$ for lubricants varies in a similar manner in the Pressure-Viscosity Report [34]. If it is assumed that the ratio
\( K_f/K_0 \) remains independent of pressure, then \( K_f = (3.5 \times 10^5 + 9P) \) psi, and the retardation time \( \lambda_2 \) is given by

\[
\lambda_2 = \frac{50\eta_2}{(3.5 \times 10^5 + 9P)} \tag{3.32}
\]

For the values given above, \( \lambda_2 \) has a value of the order of \( 2 \times 10^{-2} \) sec, which is much greater than the residence time of the lubricant in the contact zone. The "instantaneous viscosity" of the lubricant will therefore be much less than the equilibrium value, resulting in greatly reduced values of effective viscosity in accordance with experimental observations.
CHAPTER IV
MATHEMATICAL FORMULATION

The present analysis of traction in elastohydrodynamic contacts includes the iterative solution of the momentum and energy equations with the fluid properties functions of pressure and temperature. The shear rate and transient time effects have been isolated as discussed in Chapters II and III.

In this chapter the momentum and energy equations are developed and the pressure profile, the film thickness and the material property functions are discussed. The set of equations developed are then solved numerically.

4.1 Geometry and Coordinates

The geometry of a typical disk machine is shown in Figure 4.1. Two cylinders of radii $R_1$ and $R_2$, rolling with velocities $U_1$ and $U_2$, respectively, are separated by a lubricant film of thickness $2h$. A closer view of the contact zone as shown in Figure 4.2 is more useful for the purposes of this analysis. The disks have deformed elastically to form a contact zone of width $2b$ and the film thickness is approximately constant with the surfaces of the disks remaining nearly parallel.

The coordinate system is defined to have the origin on the center lines of both the fluid film and the flat contact zone. The $x$-axis is the center line of the lubricant film with the positive direction in the direction of flow; while the $y$-axis, the perpendicular bisector of the flat contact zone, is arbitrarily taken positive toward the disk rolling with velocity $U_2$. The $z$-axis, not shown in Figure 4.2,
Figure 4.1. Typical disk machine geometry.
Figure 4.2. Contact zone geometry and Hertzian pressure profile.
is perpendicular to both the x and y-axes with the positive direction consistent with a right-handed Cartesian coordinate system.

The control volume of interest is defined as an element of fluid of length \( dx \) in the direction of flow, bounded by the disk surfaces in the y-direction and of unit thickness in the third direction.

### 4.2 Pressure Distribution

The pressure distribution in the contact zone is assumed to have the elliptical Hertzian dry contact profile given by

\[
p(x) = p_{Hz} \sqrt{1 - \left(\frac{x}{b}\right)^2}
\]

where

- \( p_{Hz} \) = maximum Hertzian pressure
- \( x \) = distance from the center of the contact
- \( b = \frac{4R p_{Hz}}{E} \)

= half Hertzian width

- \( R \) = effective radius of the disks
- \( E \) = effective modulus of elasticity

The deviations from this assumed distribution are mainly in the entrance zone at low pressure levels. Their effect on the sliding friction is very small and is neglected.

### 4.3 Film Thickness

The minimum film thickness in elastohydrodynamic contacts at moderate rolling speeds can be accurately predicted by the Dowson and Higginson [35] formula:

\[
h_o = \frac{1.6 \, \alpha^{0.6} \, \rho_{ent}^{0.7} \, (E)^{0.03} \, R^{0.43}}{\omega^{0.13}}
\]

(4.3)
where \( h_0 \) = minimum film thickness
\( \alpha \) = viscosity-pressure exponent
\( \eta_{\text{ent}} \) = viscosity of the lubricant at the conditions of entry to the contact
\( U \) = mean rolling speed
\( E \) = effective modulus of elasticity
\( R \) = effective radius of the disk pair
\( w \) = load per unit length of cylinder

Note that the minimum film thickness is only slightly dependent on the load \( w \) and virtually independent of the elastic modulus \( E \).

Dowson and Higginson \([36]\) suggest the parallel film thickness \( 2h \) is 20% greater than the minimum film thickness \( h_0 \).

The Dowson and Higginson prediction of film thickness is based on an isothermal analysis which is no longer adequate for heavily loaded contacts operating at high rolling speeds. Cheng \([37]\) has calculated the lubricant film thickness in the Hertzian flat for high speed and heavily loaded rolling and sliding contacts. He used a Grubin-type inlet analysis including full thermal-hydrodynamic effects. The results obtained for a wide range of load, speeds and lubricant properties showed that the loss of film due to thermal effects is strongly influenced by the rolling velocity and the inlet viscosity of the lubricant, while it is somewhat insensitive to the change of load. The presence of sliding does not have a significant influence on the calculated film thickness, whereas the rolling speed has a far more predominant effect at the inlet.

The loss of film thickness due to thermal effects can be most conveniently represented by a thermal reduction factor \( \delta_T \), which is
defined as the ratio of the actual film thickness to that predicted by isothermal theory. Run #29 of Cheng's work is most applicable to the lubricant properties of this study and has been reproduced in Figure 4.3. Cheng's parameter $Q_m$ is defined by

$$Q_m = \frac{2\eta_{ent} U^2}{k T_{ent}}$$  \hspace{1cm} (4.4)

where

- $\eta_{ent} =$ viscosity at entry conditions (lbf·sec/in$^2$)
- $U =$ average rolling speed (in/sec)
- $k =$ thermal conductivity of lubricant (Btu/°F-hr-ft)
- $T_{ent} =$ temperature of lubricant at entry (°R)

The film thickness including the thermal effects is calculated by multiplying the isothermal film thickness, based on the Dowson-Higginson formula, by the parameter $\xi_T$ determined in Figure 4.3. For example, at a rolling speed of 500 in/sec at 175 °F, equation (4.4) requires

$$Q_m = \frac{2(0.87 \times 10^{-5})(500)^2}{(0.0216)(635)} = 0.32$$

and Figure 4.3 determines the thermal reduction factor

$$\xi_T = 0.81$$

Values for other conditions are similarly calculated. The results are shown in Table 4.1.

4.4 Momentum Equation

In applying the principle of conservation of momentum to the lubricant in the contact zone, we make the following assumptions:

1. For the case of a line contact, all the variables are independent of $z$, the direction of the axes of the disks.
Figure 4.3. Curve for the determination of the thermal reduction factor $\phi_T$, taken from Cheng [37].
Table 4.1
VALUES OF THERMAL REDUCTION FACTOR AT
TYPICAL EXPERIMENTAL CONDITIONS

<table>
<thead>
<tr>
<th></th>
<th>175 °F</th>
<th>220 °F</th>
</tr>
</thead>
<tbody>
<tr>
<td>500 in/sec</td>
<td></td>
<td></td>
</tr>
<tr>
<td>$Q_m$</td>
<td>.32</td>
<td>.15</td>
</tr>
<tr>
<td>$\phi_T$</td>
<td>.81</td>
<td>.89</td>
</tr>
<tr>
<td>1000 in/sec</td>
<td></td>
<td></td>
</tr>
<tr>
<td>$Q_m$</td>
<td>1.27</td>
<td>.60</td>
</tr>
<tr>
<td>$\phi_T$</td>
<td>.58</td>
<td>.72</td>
</tr>
</tbody>
</table>
2. As compared with the lubricant film thickness, the radii of curvature of bearing components are generally very large. In the specific case of the disk machine, the radii of the disks $R_1$ and $R_2 \gg 2h$. Accordingly, all effects due to curvature of the fluid film are neglected.

3. As compared with the much larger pressure and viscous forces, the inertia and body forces of the lubricant are negligible. This implies that the pressure and viscous forces acting on the fluid are in equilibrium.

4. As compared with the other dimensions of a lubricated concentrated contact, the film thickness is very small. Therefore, the derivative of $u$ with respect to $y$ is large in comparison with all other velocity gradients.

5. The pressure gradient across the lubricant film is also insignificant due to the relative smallness of the film thickness. Accordingly, $p = p(x) \neq p(x,y)$.

The assumptions outlined above reduce the surface forces acting on a fluid element in the contact zone to those shown in Figure 4.4. The momentum equation can be derived directly from the balance of these surface forces. Equilibrium in the $x$-direction requires

$$\frac{dp}{dx} = \frac{\partial \tau_{xy}}{\partial y} \quad (4.5)$$

The shear stress $\tau_{xy}$ must have two components. The rolling of the two cylinders produces the first component, while the second component results from the difference in rolling velocities, or slip. Consider a control volume bounded in the $y$-direction by the surfaces of the disks as shown below.
Figure 4.4. Surface forces acting on a fluid element.

Figure 4.5. Forces acting on the control volume.
The size of the fluid element considered in Figure 4.4 can be increased to that of the control volume used in Figure 4.5 by integrating equation (4.5) with respect to y over the film thickness.

\[ \int_{-h}^{h} \frac{dp(x)}{dx} dy = \int_{-h}^{h} \frac{\partial \tau_{xy}}{\partial y} dy \]

or

\[ 2h \frac{dp}{dx} = (\tau_{xy})_{y=h} - (\tau_{xy})_{y=-h} \]

\[ h \frac{dp}{dx} = \tau_{roll} \quad (4.6) \]

Thus, the rolling component is independent of the slip and is a function of the pressure gradient through the contact zone. Only the component of stress that arises due to the relative sliding of the two disks is of interest in this study. This component is easily separated by neglecting the pressure gradient term of equation (4.5). For convenience, we redefine \( \tau_{xy}^{slip} = \tau \) and the final form of the momentum equation becomes

\[ \frac{\partial \tau}{\partial y} = 0 \quad (4.7) \]

The shear stress \( \tau \) is supplied by one of the rheological models considered for the lubricant. The form of the model is

\[ \tau = f\left(\frac{\partial u}{\partial y}\right) \quad (4.8) \]

Therefore at any position \( x \)

\[ \frac{\partial f\left(\frac{\partial u}{\partial y}\right)}{\partial y} = 0 \quad (4.9) \]

and

\[ f\left(\frac{\partial u}{\partial y}\right) = C_1 \quad (4.10) \]

where \( C_1 \) is a constant of integration. When a specific model is used, one can isolate \( \frac{\partial u}{\partial y} \) and integrate with respect to \( y \) from \( y = -h \) to \( y = h \). Moreover, it is assumed that the profile for \( \frac{\partial u}{\partial y} \) is symmetric with respect
to the x-axis. The symmetry of \( \frac{\partial u}{\partial y} \) allows the use of twice the integral from \( y = 0 \) to \( y = h \) for evaluation of \( U_1 - U_2 \).

At the surface \( y = h \), the fluid must have the same velocity as the disk. Therefore,

\[
u = U_2 \quad \text{at} \quad y = h \quad (4.11)
\]

Similarly, at the surface \( y = -h \), the fluid will have a velocity \( u = U_1 \). Due to the symmetry of \( \frac{\partial u}{\partial y} \), one can specify the velocity at the center line of the film as the boundary condition. Therefore,

\[
u = \frac{1}{2}(U_1 + U_2) \quad \text{at} \quad y = 0 \quad (4.12)
\]

4.5 Energy Equation

Figure 4.6 illustrates the energy transfer from the fluid element in the concentrated contact.

\[
\begin{align*}
&-\left( k \frac{\partial T}{\partial y} + k \frac{\partial^2 T}{\partial y^2} \right) \delta x \\
&\rho c \left( v T + \frac{\partial (vT)}{\partial y} \right) \delta y \\
&\rho c (u T + \frac{\partial (uT)}{\partial x} \delta x) \delta y \\
&\rho c uT \delta y \\
&-k \frac{\partial T}{\partial y} \delta x \\
&\rho c vT \delta x
\end{align*}
\]

Figure 4.6. Energy balance for a fluid element.
It is assumed that the heat generated in the fluid element will be dissipated in two modes. Convection, the first mode of heat transfer, is the process by which energy is carried out of the contact zone with the lubricant. In the second mode, heat will be conducted across the film to the disks. Since the lubricant film thickness is small in comparison with the Hertzian contact width (x-direction) and even smaller in comparison with the cylinder width (z-direction), the temperature gradients in the x and z-directions must be small in comparison with those across the film. Therefore, only conduction in the y-direction is considered. These modes of energy transfer are shown for the fluid element in Figure 4.6.

The rate of heat generation per unit volume \( q \) is given by the product of stress with the rate of strain.

\[
q = \tau \frac{\partial u}{\partial y}
\]

(4.13)

An energy balance on this fluid element requires that the net energy into the control volume be zero.

\[
- \rho c \frac{\partial (uT)}{\partial x} - \rho c \frac{\partial (vT)}{\partial y} + k \frac{\partial^2 T}{\partial y^2} - \tau \frac{\partial u}{\partial y} = 0
\]

(4.14)

Heat Transported to Control Volume  Heat Conducted to Control Volume  Heat Generated in Control Volume

where

\( T \) = temperature of the lubricant

\( \rho \) = density of the lubricant

\( c \) = specific heat of the lubricant

\( k \) = thermal conductivity of the lubricant

Continuity requires that

\[
\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0
\]

(4.15)
Therefore,

\[- \rho \varepsilon \left( u \frac{\partial T}{\partial x} + v \frac{\partial T}{\partial y} \right) + k \frac{\partial^2 T}{\partial y^2} = -\tau \frac{\partial u}{\partial y} \]  

(4.16)

Convection Cross-Conduction Viscous Heat Generation

The ratio of convection to conduction is estimated, by assuming a triangular temperature profile, to be

\[ \frac{\rho \varepsilon U h^2}{2bk} \]  

(4.17)

Equation (4.17) demonstrates that convection will have its largest effect for a maximum value of \( \frac{U h^2}{b} \). This corresponds to the condition of maximum rolling speed and minimum load.

For the thin lubricant films in EHD contacts, where \( h \ll b \), the convective heat transfer can usually be neglected. Therefore, the governing energy equation may be written as a balance of viscous heat generation and heat transported by conduction.

\[ k \frac{\partial^2 T}{\partial y^2} = -\tau \frac{\partial u}{\partial y} \]  

(4.18)

The consequences of this assumption are discussed in section 6.3.

The lubricant in contact with the disks assumes the surface temperature of the disks. Blok [38] has analyzed the problem of a moving heat source. His results demonstrate that the disk surfaces in the concentrated contact will have a mean "flash temperature" higher than the bulk temperature of the disk. Equation (4.19) is the expression Blok derived for the flash temperature.

\[ T_s - T_b = \frac{0.48 \mu w |U_1 - U_2|}{(k m c m u b)^{1/2}} \]  

(4.19)
where 

\[ T_s = \text{mean surface temperature in the contact zone,} \]

"flash temperature"

\[ T_b = \text{surface temperature entering the junction, bulk} \]

temperature of the disk

\[ k_m = \text{thermal conductivity of the disks} \]

\[ c_m = \text{specific heat of the disks} \]

\[ \rho_m = \text{density of the disks} \]

4.6 Equilibrium Viscosity Function

The equilibrium viscosity is the viscosity measured after the lubricant has reached a state of static equilibrium under a given temperature and pressure. Viscosity deserves special attention in the study of friction in concentrated contacts. Unlike other physical properties, which change only slightly with temperature and pressure, the viscosity of a lubricant can change by several orders of magnitude.

Viscosity is most simply defined by Newton's Law:

\[ \tau = \eta \dot{\gamma} \quad (4.20) \]

where 

\[ \tau = \text{shear stress (dynes/cm}^2) \]

\[ \dot{\gamma} = \text{shear rate (sec}^{-1}) \]

\[ \eta = \text{viscosity (Poise)} \]

This can be generalized to equation (4.21) for a viscoelastic fluid.

\[ \eta = \lim_{\dot{\gamma} \to 0} \left( \frac{\tau}{\dot{\gamma}} \right) \quad (4.21) \]

The viscosity of a liquid is basically the resistance of molecules to move past the force fields of neighboring molecules. It is a complicated pressure and temperature-dependent function.

The viscosity of a liquid and the rate of change of the viscosity
due to a temperature change decrease with increasing temperature. An increase in temperature of the fluid increases the thermal agitation of the molecules which, in turn, lessens the forces of attraction between molecules. Thus the viscosity decreases.

The has been considerable effort to find an accurate relationship for predicting the variation of viscosity with temperature. Some of these relationships have theoretical foundations but the empirical formulas provide the most satisfactory predictions. The viscosity-temperature relationship found by Herschel [46] is

$$\log_{10}\left(\frac{\eta}{\eta_{ref}}\right) = \beta \cdot \log_{10} T$$  \hspace{1cm} (4.22)

where \(\eta\) is the viscosity (centipoise) at the temperature \(T (^\circ F)\) and \(\eta_{ref}\) and \(\beta\) are constants. Thus the "Herschel Chart", a plot of equation (4.22) on log-log graph paper, is a straight line for a given lubricant. The equation is simple but Appeldoorn [40] has found it surprisingly accurate for oils of very different viscosities.

The viscosity-temperature data for Mobil XRM 109 F4 and Shell Turbo 33 is given in Table 4.2. This data determines the Herschel equations:

$$\log_{10} \eta = 8.974 - 3.2 \log_{10} T$$  \hspace{1cm} (4.23)

for Mobil XRM 109 F4 and

$$\log_{10} \eta = 7.3409 - 2.8 \log_{10} T$$  \hspace{1cm} (4.24)

for Shell Turbo 33. Both of the equations above may be plotted on log-log graph paper as shown in Figures 4.7 and 4.8 and used as convenient Herschel Charts.

The effect of pressure on viscosity is influenced by both the
## TABLE 4.2

**VISCOSITY-TEMPERATURE DATA**

<table>
<thead>
<tr>
<th>Mobil XRM-109 F4</th>
<th>Shell Turbo 33</th>
</tr>
</thead>
<tbody>
<tr>
<td>Viscosity (cP)</td>
<td>Viscosity (cP)</td>
</tr>
<tr>
<td>Temperature (°F)</td>
<td>Temperature (°F)</td>
</tr>
<tr>
<td>32,150</td>
<td>84</td>
</tr>
<tr>
<td>375.0</td>
<td>21</td>
</tr>
<tr>
<td>32.5</td>
<td>21</td>
</tr>
<tr>
<td>4.46</td>
<td>8.5</td>
</tr>
<tr>
<td>0</td>
<td>100</td>
</tr>
<tr>
<td>100</td>
<td>140</td>
</tr>
<tr>
<td>210</td>
<td>194</td>
</tr>
<tr>
<td>400</td>
<td>194</td>
</tr>
</tbody>
</table>
Figure 4.7. Herschel Chart for Mobil XRM 109 F4 calculated from equation (4.24).
Figure 4.8. Herschel Chart for Shell Turbo 33 calculated from equation (4.25).
pressure level and the bulk viscosity of the fluid. The same increase in pressure will have a greater effect on the viscosity at a high pressure level than at a lower level. This results from the fact that more of the free space between molecules is already taken up at the higher pressure level. The same effect is responsible for a fluid of high viscosity undergoing a greater viscosity change than a fluid of lower viscosity for the same increase in pressure.

The Pressure-Viscosity Report [34], which includes data on several paraffinic and naphthenic mineral oils, pure hydrocarbons and synthetics, is an excellent source of pressure-viscosity data.

Chu and Cameron [41] have analyzed the results of this report in an attempt to find a sufficiently accurate pressure law and correlation. The usual simple exponential law was found inadequate for paraffinic oils. Paraffinics were found to obey the law

\[(\log_{10} \eta)^{3/2} = m(p + a)\]  

and there was a simple correlation between \(m\) and \(\eta_{base}\) the base viscosity. Including this correlation, equation (4.25) becomes

\[
\log_{10} \eta = 0.18(\log_{10} \eta_{base})^{2/3} \left(p + 13.2 \sqrt{\log_{10} \eta_{base}}\right)^{2/3} 
\]

(4.26)

where \(\eta\) = viscosity in centipoise at pressure \(p\)

\(\eta_{base}\) = base viscosity at \(p = 0\)

\(p\) = pressure in kpsi

Note that this convenient form of the Chu and Cameron viscosity-pressure law automatically correlates to each lubricant through \(\eta_{base}\).

Cheng [37] has analyzed the data of the same Pressure-Viscosity Report. He used the following alternative viscosity-pressure relationship:
\[ \frac{\eta}{\eta_0} = \exp\left[ \alpha p + (\beta + \gamma p)\left(\frac{1}{T} - \frac{1}{T_0}\right) \right] \]  

(4.27)

where

- \( \eta \) = viscosity at pressure \( p \)
- \( \eta_0 \) = reference to viscosity at \( p = 0 \) and \( T = T_0 \)
- \( p \) = pressure (psi)
- \( T \) = absolute temperature (°R)
- \( \alpha \) = viscosity-pressure coefficient
- \( \beta = 5.1 \times 10^7 \alpha \)
- \( \gamma = 930 \alpha \)

Figure 4.9 exemplifies the pressure dependence of the equilibrium viscosity function according to Cheng, and Chu and Cameron. It has been calculated for the Mobil XRM 109 F4 lubricant at 175 °F.

4.7 Limiting Shear Modulus

The pressure and temperature function for the high frequency limiting shear modulus has been developed by Dyson [15] in a correlation with Smith's [29] experimental data. The development is outlined below.

Hutton [33] experimentally determined that a high viscosity index mineral oil at atmospheric pressure varies with temperature according to

\[ \frac{1}{G} = 2.52 + 0.024 T \]  

(4.28)

and referenced to conditions at 20 °C,

\[ \frac{G(T)}{G(20°C)} = \frac{3}{2.52 + 0.024 T} \]  

(4.29)

where \( G \) is in GNm\(^{-2}\) (10\(^{10}\) dynes/cm\(^2\)) and \( T \) is in °C.
Figure 4.9. Example of $\eta(p)$ for XRM 109 F4 at 175 °F calculated from equations (4.26) and (4.27).
Variation with pressure of the high frequency limiting shear modulus is more difficult to estimate, since there is insufficient lubricant data at high pressures. Guided by equation (4.29), Dyson looked for a correlation of the shear modulus with the quantity

\[
\frac{3p}{2.52 + 0.024 T}
\]

Figure 4.10 is the correlation found with Smith's experimental results. Although this correlation predicts an impossible negative value for \( \bar{G}_\infty \) at \( p = 0 \), the predicted values at higher pressures are the best available. The limiting shear modulus function, as determined from Figure 4.10, is

\[
\bar{G}_\infty(p,T) = 0.4 \left[ \frac{3p}{2.52 + 0.024 T} \right] - 10^8
\]  

(4.30)

Converting equation (4.30) into English units, one obtains

\[
\bar{G}_\infty(p,T) = \frac{1.2p}{2.52 + 0.0133(T-492)} - 1.45 \times 10^4
\]

(4.31)

where \( \bar{G}_\infty \) is now in psi, \( p \) is in psi and \( T \) is in °R. Equation (4.31) has been used in determining the limiting shear stress in the liquid models.

4.8 Numerical Solution

The numerical solution of the equations governing the friction in elastohydrodynamic lubrication is a Fortran IV coding for use on a CDC 6400 digital computer. The program is outlined in Figure 4.11 and a complete listing is given in Appendix B.

Program CONTROL is the backbone of the traction calculation calling on several subroutines as they are needed. The load is a Hertzian elliptical pressure profile, as developed in section 4.2, and the lubricant film of uniform thickness is calculated according to
Figure 4.10. Correlation of limiting shear modulus with the experimental values of Smith [29]. Curve from Dyson [15].

\[ \frac{10^{-8} G}{K_{nm}} = 3 \times 10^{-8} p + 2.52 + 0.024 T \]
Read Data and
Set Physical Constants

Initialize Variables and
Set Boundary Conditions

Calculate Load, Film Thickness and Pressure Profile

Calculate Transient Viscosity

Solve Momentum Equation for Shear Stress

Solve Energy Equation for Temperature Gradient

Solve for Temperature Profile

Has Temperature Converged?

Yes

Calculate Friction Coefficient

Numerical Output

Plotted Output

No

Figure 4.11. Simplified diagram of Program CONTROL and its subroutines.
section 4.3. The material properties of the lubricant are allowed to vary as functions of local pressure and temperature as discussed in sections 4.4 and 4.5. The variation of the lubricant properties also includes the shear rate effects proposed in Chapter II and the transient time dependence analyzed in Chapter III. An iterative solution of the momentum and energy balances (sections 4.6 and 4.7) is used to determine the shear stress, and the velocity and temperature profiles in the contact zone. The tractive force on a disk surface, resulting from a given sliding velocity, is determined by integrating the shear stress at the disk surface over the contact area. The traction coefficient is then defined as the tractive force divided by the applied normal load.

Function VISC supplies the transient value of the viscosity according to the model analyzed in Chapter III. For the purposes of computation, the contact area is divided into six equal zones, the pressure being taken as constant within each zone. For the first two zones, the pressure step is assumed to be applied at the beginning of each zone. The viscosity attained at a time corresponding to passage through half the zone is calculated, and this value is used as an average viscosity for the zone. For a rolling speed U and contact width 2b, this time is b/6U sec. For the third zone, allowance is made for the viscosity increase in the preceding zones by calculating the viscosity at a time b/2U sec. In each case, the initial viscosity, at the instant of applying the pressure step, is taken as the viscosity at atmospheric pressure and the disk temperature. For simplicity, a viscosity distribution which is symmetrical about the center of the contact is assumed, although the actual distribution is asymmetrical,
with the maximum viscosity occurring on the exit side of the center. This approximate method provides a rapid and simple method of computing the effect of the rolling speed on the viscosity of the lubricant.

If the transient effect is to be neglected for any reason, the following trivial subprogram may be substituted for Subroutines VISC, RTMI, ZERO and EXPI.

```fortran
FUNCTION VISC (P, ETA2, CODE)
    VISC = ETA2
    RETURN
END
```

Subroutine RTMI supplies the solution $s_1$ of equation (3.30)

$$
Ei(-s_1) - Ei(-s) + \frac{f_2}{B} \left[ \exp(-s) - \exp(-s_1) \right] = \frac{e}{\lambda_2}
$$

to Function VISC. Muller's iteration scheme of successive bisections and inverse parabolic interpolations is used. RTMI is available in the Vogelback Computing Center at Northwestern University. Its listing is included in Appendix B for completeness. It is a requirement of RTMI that equation (3.30) be represented as a separate function subprogram. Function ZERO meets this need.

The evaluation of the exponential integral in equation (3.30) is performed in Function EXPI. This routine computes the exponential integral for negative arguments in the range -20 to zero. For negative values of argument $x$ the exponential integral is defined by

$$
Ei(x) = \int_{-x}^{\infty} \frac{e^{-t}}{t} \, dt \quad (4.32)
$$

In equation (4.32), a polynomial approximation is obtained, for values of the argument between zero and -5, by means of the Taylor series
expansion by Luke and Wimp [42.]:

\[ \text{EXPI}(x) = \ln|x| - \sum_{v=0}^{14} b_v (-x)^v \]  \hspace{1cm} (4.33)

where

\[ b_0 = -0.57721566 \]
\[ b_1 = 1.0 \]
\[ b_2 = -0.25 \]
\[ b_3 = 0.055555520 \]
\[ b_4 = -0.010216662 \]
\[ b_5 = 0.0016666906 \]
\[ b_6 = -0.23148392 \times 10^{-3} \]
\[ b_7 = 0.28337590 \times 10^{-4} \]
\[ b_8 = -0.30996040 \times 10^{-5} \]
\[ b_9 = 0.30726221 \times 10^{-6} \]
\[ b_{10} = -0.27635830 \times 10^{-7} \]
\[ b_{11} = 0.21915699 \times 10^{-8} \]
\[ b_{12} = -0.16826592 \times 10^{-9} \]
\[ b_{13} = 0.15798675 \times 10^{-10} \]
\[ b_{14} = -0.10317602 \times 10^{-11} \]

Equation (4.34) is the exponential approximation used for arguments in the range -5 to -20.

\[ \text{EXPI}(x) = -2.658760 \times 3 \]  \hspace{1cm} (4.34)

Function PSI specifies the shear stress-strain relationship to be used in the momentum equation. The liquid model may be changed simply by replacing the deck of this function. Routines for the following three liquid models are included in the listing of the program:
1. Maxwell - Limiting Shear Stress Model
2. B. E. L. - Limiting Shear Stress Model
3. Hyperbolic Shear Stress-Strain Model.

The routine defines the function \( \gamma \) as:

\[
\gamma = \log_{10} \frac{2}{0} \frac{\partial u}{\partial y} dy
\]

where the velocity gradient \( \frac{\partial u}{\partial y} \) is a function of the shear stress, and therefore, dependent upon the liquid model.

\[
\frac{\partial u}{\partial y} = \frac{\bar{G}}{\eta} \cdot \frac{1 - \sqrt{1 - 4 \left( \frac{\tau}{\bar{G}_{\infty}} \right)^2}}{2 \frac{\tau}{\bar{G}_{\infty}}}
\]

(4.36)

for the Maxwell model;

\[
\frac{\partial u}{\partial y} \approx \frac{\bar{G}}{\eta} \left[ 55.2 \left( \frac{\tau}{\bar{G}_{\infty}} \right)^2 + \frac{\tau}{\bar{G}_{\infty}} \right]
\]

(4.37)

for the B. E. L. model; and

\[
\frac{\partial u}{\partial y} = \frac{\bar{G}}{\eta} \cdot \frac{c \frac{\tau}{\bar{G}_{\infty}}}{c - \frac{\tau}{\bar{G}_{\infty}}}
\]

(4.38)

for the hyperbolic model. The boundary conditions specified by equations (4.11) and (4.12) require

\[
\int_{0}^{h} \frac{\partial u}{\partial y} dy = \frac{(U_2 - U_1)}{2}
\]

(4.39)

Equations (4.35) and (4.39) combine to require

\[
\gamma = 0
\]

(4.40)
Subroutine SECANT determines the shear stress solution of the momentum equation by solving equation (4.40). This routine is a modification of Newton's method. For any function \( f(x) \), two initial guesses of the root \( x_1 \) and \( x_2 \) are required. A straight line is "drawn" through \( f(x_1) \) and \( f(x_2) \) and extended to cross the x-axis. This new point \( x_3 \) determines \( f(x_3) \) which is then connected with \( f(x_2) \) to determine \( x_4 \), as depicted in Figure 4.12. This process continues until the root of the function is determined. A number of checks are also included in Subroutine SECANT to both guarantee and expedite convergence.

Subroutine INTEG integrates any non-equidistantly tabulated function \( f(x_i) \) between the limits \( a \) and \( b \), where \( a \) or \( b \) must equal \( f(x_1) \). The integrated function may be defined as

\[
\text{INTEG} \left[ f(x_i) \right] = \int_{a}^{b} f(x_i) \, dx_i
\]

A method of overlapping parabolas is employed with suitable modifications to yield the fastest possible integration with second order accuracy. The development of the quadrature for this subroutine is shown in Appendix A. Subroutine INTEG is called upon to integrate the velocity gradient in the solution of the momentum equation and again, to integrate both the Laplacian and temperature gradient in the solution of the energy equation.

Subroutine PRINTS provides the numerical output of the program, and Subroutine SETPLOT issues rapid line printer plotting of the temperature profiles and traction coefficient versus slip curves. SETPLOT is a library routine of Vogelback Computing Center and will probably require considerable changes in the coding for use at another facility. Since it is not required for the traction analysis, the listing for SETPLOT is not included with that of the program.
Figure 4.12. Secant method of solving $f(x) = 0$. 
CHAPTER V
EXPERIMENTAL INVESTIGATION

The design and manufacture of a disk machine was completed as part of this study. The purpose of the experimental investigation was to gather extensive data for two new synthetic lubricants, Mobil XRM 109 F4 and Mobil XRM 177 F4. The conditions under study were those of high loads and high rolling speeds where there was a paucity of experimental data. Special emphasis was given to the effect of additives upon the frictional torque. This chapter describes the disk machine, the lubricant properties and the test procedure.

5.1 The Disk Machine

The design of the disk machine for this experimental investigation was guided by the following requirements. The disk machine must be capable of accurately measuring the tractive force transmitted across the line contact of the two disks for a wide range of loads, rolling speeds and slips. A sufficient normal load is required between the disks to insure operation in the elastohydrodynamic regime. The drive to the disks must allow easy adjustments of the mean rolling speed and the amount of sliding at the contact. The lubricant must be delivered to the contact at a controlled rate and temperature. Instrumentation is required to measure the normal and tangential forces on the disks. The angular velocities of the disks, as well as the slip or difference in the disk velocities, must also be accurately measured. Finally, the surface temperature of the disks as they enter the contact zone is required for an accurate knowledge of the friction. A detailed description of the machine designed to meet these requirements follows.
The disk machine, pictured in Figures 5.1 through 5.3, was designed with two 6-inch diameter disks. These large disks were selected to allow a direct drive system at high speeds, thus minimizing any possible vibrations. The lower disk is supported on two high-speed roller bearings which are mounted in the main frame and the upper disk is contained in a loading arm which is hinged on the frame with a spherical roller bearing.

The load is applied by an air cylinder at the far end of the loading arm. With a 30 psi air supply, a maximum Hertzian stress of 300,000 psi can be obtained for a \( \frac{1}{4} \)-inch contacting width. The applied normal load is monitored by a four-strain gauge bridge mounted on the air cylinder shaft. This is necessary for accurate measurement of the normal load, as the friction in the air cylinder is inconsistent. The loading arm was designed to permit the necessary alignment to insure a uniform load across the line contact in the axial direction.

Each of the disks is attached through flexible couplings to separate 40 hp D.C. field controlled electrical machines. The shunt-field current method of speed control is simple and efficient and the speed regulation, for a given speed adjustment, is excellent. The complete electrical circuit, schematically shown in Figure 5.4, is the Hopkinson mechanical-loss-supply feedback circuit described, for example, by Kloeffler, Kerchner and Brenneman [43]. This circuit was inspired by a related feedback system successfully used in the experiments of Jefferis and Johnson [12]. One of the D.C. machines behaves as a motor driving one of the disks. The second disk is driven by the friction force transmitted at the contact and drives
Figure 5.2. The disk machine.
Figure 5.4. Electrical connections for the disk machine.
the second D. C. machine as a generator. The energy is electrically recycled by supplying the generated voltage to the armature of the motoring machine. The losses in this cycle are mechanically supplied by a 20 hp motor connected to the double-ended armature shaft of the 40 hp D. C. motor.

The speed of the upper shaft is controlled by the small 20 hp booster motor. A wide range of rolling speeds can be obtained by varying the supply voltage and the field resistance of the booster motor. The speed differential between the two shafts is controlled by adjusting the field resistors of the 40 hp D.C. motors which are connected together across the armature terminals. This arrangement allows a wide and continuous range of sliding speeds to be easily obtained. This differs greatly from the typical two-disk machine arrangement in which the slide-to-roll ratio is fixed by a gear ratio, or the ratio of the diameters to the disks.

Jefferis and Johnson reported torsional vibration difficulties with the original design of their disk machine. Every effort was made to eliminate vibration problems in this design. The lower shaft of the disk machine can be approximated as indicated in Figure 5.5, for the purpose of determining the natural torsional frequency of the shaft. The lowest natural frequency of this system is calculated to be 55 cycles per second which is slightly above the condition existing at the maximum experimental rolling speed of 1000 in/sec. The flexible couplings used in the system were chosen, in part, for their high damping characteristics to further insure smooth operation. Vibration problems were not encountered in the course of the experiments.

The frictional torque transmitted through the line contact is
Figure 5.5. Model of the disk machine lower shaft used for vibration analysis.
measured by a torquemeter mounted in the lower shaft of the disk machine. The measured torque and the known radius of the disk are used to calculate the tangential tractive force of the contact. The torquemeter consists of a four-strain gauge bridge mounted on a calibrated torsion shaft. The electrical output signal of the torquemeter, passed from the rotating shaft through a set of slip rings, is a measure of the instantaneous torque in the lower shaft. This signal is displayed on an oscilloscope or measured by a digital voltmeter if an integrated average value is desired. The torque measured in this manner includes the frictional torque of the two lower support bearings which is accounted for as follows. The Hopkinson electrical circuit allows rotation of the disks in both directions and, therefore, pure rolling is possible. The bearing friction and any minute rolling friction are calculated from the measurements at pure rolling. The combined value is small and averages to a frictional torque corresponding to a friction coefficient of 0.002.

The angular velocity of each disk is measured by a timing wheel, seen in Figure 5.3. Each timing wheel has 100 equally spaced holes along the circumference. Light supplied from a high intensity source to the timing wheels is chopped into a stream of pulses as the timing wheel rotates. A pair of photomultiplier tubes converts these light pulses to electrical pulses which are then counted electronically. Thus the speed of the disk is measured. Fiber optic light guides are used for the transport of the light beams throughout this system. The sliding velocities are calculated as the difference of the measured velocities of the disks.

Filtered oil is supplied to the exit side of the contact allowing
the lubricant one revolution with the disk before entering the con-
junction. The lubricant is pumped from a 5 gallon supply tank, with
a thermostatically controlled electric immersion heater and circulator,
at rates up to 1 gallon per minute. The filter has a paper filter
element which removes particles down to 1 micron.

The surface temperature of the disk as it enters the contact zone
is monitored by an iron-constantan thermocouple trailing on the moving
surface. An ice bath reference junction is used with the thermocouple.
Crook [ 44 ] has demonstrated that this method gives accurate results;
and Johnson and Cameron [ 11 ] have found this method agrees closely
with the temperatures measured by a thermocouple embedded in the surface
of the disk.

The electrical output signals of the strain gauges, photomultiplier
tubes and the thermocouple are continually monitored by a scanning
digital voltmeter.

A surface trace of a disk, shown in Figure 5.6, indicates that
the disks were manufactured with a maximum peak-to-valley roughness
of 4 micro-inches.

5.2 The Lubricants

The experimental program consisted of the gathering of extensive
friction data for the two experimental fluids, Mobil XRM 109 F4 and
Mobil XRM 177 F4. Mobil XRM 109 F4 is a synthesized paraffinic hydro-
carbon base fluid. Mobil XRM 177 F4 is comprised of Mobil XRM 109 F4
formulated to improve its anti-fatigue properties. Table 5.1 is the
physical property data, kindly supplied by the Mobil Research and
Development Corporation, determined on Mobil XRM 109 F4. The proper-
ties of Mobil XRM 177 F4 are expected to be the same within experi-
mental error.
Figure 5.6 Trace of the disk surface.
Table 5.1
PHYSICAL PROPERTIES OF MOBIL XRM 109 F4

<table>
<thead>
<tr>
<th>Property</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Kinematic Viscosity, cs @ 400°F</td>
<td>6.0</td>
</tr>
<tr>
<td>@ 210°F</td>
<td>40.4</td>
</tr>
<tr>
<td>@ 100°F</td>
<td>447</td>
</tr>
<tr>
<td>@ 0°F</td>
<td>37,000</td>
</tr>
<tr>
<td>Total Acid No.</td>
<td>0.0</td>
</tr>
<tr>
<td>Flash Point, °F</td>
<td>520</td>
</tr>
<tr>
<td>Fire Point, °F</td>
<td>595</td>
</tr>
<tr>
<td>Pour Point, °F</td>
<td>-60</td>
</tr>
<tr>
<td>Density @ 100°F</td>
<td>0.8389</td>
</tr>
<tr>
<td>@ 200°F</td>
<td>0.8082</td>
</tr>
<tr>
<td>@ 300°F</td>
<td>0.7777</td>
</tr>
<tr>
<td>@ 400°F</td>
<td>0.7428</td>
</tr>
<tr>
<td>Specific Heat @ 300°F</td>
<td>0.635</td>
</tr>
<tr>
<td>@ 400°F</td>
<td>0.692</td>
</tr>
<tr>
<td>Autogeneous Ignition Temp., °F</td>
<td>760</td>
</tr>
<tr>
<td>Surface Tension</td>
<td>30.9</td>
</tr>
</tbody>
</table>
5.3 Test Procedure

The strain gauges, photomultiplier tubes, voltage supplies, and digital voltmeter must warm up and reach a stable temperature before any calibrations are performed. The lubricant supply is also heated to the desired temperature during this time. After the warm-up, the strain gauge bridges measuring the normal load and frictional torque are calibrated. The air cylinder gauges are calibrated to zero load, while the torquemeter gauges are calibrated against a shunt resistance simulating a known torque.

The oil supply is then turned on and the disk machine may be started at minimum load with the field resistances of the two 40 hp machines at equal settings. The load and rolling speed are then increased to the desired values and the bearing torque is measured at pure rolling conditions.

The sliding speed is now varied, while maintaining a constant mean rolling speed, to obtain the data for a friction versus sliding speed curve. It is easiest to keep the surface temperature within a 5 degree C range by making some high slip torque measurements first and then returning to the low and middle slip values.

5.4 Results

A typical set of experimental results is shown in Figure 5.7. The friction coefficient, defined as the tractive force divided by the applied normal load, is plotted against the sliding speed. The maximum Hertzian pressure, rolling speed and lubricant inlet temperature remain constant. The friction coefficient rises from zero at pure rolling to a maximum value and then decreases with any further increase in sliding speed.
Figure 5.7. Typical experimental results.
Complete results of the experimental study are presented with discussion in the next chapter.
CHAPTER VI
DISCUSSION OF RESULTS

A new elastohydrodynamic friction analysis has been developed in Chapters II, III and IV. The separate effects of both shear rate and time have been included. Shear viscoelasticity results in a non-Newtonian relation between the shear stress and shear rate, while compressional viscoelasticity results in a time-dependent viscosity function. A numerical solution of the momentum and energy equations, with pressure, temperature and time-dependent parameters, is achieved.

Traction measurements have been made on two synthesized hydrocarbon fluids under elastohydrodynamic conditions. The experimental apparatus and procedure have been described in Chapter V.

This chapter discusses the results of these analytical and experimental programs. A good correlation of the friction coefficients determined by analysis and experiment is shown.

6.1 Values of the Friction Coefficient Determined by Experiment

The tractive force transmitted by a thin lubricant film under elastohydrodynamic conditions has been measured for a wide range of loads and sliding speeds at high rolling speeds. Specifically, the loads ranged from 115,000 psi maximum Hertzian stress to 250,000 psi; the sliding speeds varied from zero to over 60 in/sec; the high rolling speeds were 500 and 1000 in/sec; and the oil entrance temperatures were 175 °F and 220 °F. The friction coefficient, or traction coefficient, calculated from this data for two synthetic paraffinic fluids, show variations similar to those found for other lubricating oils by
Johnson and Cameron [11], Crook [4] and Bell, and Kannel and Allen [6].

The friction coefficient rises to a maximum value with increasing sliding speed and then decreases with any further increase in the sliding speed. The coefficient is also found to increase with increasing pressure and to decrease with increasing rolling speed and temperature. Any parameter variation that results in an increase in the friction coefficient also results in the maximum friction occurring at a lower sliding speed. Examples of this behavior for both experimental fluids are shown in Figures 6.1 through 6.14 where the friction coefficient is plotted as a function of sliding speed for fixed values of maximum Hertzian pressure, rolling speed and oil inlet temperature.

This behavior may be explained in terms of the liquid model that has been developed. At low values of sliding and therefore, low shear rate, there is no appreciable temperature gradient across the lubricant film. The shear stress increases with shear rate according to the effective viscosity predicted by the compressional viscoelastic model developed in Chapter III. At slightly higher sliding speeds, the temperature rise in the fluid film is significant and cannot be neglected. For the range of conditions under study, the analysis predicts a rise of film temperature of 15 °F to 20 °F at the sliding speed corresponding to the maximum friction coefficient. For even higher sliding speeds, it is hypothesized that the mechanism of flow changes and is dominated by a pressure and temperature-dependent limiting shear modulus. The temperature at the center of the lubricant film at the highest sliding speeds is calculated to be 100 °F to 150 °F higher than the surface temperature of the disks.
EFFECT OF PRESSURE
U=500
T=175

Figure 6.1. The effect of load on the friction coefficient.
Figure 6.2. The effect of load on the friction coefficient.
Figure 6.3. The effect of load on the friction coefficient.
Figure 6.4. The effect of load on the friction coefficient.
Figure 6.5. The effect of lubricant inlet temperature on the friction coefficient.
Figure 6.6. The effect of lubricant inlet temperature on the friction coefficient.

EFFECT OF TEMPERATURE
P=200000
U=1000
EFFECT OF TEMPERATURE

$P = 2500000$

$U = 500$

Figure 6.7. The effect of lubricant inlet temperature on the friction coefficient.
Figure 6.8. The effect of rolling speed on the friction coefficient.
Figure 6.9. The effect of rolling speed on the friction coefficient.
EFFECT OF ROLLING SPEED

$P = 2000000$

$T = 175$

Figure 6.10. The effect of rolling speed on the friction coefficient.
Figure 6.11. The effect of rolling speed on the friction coefficient.
Figure 6.12. The effect of rolling speed on the friction coefficient.
Figure 6.13. The effect of rolling speed on the friction coefficient.
Figure 6.14. The effect of rolling speed on the friction coefficient.
Therefore, the friction coefficient $\mu$, which is defined as the ratio of the tractive force transmitted to the normal load, is proportional to the following:

$$\mu \propto \frac{\eta f' \left( \frac{\partial u}{\partial y} \right)}{R}$$

for sliding speeds smaller than that corresponding to the maximum value of the friction coefficient; and

$$\mu \propto \frac{c_\infty}{P}$$

for higher sliding speeds.

An increase in the pressure level results in an increased viscosity $\eta$, which accounts for the higher friction coefficient at low sliding speeds seen in Figures 6.1 through 6.4. For higher sliding speeds, the limiting shear modulus as a function of pressure must be reviewed. Figure 6.15 simplifies the relationship of Figure 4.10 and equation (4.31).

Figure 6.15. Simplified limiting shear modulus-pressure relationship.
Although the actual limiting shear modulus probably follows the dashed line, the linear portion at higher pressure levels may be given by the relationship shown. Equation (6.2) therefore yields

\[ \mu \propto m - \frac{b}{p} \]  \hspace{1cm} (6.3)

and the friction coefficient also increases with pressure at high sliding speeds.

An increase in the inlet temperature of the lubricant results in lower values of both the viscosity and limiting shear modulus. The entire friction coefficient versus sliding speed curve is thus lowered according to equations (6.1) and (6.2). This is seen in Figures 6.5 through 6.7.

As the rolling speed increases, the film thickness increases according to equation (4.3). At low sliding speeds, this higher film thickness will reduce the shear rate, which in turn reduces the shear stress according to the viscoelastic fluid model. At higher values of sliding speed, an increased film thickness results in a higher film temperature as indicated by equation (4.18). This will then lower the limiting shear modulus. Either of these results, the lower shear stress or the lower limiting shear modulus, causes lower friction coefficients as seen in Figures 6.8 through 6.14 for the two experimental lubricants.

6.2 Correlation of Values of the Friction Coefficient Determined by Experiment and Analysis

The values of the friction coefficient determined by experiment for the Mobil XRM 109 F4 synthetic paraffinic base fluid are compared with those predicted by the new analysis in Figures 6.16 through 6.23.
Figures 6.24 through 6.31 show a similar comparison for the Mobil XRM 177 F4 paraffinic fluid with anti-fatigue additives. A further correlation is shown in Figures 6.32 through 6.37 for some of the experimental data of Johnson and Cameron [11].

As in the previous section, the values of the friction coefficient are plotted as a function of the sliding speed $U_1 - U_2$, for fixed values of maximum Hertzian pressure $P$, rolling speed $U$ and oil entrance temperature $T$. The units of $P$, $U$ and $T$ are psi, in/sec and degrees F, respectively, except for the Shell Turbo 33 correlations where the oil entrance temperature is given in degrees C. Values determined by experiment are shown as data points on the curves, while values predicted by the analysis are shown by smooth curves.

The analysis predicts friction coefficients that show the same variations as observed experimentally. The friction coefficients rise to a maximum value and then decrease with increasing sliding speed; they increase with increasing pressure and decrease with increasing rolling speed and oil temperature.

Good correlation is found between the experimental data for Mobil XRM 109 F4 and Shell Turbo 33 and the values predicted by the analysis using the straight exponential viscosity function adopted by Cheng [5] and the hyperbolic liquid model with $c = .25$. This corresponds to the Barlow, Erginsav and Lamb [23] liquid model with a limiting shear stress. In most cases, the friction coefficients agree within 10%, with a few extreme cases differing by less than 25%.

It is necessary to use the hyperbolic model with $c = .20$ to obtain the same correlation for the Mobil XRM 177 F4 lubricant. This
EXPERIMENT AND ANALYSIS

P=115000
U=500
T=175

Figure 6.16. Correlation of theoretical friction with experimental data.
Figure 6.17. Correlation of theoretical friction with experimental data.
Figure 6.18. Correlation of theoretical friction with experimental data.
EXPERIMENT AND ANALYSIS

P=250000
U=500
T=220

Figure 6.19. Correlation of theoretical friction with experimental data.
EXPERIMENT AND ANALYSIS

P=115000
U=1000
T=175

Figure 6.20. Correlation of theoretical friction with experimental data.
Figure 6.21. Correlation of theoretical friction with experimental data.
EXPERIMENT AND ANALYSIS

P = 200000
U = 1000
T = 175

Figure 6.22. Correlation of theoretical friction with experimental data.
Figure 6.23. Correlation of theoretical friction with experimental data.
Figure 6.24. Correlation of theoretical friction with experimental data.
EXPERIMENT AND ANALYSIS

P = 154000
U = 500
T = 175

Figure 6.25. Correlation of theoretical friction with experimental data.
Figure 6.26. Correlation of theoretical friction with experimental data.
Figure 6.27. Correlation of theoretical friction with experimental data.
EXPERIMENT AND ANALYSIS

P = 250000
U = 500
T = 220

Figure 6.28. Correlation of theoretical friction with experimental data.
Figure 6.29. Correlation of theoretical friction with experimental data.
EXPERIMENT AND ANALYSIS

\[ P = 154000 \]
\[ U = 1000 \]
\[ T = 175 \]

Figure 6.30. Correlation of theoretical friction with experimental data.
Figure 6.31. Correlation of theoretical friction with experimental data.
Figure 6.32. Correlation of theoretical friction with experimental data.
Figure 6.33. Correlation of theoretical friction with experimental data.
Figure 6.34. Correlation of theoretical friction with experimental data.
Figure 6.35. Correlation of theoretical friction with experimental data.
Figure 6.36. Correlation of theoretical friction with experimental data.
EXPERIMENT AND ANALYSIS

P = 176000
U = 47
T = 30

Figure 6.37. Correlation of theoretical friction with experimental data.
lubricant consists of the Mobil 109 F4 as a base with an anti-fatigue polymer additive. This additive may change the limiting shear modulus function which corresponds to a change in the hyperbolic model constant c. A more likely possibility, however, is that the additive increases the film thickness. If this is the case, the shear rates would be lower in the low sliding speed region and the film temperatures would be higher in the high sliding speed region. As previously discussed in section 6.1, this would lower the entire friction coefficient curve. Thus, if the additive does cause an increase in the film thickness, the hyperbolic model with c = .25 might hold true for this lubricant also.

Figure 6.38 shows a comparison of friction versus sliding speed curves analytically determined using the viscosity relationships adopted by Cheng (equation 4.27) and Chu and Cameron (equation 4.26). The curves are extremely close at low sliding speeds but begin to diverge at higher sliding speeds as the temperature rise in the lubricant film becomes larger. The divergence is due to the higher temperature dependence of the Chu and Cameron formulation. It is of little consequence which formulation is used at low sliding speeds since the compressional viscoelastic effects dampen the effects of small changes in equilibrium viscosity. The Cheng formulation gives a slightly better correlation with all experimental data. Until extremely high pressure viscosity data is available for lubricants, there will be no other means of choosing among empirical pressure-temperature-viscosity functions.

140
EFFECT OF VISCOSITY-PRESSURE FUNCTIONS
P=200000
U=500
T=175
X CHU-CAMERON
O CHENG

Figure 6.38. Comparison of friction curves based upon different viscosity functions.
6.3 Fluid Property Profiles

In addition to calculating the friction coefficient, the numerical solution of the momentum and energy equations also determines the fluid property profiles in the lubricant film. The profiles confirm the qualitative estimates of Flint [13].

As the lubricant enters the contact zone, the temperature, and therefore the viscosity and the limiting shear modulus are constant across the film. At small sliding speeds and low pressures, the temperature across the film remains constant and equal to the disk surface temperature. The velocity profile is linear and the other property profiles are easily predicted.

Under more severe conditions such as higher sliding speeds and pressures, the thermal effects dominate the profiles. The temperature profile becomes parabolic, and at the most severe conditions, almost triangular. The central plane temperature is 100°F to 150°F higher than the disk surface temperature. This results in a sharp S-shaped velocity profile with an enormous velocity gradient at the central plane of the lubricant film. The viscosity, and usually more important under these conditions, the limiting shear modulus have minimum values on the central plane. Thus, even though the material properties and the fluid flow are continuous, the conditions are close to those that would occur in a fluid undergoing a discontinuous shear failure on the plane of minimum limiting shear stress.

Two examples of the profiles at the center of the contact zone, those of temperature, velocity, viscosity and limiting shear modulus, are shown in Figures 6.39 and 6.40 for the analysis of Mobil XRM 109 F4.
at 200,000 psi maximum Hertzian pressure, 500 in/sec rolling speed and 175 °F oil inlet temperature. Figure 6.39 is for a 2 in/sec sliding velocity where the velocity profile is no longer linear. Figure 6.40 is for a 50 in/sec sliding velocity where the shear rates at the center plane are extremely high.

6.4 Effect of Convective Heat Transfer

The simplified energy balance given by equation (4.18) was derived by assuming the heat transported by convection was negligible as compared with the heat conducted in the two disks. A ratio of convection to conduction is estimated in equation (4.17) as

\[ \frac{pcuH^2}{2 bk} \]

Convection has its largest effect for a maximum value of \( (UH^2/b) \). This corresponds to the condition of maximum rolling speed and minimum load. A program was written to include the effects of convection. Figure 6.41, the friction coefficient-sliding speed curves, include and neglect the convective heat transfer.

As expected, the convection will carry some heat from the contact zone and the lubricant will be slightly cooler. For example, under the conditions for maximum convection, the mid-film temperature at the center of the contact zone is determined by the analysis to be 5 °F cooler; that is, 210 °F compared with 215 °F, at the point of maximum friction coefficient. This only affects the friction coefficient at higher sliding speeds where the temperature gradient in the film becomes significant.

The program including the convective effects takes four times
Figure 6.39. Fluid property profiles in the concentrated contact.
Figure 6.40. Fluid property profiles in the concentrated contact.
Figure 6.41. The effect of convective heat transfer on the friction coefficient.
the time required by the simpler program which considers conduction only. The results, differing by 5-10% at a maximum, do not warrant this expenditure.

6.5 Effect of Compressional Viscoelasticity

The overall effect of compressional viscoelasticity on the friction coefficient is seen in Figure 6.42. The most prominent feature is the shifting of the maximum value to higher sliding speeds. This is the same effect a longer shear relaxation time would have on the curve. At higher sliding speeds, the flow is dominated by the limiting shear modulus. Therefore, to study the effects of compressional viscoelasticity, attention is focused on the region of low sliding speeds.

The values of traction coefficient for very low sliding speeds have been calculated and are shown in Figures 6.43, 6.44 and 6.45 plotted as a function of the ratio of sliding speed to rolling speed, \( \xi = \frac{U_1 - U_2}{U} \), for fixed values of peak pressure and \( U \). To simplify comparisons with experimental data, the calculations have been made for the conditions used by Johnson and Cameron [11]; that is, hard steel disks of 3 in diameter, a lubricant of viscosity 84 cP at atmospheric pressure and 30 °C, and maximum Hertzian pressures of 87, 110, 147, 176 and 224 x 10^3 psi.

The variation of traction coefficient with rolling speed may also be presented in terms of an effective viscosity. The effective viscosity \( \overline{\eta} \) is defined as that constant viscosity which, for a contact area of width \( 2b \) and uniform thickness \( 2h \), would give rise to the measured tractional force. Then the effective viscosity may be

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EFFECT OF
COMPRESSITIONAL VISCOELASTICITY

P=200000
U=500
T=175

• NEGL. COMP. VISC.
• INCL. COMP. VISC.

Figure 6.42. The effect of compressional viscoelasticity on the friction coefficient.
Figure 6.43. Friction coefficient versus slide/roll ratio, at peak Hertzian pressures of 87,000 psi and 110,000 psi, for different rolling speeds.
Slide / Roll Ratio, $\xi = (U_1 - U_2) / U$, $\times 10^4$

Figure 6.44. Traction coefficient versus slide/roll ratio, at peak Hertzian pressures of 147,000 psi and 176,000 psi, for different rolling speeds.
Figure 6.45. Traction coefficient versus slide/roll ratio, at peak Hertzian pressures of 200,000 psi and 224,000 psi, for different rolling speeds.
calculated from the traction coefficient $\mu$ by the expression

$$\bar{\eta} = \frac{\mu W h}{(U_1 - U_2)b} \quad (6.4)$$

Alternatively,

$$\bar{\eta} = \left(\frac{\mu}{\xi}\right) \times \frac{W h}{bU} \quad (6.5)$$

where $\mu/\xi$ is the initial slope of the traction curve when plotted as a function of the slide/roll ratio. Values of $\bar{\eta}$ calculated in this way, in the limit of zero sliding speed, are plotted as a function of rolling speed in Figure 6.46.

The calculated values of traction coefficient show a small dependence on rolling speed when plotted as a function of the ratio of sliding speed to rolling speed. Johnson and Cameron [11] report that the traction coefficient was experimentally found to depend only on the slide/roll ratio and to be independent of the rolling speed. In the present analysis, this behavior is found only at the lower pressures, and then only over a limited range of rolling speeds (Figure 6.43), although the values of traction coefficient are similar to those measured by Johnson and Cameron.

The curvature of the lines in Figures 6.43, 6.44 and 6.45 reflects the departure from Newtonian behavior of the lubricant with increasing shear rate. At low sliding speeds, the heat generated due to shearing in the lubricant is negligible. No temperature gradient exists within the lubricant film, and the temperature throughout the contact zone remains equal to the disk temperature. The decreasing slope of the traction coefficient curves with increasing sliding speed is thus a consequence of the decrease in the apparent viscosity with increasing shear rate. The effect is most marked at the higher pressures and lower
Figure 6.46. Effective viscosity versus rolling speed, in the limit of zero sliding speed, for different values of peak Hertzian pressure.
rolling speeds when the small values of film thickness result in higher values of shear rate. The variation of traction coefficient with sliding speed is shown in more detail in Figure 6.47 for a peak pressure of 176,000 psi. The behavior at the other measures conforms to the same general pattern. In Figure 6.47 Newtonian behavior, a viscosity which is independent of shear rate, is shown by a straight line of unity slope.

It may be calculated from Figure 6.47 that at low rolling speeds it is experimentally impossible to obtain Newtonian conditions, as the low sliding speeds required -- less than 0.01 in/sec -- are well below the experimental range. This fact has important consequences when attempts are made to evaluate the effective viscosity from experimental data at lower rolling speed.

Two features of Figure 6.46, showing the variation of effective viscosity with rolling speed under isothermal and Newtonian conditions, merit special attention. The first of these is the great similarity in the shapes of the curves at rolling speeds above 50 in/sec, at all but the lowest pressure. This type of behavior has been observed experimentally, as seen, for example, in Figure 6.48 taken from Crook [4]. The second feature is the extremely rapid fall in the effective viscosity at low rolling speeds for pressures above 110,000 psi. Reliable measurement of the traction force is difficult at rolling speeds below 10 in/sec and high values of peak pressure, as the small film thickness becomes comparable with the dimensional irregularities of the disk surfaces, and full elastohydrodynamic conditions no longer exist. Extrapolation of results obtained at higher rolling speeds is
Figure 6.47. Traction coefficient versus sliding speed, at a peak Hertzian pressure of 176,000 psi, for different rolling speeds.
Figure 6.48.

The effective viscosity ($\eta_m$) as a function of rolling speed. (a) 30°C; (b) 45°C. —O— deduced from experimental results; ———, calculated from visco-elastic hypothesis. Load = $7.4 \times 10^7$ dyn cm$^{-1}$.

Curve from Crook [4].
therefore necessary if a value of effective viscosity at zero rolling speed is desired. Furthermore, the present analysis predicts that shear rate effects will be significant even at the lowest sliding speeds which can be reached experimentally. The measured values of the traction force will therefore be less than the values which would be obtained under Newtonian conditions. This effect, taken in conjunction with the rapid change in value of the effective viscosity at low rolling speeds, makes the extrapolation of experimental data to zero rolling speed subject to extremely large errors, the magnitude of the error increasing as the peak pressure is increased.

It is suggested therefore, that the observation of Johnson and Cameron, [11, Figure 15] whereby the same reduction in effective viscosity with rolling speed was observed at all pressures, is a consequence of the errors inherent in such an extrapolation. If this is so, it follows that the sharp change in the rate of increase of viscosity with pressure at pressures above 110,000 psi shown in Figure 6.49, from Johnson and Cameron's paper [11], is also a consequence of the errors in extrapolation, and is not a true property of the lubricant.

To explore this possibility in detail, hypothetical values of effective viscosity at zero rolling speed have been obtained by extrapolation of the curves of Figure 6.46, ignoring the calculated values at rolling speeds below 50 in/sec. The shear rate dependence of the lubricant viscosity is included by using values of effective viscosity calculated at a sliding speed of 0.02 in/sec instead of at the limit of zero sliding speed. The values so obtained are shown in Figure 6.50, plotted as a function of rolling speed over the range 50 to 500 in/sec.
Figure 6.49. Measured variation of effective viscosity versus peak Hertzian pressure, from Johnson and Cameron [11].
Figure 6.50. Effective viscosity at a sliding speed of 0.02 in/sec versus rolling speed for different values of peak Hertzian pressure. Below 50 in/sec, the curves have been extrapolated to zero rolling speed (dashed line).
The curves are then extrapolated below 50 in/sec to obtain a hypothetical value of effective viscosity at zero rolling speed. These values are plotted in Figure 6.51 to a base of peak pressure (dashed line), and show a large deviation from the true values of effective viscosity (solid line) at pressures above 120,000 psi. The experimentally determined values of effective viscosity shown in Figure 6.49 are also plotted in Figure 6.51. The close agreement between the hypothetical viscosity curve and the experimental values strongly supports the contention that the change in the slope of the viscosity-pressure curve at high pressures is an artifact arising from the difficulties inherent in the extrapolation procedure, and is not a physical property of the lubricant.

It has been found that by plotting the data of Figure 6.46 on a logarithmic, instead of a linear, scale of rolling speed, the separate curves for the different pressures can be combined into a single normalized curve. The effective viscosity values are normalized with respect to the value at the limit of zero rolling speed \( \eta_{U=0} \), and the rolling speed values are normalized with respect to \( U^* \), the rolling speed at which the effective viscosity is equal to 0.5\( \eta_{U=0} \). The resulting curve of \( \log(\eta/\eta_{U=0}) \) versus \( \log(U/U^*) \) is shown in Figure 6.52. The variation of \( U^* \) with the peak pressure in the contact is shown in Figure 6.53. These two graphs provide a quick and simple method of determining the variation of the effective viscosity with rolling speed for a given value of maximum pressure.

In this study of the role of compressional viscoelasticity in a rolling contact system, it has been necessary to simplify the analysis
Figure 6.51. Effective viscosity versus peak Hertzian pressure

- values at zero rolling speed, in the limit of zero sliding speed

- values obtained by extrapolation to zero rolling speed, from Figure 6.50

• measured values, from Johnson and Cameron [11]
Figure 6.52. Normalized plot of effective viscosity versus rolling speed.
Figure 6.53. Variation of $U^*$, the rolling speed at which the effective viscosity is reduced by a factor 2, versus peak Hertzian pressure.
as developed in Chapter III. Among these simplifications, the most important are the use of a viscoelastic model with only a single retardation time, and the assumption of a symmetrical viscosity distribution over the contact zone. It has also been necessary to estimate the viscosity $\eta_f$ and the bulk modulus $K_f$ as discussed in detail in section 3.3. The simplifications could be eliminated in a more detailed analysis. Such improvements are of little value, however, unless they are matched by improved information about the physical properties of the lubricant under the extreme conditions found in bearings and other heavily loaded contacts.

### 6.6 Comparison of Thermal Theories

A comparison of several thermal analyses is shown in Figure 6.54. These include the Johnson-Crook analysis and the author's numerical analysis for the Maxwell-limiting shear stress model and the hyperbolic liquid model ($c = .25$) corresponding to the B. E. L.-limiting shear stress model. Friction coefficients were also calculated neglecting the effects of compressional viscoelasticity. It is apparent that these effects must be considered to predict an accurate value of the maximum friction coefficient at the correct value of sliding speed.

### 6.7 Summary and Conclusions

1. The results of this study demonstrate that values of friction coefficient calculated according to the hyperbolic liquid model ($c = .25$) have a good correlation with those determined by experiment for the two lubricants, Mobil XRM 109 F4 and Shell Turbo 33. A similar correlation is obtained using $c = .20$ for Mobil XRM 177 F4.
Figure 6.54. Comparison of the experimental friction curve for Shell Turbo 33 at $P = 176,000$ psi, $U = 260$ in/sec and $T = 30$ °C and those calculated by several theoretical analyses.
This change in the value of c might possibly be explained by an increased film thickness due to the polymer additives in this fluid. This hypothesis is described in section 6.2.

2. The effects of shear rate and time are separated and explained by the two phenomena of shear viscoelasticity and compressional viscoelasticity, respectively.

3. A unified description of the non-Newtonian shear rate dependence of the viscosity is presented as a new hyperbolic liquid model. With this model, the transition from the non-linear region to the shear modulus dominated region is shown to be a smooth one. In the high-slip region, where the friction is dominated by the shear modulus, the variation of friction with load is very sensitive to the pressure dependence of the shear modulus.

4. The friction coefficient rises to a maximum value with increasing sliding speed and then decreases with any further increase in the sliding speed. The coefficient is also found to increase with increasing load and to decrease with increasing rolling speed and temperature.

5. The effects of compressional viscoelasticity are developed in terms of a simple model for the volume creep of a liquid following the application of a pressure step. This model is used to determine the dependence on rolling speed of the friction coefficient between highly loaded rolling contacts. Curves are presented which show the variation with rolling speed of the effective viscosity of the lubricant in the contact zone under isothermal conditions. Both the shape of the curves and the values of effective viscosity are consistent with the results of experimental measurements. The shape of the curves
in this region is found to be nearly independent of the peak pressure in the contact.

6. At very low values of rolling speed, in a region which is experimentally inaccessible, the analysis predicts a very rapid variation of effective viscosity with rolling speed. It is shown that, as a consequence, the extrapolation of experimental data to zero rolling speed can result in extremely large errors in the estimated values of effective viscosity.

The results of this study suggest future work that will increase the understanding of friction in elastohydrodynamic lubrication. The most urgently needed research is in the field of fluid rheology. The viscosity and density of lubricants at high pressures would be extremely helpful, and shear and compressional relaxation experiments must be performed to measure the fluid moduli at high pressures. This work is needed to confirm and expand our understanding of the mechanism of flow under EHD conditions.
APPENDIX A

NUMERICAL INTEGRATION OF A
NON-EQUIDISTANTLY TABULATED FUNCTION

A method of overlapping parabolas is employed to yield a second order approximation to the integral of a non-equidistantly tabulated function graphically represented in Figure A.1.

The function $f(x)$ may be represented by a second order Taylor's series expansion about $x_n$:

$$f(x) = f(x_n) + \frac{(x-x_n)}{1!} f'(x_n) + \frac{(x-x_n)^2}{2!} f''(x_n) + \cdots \quad (A.1)$$

Accordingly,

$$f_{n+1} = f_n + h_n f'_n + \frac{h_n^2}{2} f''_n + \cdots \quad (A.2)$$

and

$$f_{n-1} = f_n - h_{n-1} f'_n + \frac{h_{n-1}^2}{2} f''_n + \cdots \quad (A.3)$$

where

$$f_n + 1 = f(x_{n+1})$$

$$f_n - 1 = f(x_{n-1})$$

$$h_n = (x_n + 1 - x_n)$$

$$h_{n-1} = (x_n - x_{n-1})$$

Equations (A.2) and (A.3) are solved simultaneously to yield

$$f'' = \frac{2}{h_{n-1}(h_{n-1} + h_n)} f_{n-1} - \frac{2}{h_{n-1}h_n} f_n + \frac{2}{h_n(h_{n-1} + h_n)} f_{n+1} \quad (A.4)$$

and

$$f' = \frac{h_n}{h_{n-1}(h_{n-1} + h_n)} f_{n-1} + \frac{(h_n - h_{n-1})}{h_{n-1}h_n} f_n + \frac{h_{n-1}}{h_n(h_{n-1} + h_n)} f_{n+1} \quad (A.5)$$
Figure A.1. Graphical representation of the non-equidistantly tabulated function \( f(x) \).
The shaded area of Figure A.1 is calculated by integrating $f(x)$ between the limits $x_n$ and $x_{n+1}$. The function is approximated by a parabolic curve through $f_{n-1}$, $f_n$ and $f_{n+1}$. This integral is called $I_{n+1}^n$:

$$I_{n+1}^n = \int_{x_n}^{x_{n+1}} f(x) \, dx$$

$$= \int_{x_n}^{x_{n+1}} \left[ f_n + (x-x_n)f'_n + \frac{1}{2}(x-x_n)^2f''_n \right] dx$$

$$= h f_n + \frac{h^2}{2} f'_n + \frac{h^3}{6} f''_n \quad (A.6)$$

The derivatives are evaluated by equations (A.4) and (A.5), resulting in the following expression:

$$I_{n+1}^n = -\frac{h^3}{6h_{n-1}(h_{n-1} + h_n)} f_{n-1} + \frac{h_n(3h_{n-1} + h_n)}{6h_{n-1}} f_n$$

$$+ \frac{h_n(3h_{n-1} + 2h_n)}{6(h_{n-1} + h_n)} f_{n+1} \quad (A.7)$$

Similarly, the same integral may be calculated using a parabolic approximation for $f(x)$ through the points $f_{n-1}$, $f_{n+1}$, and $f_{n+2}$. Equation (A.1) determines

$$f_{n+2} = f_n + (h_{n+1} + h_n)f'_n + \frac{1}{2}(h_{n+1} + h_n)^2f''_n \quad (A.8)$$

This second integral, $I_{n+1}^n$, is defined by the Taylor's series expansion about $x_{n+1}$:
The derivatives \( f''_{n+1} \) and \( f'_{n+1} \) are evaluated in exactly the same manner as determined equations (A.4) and (A.5). These results are substituted into equation (A.9) yielding

\[
2I^{n+1}_n = \int_{x_n}^{x_{n+1}} \left[ f_{n+1} + (x-x_{n+1})f'_{n+1} + \frac{1}{2}(x-x_{n+1})^2f''_{n+1} \right] \, dx
\]

\[
= h_n f_{n+1} + \frac{h_n^2}{2} f'_{n+1} + \frac{h_n^3}{6} f''_{n+1} \tag{A.9}
\]

The best possible second order approximation of the integral is the average of the two values just calculated; this is the method of overlapping parabolas. Therefore, the average integral, \( I^{n+1}_n \), is defined as

\[
I^{n+1}_n = \frac{1}{2} \left[ I^{n+1}_{n-1} + 2I^{n+1}_n \right] \tag{A.11}
\]

where \( I^{n+1}_{n-1} \) and \( 2I^{n+1}_n \) are defined by equations (A.7) and (A.10), respectively.

The integral of the function \( f(x) \) between the limits \( x = a \) and \( x = b \) is defined by equation (A.12):

\[
\int_a^b f(x) \, dx \approx I^{a+1}_a + \sum_{n=a+1}^{b-2} I^{n+1}_n + 2I^{b}_b \tag{A.12}
\]

In order to keep the integral a function only of values in the range of integration, the method of overlapping parabolas is not used on the two extreme intervals.
APPENDIX B

NUMERICAL ANALYSIS

The FORTRAN IV coding of the friction analysis is listed in this appendix. It is followed by examples of data cards and the optional subroutines.
PROGRAM CONTROL(TAPE60=INPUT,OUTPUT,PUNCH) CONTRO

* PROGRAM CONTROL
* ------ -----
* CALCULATES THE FRICTION AND VELOCITY AND TEMPERATURE PROFILES IN AN ELASTOHYDRODYNAMIC LUBRICATED CONTACT.

* REQUIRED SUBPROGRAMS -
* SUBROUTINE PRINTS
* FUNCTION VISC
* FUNCTION ZERO
* FUNCTION PSI
* SUBROUTINE SECANT
* FUNCTION EXPI
* SUBROUTINE INTEG
* SUBROUTINE RTMI

* PROGRAM SET UP FOR MOBIL DATA NEGLECTING CONVECTION
COMMON C+GCLLLLL CONTRO
COMMON /CPS1/GINF*ETA0*T*Y*U2U1*NP*DUDY*IC,TG*OMEGA*CH*AH CONTRO
COMMON /CPS10/XK1*XKO CONTRO
COMMON /CQ/ Q CONTRO
COMMON /CZERO/ IPTRANS,TTRANS(6) CONTRO
COMMON /CPR/ ETA2 CONTRO
DIMENSION GINF(21),ETA0(21),T(21),Y(21),DUDY(21),TG(21),OMEGA(21) CONTRO
DIMENSION Q(21) CONTRO
DIMENSION CC(3,21) CONTRO
DIMENSION ETA(21) CONTRO
DIMENSION DTDY(21),NEWT(21),TRACTCF(20),SLIP(20) CONTRO
EXTERNAL PSI CONTRO
REAL LOGETA,NEWT CONTRO
INTEGER COUNT CONTRO

* PHYSICAL CONSTANTS AND DATA
* IRDP MAXIMUM HERTZIAN PRESSURE / 1000.
* IRDU MEAN ROLLING SPEED
* IRDT LUBRICANT INLET TEMPERATURE
* PHIT THERMAL REDUCTION FACTOR
* AH, CH HYPERBOLIC LIQUID MODEL PARAMETERS

9000 READ 9001,IRDP,IRDU,IRDT,PHIT,AH,CH CONTRO
9001 FORMAT(4X13,15,14*F6.3,10XF3,16XF4,2).
9002 IF(E0F(60))9999,9002 CONTRO
9003 CONTINUE CONTRO
9004 PUNCH 9003 CONTRO
9005 FORMAT(/
* PHZ=1000.*FLOAT(IRDP) CONTRO
* U=FLOAT(IRDU) CONTRO
* TOIL=FLOAT(IRDT)+460. CONTRO
* NP NUMBER OF GRID POINTS ACROSS THE HALF-FILM THICKNESS CONTRO
* NP=11 CONTRO
* NH = NP - 1 CONTRO
* NIP NUMBER OF PRESSURE STEPS IN HALF CONTACT-LENGTH CONTRO
* NIP=3 CONTRO

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TW=TOIL
TMAX=950.*
* COND THERMAL CONDUCTIVITY OF THE LUBRICANT
COND=1*778.*/3600.*
* CYLW CONTACT WIDTH OF THE DISKS
CYLW=0.25
* R1, R2 RADII OF THE DISKS
R1=R2=3.0
* E1, E2 ELASTIC MODULUS OF THE DISKS
E1=E2=3.0E+6
* POIS1, POIS2 POISSON'S RATIO FOR THE DISKS
POIS1=POIS2=0.3
* ALPHA VISCOSITY PRESSURE COEFFICIENT FOR THE LUBRICANT
ALPHA=1.04E-4
BETA=5.1E7*ALPHA
GAMMA=930.*ALPHA
* DK THERMAL CONDUCTIVITY OF THE DISKS
DK=21.7*778./3600.*
* DRHO DENSITY OF THE DISKS
DRHO=.253
* DC SPECIFIC HEAT OF THE DISKS
DC=109.778.*12.
* HERSA, HERSB CONSTANTS FOR THE HERSHEL VISCOSITY EQUATION
HERSA=8.974
HERSB=-3.2
* NGRAPH REQUIRED NUMBER OF GRAPHS FOR EACH TEMPERATURE PROFILE
NGRAPH=0
* MGRAPH REQUIRED NUMBER OF GRAPHS OF TRACTION COEF. VS SLIP
MGRAPH=1
* PRNT GIVES ADDITIONAL OUTPUT FOR DEBUGGING PURPOSES
PRNT=3HOFF
* INITIALIZATION AND BOUNDARY CONDITIONS
*
COUNT=0
IC = 1
DTDY(I)=0.0
Y(1)=0.0
TRACT=0.0
FLASH=0.0
PI=3.1415927
DO 10 I=1,NP
NEWT(I)=10.0*(I)*Y(I)+TW
10 CONTINUE
* R=R1*R2/(R1+R2)
E=2.*(1.+POIS1*POIS1)/E1+(1.+POIS2*POIS2)/E2
B=4.*R/E*PHZ
FLASHK=0.24/SORT(PH+PHZ*DK*DRHO*DC*U*R/E)
TRANS(1)=B/U/6.*
TRANS(2)=B/U/6.*
TRANS(3)=B/U/2.*
* CALCULATION OF LOAD
*
W=PHZ*PHZ*PI*(2./E)*R
PRINT 2*PHZ*W*R,E
2 FORMAT(*1CONTROL PHZ = *E15.8, W = *E15.8, R = *E15.8)
CALCULATION OF HALF-FILM THICKNESS

\[ \eta_{\text{ENT}} = 10.^{(0.66 + 0.6 \times (\eta_{\text{ENT}} \times \text{U})^{0.7} \times \text{E}^{0.03} \times \text{R}^{0.43} / \text{W}^{0.13})} \]

\[ \text{H} = 1.6 \times \alpha^{0.6} \times (\eta_{\text{ENT}} \times \text{U})^{0.7} \times \text{E}^{0.03} \times \text{R}^{0.43} / \text{W}^{0.13} \]

\[ \text{H} = \phi \times \text{H} \]

\[ \text{H} = 0.5 \times \text{H} \]

PRINT 10, \text{ALPHA}, \eta_{\text{ENT}}, \text{U}, \text{CH}, AH

FORMAT(\text{OH} = \#E12.5, \text{ALPHA} = \#E12.5, \eta_{\text{ENT}} = \#E12.5, \text{U} = \#F6.0, \text{CH} = \#F5.3, \text{AH} = \#F7.3)

SLIP LOOP

NSLIP=15

DATA (\text{SLIP}(\text{IU}), \text{IU}=1,20) / 1,2,3,4,5,6,7,8,9,10,11,12,13,14,15,16,17,18,19,20

DO 6000 \text{IU}=1,NSLIP

\[ \text{ فلاش} = \text{FLASHK} \times \text{TRACT} \times \text{SLIP} \]

PRINT 60, \text{FLASH}

FORMAT(* CONTROL FLASH = \#E15.8)

\[ \text{UZU} = 0.5 \times \text{SLIP} \]

PRINT 80, \text{IU}, \text{SLIP} \text{IU}

FORMAT(* CONTROL IU = \#I3, \text{SLIP} = \#E15.8)

HERZIAN PRESSURE LOOP

TRACT=0.0

DO 5999 \text{IP}=1,NIP

\[ \text{XB} = (2 \times \text{FLOAT} (+\text{IP}-1)/2 \times \text{FLOAT} (\text{NIP}) \]

\[ \text{P} = \text{PHZ} \times \text{SQR} \times (\text{XB} * (2 - \text{XB})) \]

SOLVE' MOMENTUM EQUATION

ITCOUNT=0

IF (\text{PRNT} = 2 \text{HON}) PRINT 44, I COUNT

FORMAT(* CONTROL I COUNT = \#I3)

COUNT=0

DO 11 I=1,NIP

IF (ITCOUNT.EQ.0.OR.\text{ITCOUNT}.GT.10) \text{T}(I)=\text{NEWT}(I)

IF (ITCOUNT.GT.0.AND.\text{ITCOUNT}.LE.10) \text{T}(I)=0.5*(\text{T}(I)+\text{NEWT}(I))

IF (ITCOUNT.GT.100) \text{GO TO} 6003

IF (T(I).GT.TMAX) T(I)=TMAX

\[ \text{TW} = \text{TOIL} + \text{FLASH} \]

\[ \text{GINF}(I) = 1.2 \times \text{F}^{0.52} \times 0.1333 \times (T(I)-492) = 1 \times 45 \]

IF (GINF(I).LT.1) GINF(I)=1

CALCULATION OF STEADY-STATE VISCOSITY

\[ \text{ETEX} = \text{ALPHA} \times \text{P} \times (\text{BETA} + \text{GAMMA} \times \text{P}) \times (680 - T(I)) / 680 \times T(I) \]

\[ \text{ETAO}(I) = \text{VISC}(P, \text{ETA}2(I), \text{IVCODE}) \]

11 CONTINUE

DUMMY=PSIO(0.0)

CALL SECANT(XK0=XK1,TAU=PSI1,XK,001,500,1TERR)

IF (ITERR.EQ.0) STOP
**Solve Energy Equation**

```fortran
!IF (PRNT.EQ.2HON) PRINT 41:TAU
  41 FORMAT(*:OTAU = *E15.8)

  * SOLVE ENERGY EQUATION
  *
  DO 5000 I=1,NP
  Q(I)=-TAU*DUDY(I)/COND*H
  5000 CONTINUE
  IF (PRNT.EQ.2HON) CALL PR1
  DO 5010 I=2,NP
  CALL INTEG0*.Y(I)*Y,Q,NP,DTY(I)*C*G*C,LLL,1,ERR)
  5010 IF (ERR.NE.0) PRINT 5012: IERR
  5012 FORMAT(* INTEGRATING DTY, IERR = #13)
  DO 5020 I=2,NP
  CALL INTEG0*.Y(I)*DTY(NP)*NEWT(I)*C*G*C,LLL,1,ERR)
  5020 IF (IERR.NE.0) PRINT 5022:IERR
  5022 FORMAT(* INTEGRATING DTY, IERR = #13)
  XK7=TW-NEWT(NP)
  ITCOUNT=ITCOUNT+1
  NEWT(I)=0.0
  CALL INTEG0*.Y(I)*NEWT(NP)
  DO 5030 I=1,NP
  NEWT(I)=NEWT(I)+XK7
  5030 IF (ABS(1.-(T(I)/NEWT(I))) LT.0.001) COUNT=COUNT+1
  IF (COUNT.NE.NP) GO TO 4
  IF (IVCODE.NE.0) PRINT 5034
  5034 FORMAT(* MINIMUM VISCOSITY REDUCTION*)
  CALL PR1
  PRINT 99
  PRINT 5040:((NEWT(I)*I),I=1,NP)
  5040 FORMAT(* NEW TEMP = *F7.2*, I = *14)

  * CALCULATION OF TRACTION
  *
  TRACT=TRACT+2.*B/MIP*TAU
  CONTINUE
  PRINT 6002:TRACTCF(IU)
  6002 FORMAT(* CONTROL TRACTCF = *E15.8)

  * PLOT TEMPERATURE PROFILE
  *
  IF (NGRAPH.EQ.0) GO TO 6000
  DO 5050 I=1,NGRAPH
  PRINT 5050:SLIP(IU)
  5050 FORMAT(* CONTROL TRACTCF = *E15.8)
  CALL STPLT1(1,NEWT,Y,NP,1,1YH)
  PRINT 5051
  5051 FORMAT(* CONTROL TRACTCF = *E15.8)

  * PLOT TRACTION COEFFICIENT
  *
  PUNCH 6005:((SLIP(IU),TRACTCF(IU)),I=1,NSLIP)
  6005 FORMAT(* CONTROL TRACTCF = *E15.8)
```

176
IF(MGRAPH.EQ.0) GO TO 6013
DO 6012 MGR=1,MGRAPH
PRINT 98
CALL STPLTI(1,SLIP,TRACTCF,NSLIP,14,14HTRACTION COEF.)
PRINT 6011
6011 FORMAT(1H0,100X,7HU2 - U1)
6012 CONTINUE
6013 CONTINUE
98 FORMAT(*)
99 FORMAT(*)
GO TO 9000
9999 CONTINUE
STOP
END
SUBROUTINE PRINTS
COMMON /CPSI/ GINF, ETAU, T, Y, U2, U1, NP, DUDY, IC, TG, OMEGA, CH, AH
COMMON /CO/ Q
COMMON /CPR/ ETA2
DIMENSION GINF(21), ETAU(21), T(21), Y(21), DUDY(21), TG(21), OMEGA(21)
DIMENSION Q(21)
DIMENSION ETA2(21)
ENTRY PR1
PRINT 1
1 FORMAT(*0 I Y(I) T(I) GINF(I) TG(I) ETA0(I) ETA2(I) OMEGA(I) DUDY(I) O(I)*)
PRINT 99
3 FORMAT(* I2.F7.2*F9.2*7(2XE11.4))
99 FORMAT(*0*)
RETURN
END
FUNCTION VISC(P,ETA2,CODE)

* * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * *

* FUNCTION VISC
* ----------------
* * CALCULATES THE TRANSIENT VALUE OF THE LUBRICANT VISCOSITY.
* * ARGUMENTS -
* * P PRESSURE
* * ETA2 EQUILIBRIUM VALUE OF VISCOSITY
* * CODE ERROR PARAMETER
* * REQUIRED SUBPROGRAMS -
* * FUNCTION ZERO
* * FUNCTION EXPI
* *
* COMMON STORAGE -
* * THE VARIABLE TAUP MUST BE IN LABELED COMMON CVISC.
* *
* ERROR INDICATIONS -
* * CODE = 0 INDICATES NO ERROR.
* * CODE = 1 INDICATES THE TRUE TRANSIENT VISCOSITY IS OUT OF THE
* * RANGE OF THE PROGRAM AND THE MAXIMUM POSSIBLE
* * VISCOSITY REDUCTION WAS ASSUMED.
* *
* COMMON /CVISC/ TAUP
EXTERNAL ZERO
REAL KF
INTEGER RTMIERR, CODE
PRNT=3HOFF

KF=3.5E5+9.*P
TAUP=50.*ETA2/KF

IF (PRNT.EQ.2HON) PRINT 81,KF,TAUP,P,ETA2

81 FORMAT(* VISC KF = *E13.6, TAUP = *E13.6, ETA2 = *E13.6)

* TEST OF RANGE
* IF (ZERO(-101)*ZERO(-201)*GT.01) GO TO 60
*
* ROUGH BOUNDING OF ZERO
*
* IF (ZERO(-51)*ZERO(-201)*GT.01) GO TO 30
* ROOT IS BETWEEN -201 AND -51
* XLI=-201
* XRI=-51
* GO TO 50
*
* ROOT IS BETWEEN -51 AND -01
* ICOUNT=0
* XLI=-51
* XRI=-31
* ZEROXRI=ZERO(XRI)
* ZEROXLI=ZERO(XLI)
* IF (ZEROXRI*ZEROXLI*LE.01) GO TO 50

179
IF (XRI.GT.-0.01) GO TO 60
XLI=XRI
XRI=XRI/3.
ZEROXLI=ZEROXRI
ZEROXRI=ZERO(XRI)
ICOUNT=ICOUNT+1
IF (ICOUNT.GT.100) GO TO 92
GO TO 40
*
* PRECISE DETERMINATION OF ZERO
*
50 CALL RTMI(SI,ZEROSI,ZERO,XLI,XRI,.05,100,RTMIERR)
IF (RTMIERR.NE.0) GO TO 90
VISC=ETA2*EXP(SI)
RETURN
*
* OUT OF RANGE
*
60 IF (ZERO(-20.).LT.0.) GO TO 70
*
* NEGLECT VISCOSITY REDUCTION OF LESS THAN 1. PER CENT
*
VISC=ETA2
RETURN
*
* MAXIMUM VISCOSITY REDUCTION
*
70 VISC=ETA2*EXP(-20.)
CODE=1
RETURN
*
* ERROR MESSAGES
*
90 PRINT 91,RTMIERR
91 FORMAT(* VISC ERROR IN RTMI ERR = *I2)
STOP
92 PRINT 93
93 FORMAT(* VISC MAX NO. OF ITERIONS EXCEEDED IN PRERTMI PROC*)
STOP
END
FUNCTION ZERO(SI)

* THIS EXTERNALLY SUPPLIED FUNCTION IS NEEDED BY SUBROUTINE RTMI
* CALLED FOR IN FUNCTION VISC. IT MUST BE PRESENT WHEN A TIME-
* DEPENDENT VISCOSITY FUNCTION IS USED.

COMMON /CZERO/ IPTRANS, TRANS(6)
COMMON /CVISC/ TAUP
INTEGER EXPIERR
PRNT=3H0FF
IF (SI.GE.-20.) GO TO 10
SI=-20.
PRINT 20
20 FORMAT(* SI HAS BEEN READJUSTED TO -20. *)
10 CONTINUE
CALL EXPI1(EIS1, SI, EXPIERR)
IF (EXPIERR.NE.0) GO TO 90
ZERO=TRANS(IPTRANS)+TAUP*EIS1-0.05*EXP(SI))
IF (PRNT.EQ.2H0N) PRINT 81, SI, EIS1, TRANS(IPTRANS)
81 FORMAT(* ZERO SI = *E13.6, EIS1 = *E13.6, TRANS = *E13.6)
RETURN
* ERROR MESSAGE
*
90 PRINT 91, EXPIERR
91 FORMAT(* ZERO ERROR IN EXPI ERR = *I3)
STOP
END
**FUNCTION PSI(TAU)**

*FUNCTION PSI*

-------------

* THIS FUNCTION SUBROUTINE SUPPLIES THE RHEOLOGICAL MODEL FOR THE LUBRICANT. IN THIS CASE, WE ARE USING THE HYPERBOLIC MODEL.*

* *

**COMMON C,GC,LLLL**

**COMMON /CPSI/ GINF,ETAO,T,Y,H,U2U1,NP,DUDY,IC,TG,OMEGA,CH,AH**

**COMMON /CPSIO/ XK1,XKO**

**DIMENSION GINF(21),ETAO(21),T(21),Y(21),DUDY(21),TG(21),OMEGA(21)**

**DIMENSION C(2,3,21),GC(?lt21)**

**REAL INTG**

**PRNT=3HOFF**

**100 CONTINUE**

**IF (PRNT.EQ.2HON) PRINT 200,TAU**

**200 FORMAT(*OTAU = *E15.8)**

**DO 210 I=1,NP**

**TG(I)=TAU/GINF(I)**

**IF(TG(I).LT.CH) GO TO 205**

**TAU=0.99999999*CH*GINF(I)**

**GO TO 100**

**205 OMEGA(I)=CH*TG(I)*(AH#CH*TG(I)+(TG(I)-CH)*(TG(I)-CH))/(CH-TG(I))**

**DUDY(I)=GINF(I)/ETAO(I)*OMEGA(I)*CH**

**210 CONTINUE**

**GO TO (220*230),1C**

**220 CALL INTEG (O.,*1.,*Y,DUDY,NP,INTG,C,GC,LLLL,IERR)**

**IC = 2**

**GO TO 240**

**230 CALL INTEG2(0.,*I.,*Y,DUDY,NP,INTG,C,GC,LLLL,IERR)**

**240 IF (IERR.NE.0) PRINT 241,IERR,TAU**

**241 FORMAT(* IERR = *I4,** AT TAU = *E15.8)**

**PSI=ALOG1C(INTG/U2U1)**

**IF (PRNT.EQ.2HON) PRINT 250,INTG,PSI**

**250 FORMAT(* INTG = *E15.8,** PSI = *E15.8)**

**TDUDY=0.0**

**DO 310 I=1,NP**

**TDUDY=TDUDY+DUDY(I)**

**ADJ=U2U1*FLOAT(NP)/TDUDY**

**IF (ADJ.GT.0.99) GO TO 390**

**DO 320 I=1,NP**

**320 DUDY(I)=ADJ*DUDY(I)**

**390 CONTINUE**

**ENTRY PSI0**

**XK1=0.30*CH*GINF(1)**

**XK0=0.5*XK1**

**PSI=0.0**

**RETURN**

**END**
SUBROUTINE SECANT(X0, X1, XF1NAL, FUNC, X, CONV, MAXIT, IERR)

* * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * *

* SUBROUTINE SECANT
* --------------
* SECANT METHOD SOLUTION OF FUNC(X) = 0
* *** ALLOWS POSITIVE X ONLY
*
* ARGUMENTS -
* X0, X1 TWO INITIAL GUESSES OF ROOT
* XF1NAL FINAL ESTIMATE OF ROOT
* FUNC EXTERNALLY SUPPLIED FUNCTION FUNC
* X PARAMETER OF FUNCTION FUNC
* CONV TEST FOR CONVERGENCE
* MAXIT MAXIMUM NUMBER OF ITERATIONS TO FIND SOLUTION
* IERR ERROR PARAMETER
*
* REQUIRED SUBPROGRAMS -
* FUNCTION FUNC(X)
* *
* COMMON STORAGE - NONE
*
* ERROR INDICATIONS -
* IERR = 0 INDICATES NO ERROR.
* IERR = 1 INDICATES THE MAX. NO. OF ITERATIONS WERE EXCEEDED
*
* EDWARD G. TRACHMAN
*
* * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * *

PRNT=3HOFF
IT = 0
IERR=0
XMIN=-1.E99
XMAX= 1.E99
FXMIN= -2.E99
FXMAX= 2.E99
FO=FUNC(X0)
IF (FO.LT.O.) XMIN=XO
IF (FO.LT.O.) FXMIN=FO
IF (FO.GT.O.) XMAX=XO
IF (FO.GT.O.) FXMAX=FO
Fl=FUNC(X1)
1 IF (Fl.LT.O..AND.X1.GT.XMIN) GO TO 50
51 IF (Fl.GT.O..AND.X1.LT.XMAX) GO TO 52
53 IF (PRNT.EQ.2HOFF) PRINT 2«XOtX1
2 FORMAT(* SECANT XO = *E15.8, XI = *E15.8)
   IF (ABS(F1).LT.CONV) GO TO 10
   IT = IT + 1
   IF (IT.GT.MAXIT) GO TO 99
   SLOPE=(F1-FO)/(X1-XO)
   CLOSE=0.01*ABS(FO)*ABS(F1))
   IF (PRNT.EQ.2HOFF) PRINT 80«X0«X1«FO«F1«SLOPE
80 FORMAT(*SECANT XO = *E15.8, XI = *E15.8, FO = *E15.8, F1 = *E15.8)
   I= *E15.8, SLOPE = *E15.8)
   USE XI
5 DELX=-F1/SLOPE

SECNT 1
SECNT 2
SECNT 3
SECNT 4
SECNT 5
SECNT 6
SECNT 7
SECNT 8
SECNT 9
SECNT 10
SECNT 11
SECNT 12
SECNT 13
SECNT 14
SECNT 15
SECNT 16
SECNT 17
SECNT 18
SECNT 19
SECNT 20
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SECNT 47
SECNT 48
SECNT 49
SECNT 50
SECNT 51
SECNT 52
SECNT 53
SECNT 54
SECNT 55
SECNT 56
SECNT 57
SECNT 58

183
X0=X1
F0=F1
GO TO 7
* USE X0
6 DELX=-F0/SLOPE
7 X1=X0+DELX
*** ALLOWS POSITIVE X ONLY
   IF (X1.LE.0.) X1=1.E-8
   IF (X1.GT.XMIN) GO TO 8
   X1=XMIN
   F1=FXMIN
   GO TO 1
8 IF (X1.LT.XMAX) GO TO 9
   X1=XMAX
   F1=FXMAX
   GO TO 1
9 F1=FUNC(X1)
GO TO 1
10 XFINAL=X1
   IF (PRINT.EQ.2HON) PRINT 11*XFINAL,F1,IT
11 FORMAT(*SECANT XFINAL = *E15.8*, F(XFINAL) = *E15.8*, IT = *I4*)
RETURN
50 XMIN=X1
   FXMIN=F1
   GO TO 51
52 XMAX=X1
   FXMAX=F1
   GO TO 53
99 PRINT 100
100 FORMAT(* SECANT MAXIMUM NUMBER OF ITERATIONS EXCEEDED*)
   IERP=1
   RETURN
END
SUBROUTINE EXPI (RES, X, IERR)

COMPUTES THE EXPONENTIAL INTEGRAL FOR NEGATIVE ARGUMENTS.
IN THE RANGE -20 TO ZERO.
FOR X EQUAL TO 0 THE RESULT VALUE IS SET TO 1.E75.
FOR X LESS THAN -20 OR GREATER THAN ZERO THE CALCULATION IS
BYPASSED AND THE ARGUMENT REMAINS UNCHANGED.
THE EXPONENTIAL INTEGRAL IS DEFINED AS THE
RES INTEGRAL EXP(-T)/T SUMMED OVER T FROM X TO INFINITY.
A POLYNOMIAL APPROXIMATION IS USED FOR ARGUMENTS IN THE
RANGE -5 TO ZERO.
REFERENCE: LUKE AND WIMP-JACOBI POLYNOMIAL EXPANSIONS OF A
GENERALIZED HYPERGEOMETRIC FUNCTION OVER A SEMI-INFINITE RANGE-
MATHEMATICAL TABLES AND OTHER AIDS TO COMPUTATION;
VOL. 17, 1963, ISSUE 84, PP. 395-404.
AN EXPONENTIAL APPROXIMATION IS USED FOR ARGUMENTS IN THE
RANGE -20 TO -5.
ARGUMENTS -
RES RESULT VALUE.
X ARGUMENT OF EXPONENTIAL INTEGRAL.
IERR RESULTANT ERROR PARAMETER.
REQUIRED SUBPROGRAMS - NONE
COMMON STORAGE - NONE
ERROR INDICATIONS -
IERR = 0 INDICATES NO ERROR.
IERR = 1 INDICATES X IS LESS THAN -20.
IERR = 2 INDICATES X IS POSITIVE.
EDWARD G. TRACHMAN M.E. DEPT. 492-5640
TEST OF RANGE
IERR=0
IF (X GT 0.) GO TO 60
IF (X GT -5.) GO TO 35
IF (X LT -20.) GO TO 55
ARGUMENT IS BETWEEN -20 AND -5.
C=-1.09414E-3/3.*X
RES=C*3.*X
RETURN
ARGUMENT IS BETWEEN -5 AND ZERO.

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* 35 X=-X
  IF (X) 40, 50, 40
  40 RES=-ALOG(ABS(X))-((((10317600E-11*X-15798675E-10)*X+
        1.16826592E-9)*X-.21915639E-8)*X+.27635830E-7)*X-.30726221E-6)*X+
        2.30996040E-5)*X-.28337590E-4)*X+.23148392E-3)*X-.0016666906)*X+
        3.0104166620)*X-.0555855520)*X+.25)*X-.57721566
  RES=-RES
  X=-X
  RETURN

* ARGUMENT IS EQUAL TO ZERO.

* 50 RES=1.E75
  RES=-RES
  X=-X
  RETURN

* ARGUMENT IS LESS THAN -20.

* 55 IERR=1
  RETURN

* ARGUMENT IS POSITIVE.

* 60 IERR=2
  RETURN
  END
SUBROUTINE INTEG(A,B,X,F,NP,VALUE,C,GC,L,IERR)

* INTEGRATES THE NON EQUIDISTANTLY TABULATED FUNCTION F(X(I))
* BETWEEN THE LIMITS A AND B, WHERE A OR B MUST EQUAL F(X(I)).
* A MODIFIED METHOD OF OVERLAPPING PARABOLAS IS EMPLOYED.
* ENTRY POINTS -
* INTEG FIRST TIME SUBROUTINE IS CALLED AND
* WHEN NEW ABSCISSAS ARE USED,
* INTEG1 WHEN NEW LIMITS OF INTEGRATION ARE USED,
* INTEG2 WHEN USING THE SAME ABSCISSAS AND
* LIMITS OF INTEGRATION AS THE LAST CALL.
* ARGUMENTS -
* A LOWER LIMIT OF INTEGRATION.
* B UPPER LIMIT OF INTEGRATION.
* X ARRAY OF ARGUMENT VALUES. MUST BE MONOTONICALLY
* INCREASING AND MUST BE DIMENSIONED NP.
* F ARRAY OF FUNCTION VALUES. MUST BE DIMENSIONED NP.
* NP NUMBER OF POINTS. NP MUST BE GREATER THAN 3.
* VALUE RESULTANT VALUE OF THE INTEGRATION.
* C, GC WEIGHTING FUNCTIONS PASSED TO THE MAIN PROGRAM
* FOR STORAGE.
* L LIMITS OF INTEGRATION PASSED TO THE MAIN PROGRAM
* FOR STORAGE.
* IERR RESULTANT ERROR PARAMETER.
* REQUIRED SUBPROGRAMS - NONE
* COMMON STORAGE -
* THE WEIGHTING FUNCTIONS C AND GC ARE STORED IN THE MAIN PROGRAM
* AND REQUIRE THE FOLLOWING DIMENSION STATEMENT WHERE DG*GE*NP.
* DIMENSION C(2,3,0),GC(0,0)
* ERROR INDICATIONS -
* IERR = 0 INDICATES NO ERROR.
* IERR = 1 INDICATES NP IS LESS THAN 4.
* IERR = 2 INDICATES THE LIMITS OF INTEGRATION ARE NOT AT NODES
* OR ARE OUT OF THE RANGE OF THE TABLE.

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M.E. DEPT. 492-5640

DIMENSION X(NP),F(NP),C(2,3,NP),GC(NP,NP)

NP MUST BE GREATER THAN 3
IF (NP.LE.3) GO TO 96

CALCULATION OF INTERVALS OF X

NH=NP-1
DO 10 I=1*NH
10 H(I)=X(I+1)-X(I)
DO 20 I=1*NH
IF (I.EQ.1) GO TO 15
*
DEFINE COEFFICIENTS OF FIRST PARABOLA
*
C(1*1,I)=-((H(I))**3/(6*H(I-1)*(H(I-1)+H(I))))
C(1*2,I)=H(I)**3*(3*H(I-1)+H(I))/((6*H(I-1))
C(1*3,I)=H(I)**3*(3*H(I-1)+2*H(I))/((6*(H(I-1)+H(I))))
15 CONTINUE
IF (I.EQ.NH) GO TO 20
*
DEFINE COEFFICIENTS OF SECOND PARABOLA
*
C(2*1,I)=H(I)**3*(2*H(I)+3*H(I+1))/((6*H(I)+H(I+1))
C(2*2,I)=H(I)**3*(H(I)+3*H(I+1))/((6*H(I+1))
C(2*3,I)=H(I)**3*(6*H(I+1)*(H(I)+H(I+1))))
20 CONTINUE
*
DEFINE GROUPED COEFFICIENTS
*
DO 61 L=1*NP
DO 61 1=1*NP
61 GC(I*L)=0.0
GC(I+1)=C(2*1*1)
GC(I+1)=C(2*2*1)
GC(I+3)=C(2*3*1)
NPM2=NP-2
DO 65 L=-2*NP
LP2=L+2
DO 65 1=1*LP2
CA=0.0
CB=0.0
IF (L-L+2.GT.0.AND.1-L+2.LT.4)
CA=C(I+1,L+2)
IF (L-L+4.GT.0.AND.1-L+4.LT.4)
CB=C(2*1*L+1)
65 GC(I,L)=GC(I,L-1)*0.5*(CA+CB)
NPM3=NP-3
DO 66 1=1*NPM3
66 GC(I*NP-1)=GC(I*NP-2)
GC(NP-2*NP-1)=GC(NP-2*NP-2)+C(I*NP-1)
GC(NP-1*NP-1)=GC(NP-1*NP-2)+C(I*NP-1)
GC(NP-1*NP-1)=GC(NP-1*NP-2)+C(I*NP-1)
*
ENTRY INTEGRATION
*
SETTING LIMITS OF INTEGRATION
*
IF (B-A) .GT.20.30
*
B IS GREATER THAN A
*
30 ALIM = A
BLIM = B
SIGN = 1.0
GO TO 50
*
A IS GREATER THAN B
*
ENTRY INTEG2

CALCULATION OF INTEGRAL OVER SUBINTERVAL

VALUE = 0.0

LP2=L+2
IF (LP2.GT.NP) LP2=NP
DO 80 1=1,LP2
80 VALUE=VALUE+GC(I*L)*F(I)

CALCULATE THE FINAL VALUE OF THE INTEGRAL

VALUE=SIGN*VALUE

SET ERROR PARAMETER FOR NORMAL RETURN
92 IERR = 0
RETURN

SET ERROR PARAMETER FOR TOO FEW POINTS
96 IERR = 1
RETURN

SET ERROR PARAMETER FOR A AND/OR B NOT AT NODES OR OUT OF RANGE OF THE TABLE
97 IERR = 2
RETURN
END
TYPICAL DATA CARDS

00000000011111111112222222233333333333444444444455555555555555555566666666666666666666666666677777777777
123456789012345678901234567890123456789012345678901234567890123456789012345678901234567890

16 115 500 175 .81 AH=0.0 CH=0.25 PROG (MOBIL) VISC 23J71
16 154 500 175 .81 AH=0.0 CH=0.25 PROG (MOBIL) VISC 23J71
16 200 500 175 .81 AH=0.0 CH=0.25 PROG (MOBIL) VISC 23J71
16 250 500 220 .89 AH=0.0 CH=0.25 PROG (MOBIL) VISC 23J71
16 115 1000 175 .58 AH=0.0 CH=0.25 PROG (MOBIL) VISC 23J71
16 154 1000 175 .58 AH=0.0 CH=0.25 PROG (MOBIL) VISC 23J71
16 200 1000 175 .58 AH=0.0 CH=0.25 PROG (MOBIL) VISC 23J71
16 200 1000 220 .72 AH=0.0 CH=0.25 PROG (MOBIL) VISC 23J71
16 115 1000 175 .58 AH=0.0 CH=0.25 PROG (MOBIL) VISC 23J71
16 154 1000 175 .81 AH=0.0 CH=0.20 PROG (MOBIL) VISC 23J71
16 200 1000 175 .81 AH=0.0 CH=0.20 PROG (MOBIL) VISC 23J71
16 250 1000 220 .89 AH=0.0 CH=0.20 PROG (MOBIL) VISC 23J71
16 115 1000 175 .58 AH=0.0 CH=0.20 PROG (MOBIL) VISC 23J71
16 154 1000 175 .58 AH=0.0 CH=0.20 PROG (MOBIL) VISC 23J71
16 250 1000 220 .72 AH=0.0 CH=0.20 PROG (MOBIL) VISC 23J71
PROGRAM CONTROL INPUT, TAPE60=INPUT, OUTPUT, PUNCH

* * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * *

PROGRAM CONTROL

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CALCULATES THE FRICTION AND VELOCITY AND TEMPERATURE PROFILES
IN AN ELASTOHYDRODYNAMIC LUBRICATED CONTACT. THIS VERSION OF
THE PROGRAM INCLUDES THE EFFECTS OF CONVECTIVE HEAT TRANSFER.

REQUIRED SUBPROGRAMS -
SUBROUTINE PRINTS
FUNCTION VISC
FUNCTION ZERO
FUNCTION PSI
SUBROUTINE SECANT
FUNCTION EXPI
SUBROUTINE INTEG
SUBROUTINE RTMI

* * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * *

PROGRAM SET UP FOR MOBIL XRM OIL WITH CONVECTION
COMMON C, GC, LLLL
COMMON /CPSI/ GINF, ETA0, T, Y, H, U2U1, NP, DUDY, IC, TG, OMEGA, CH, AH
COMMON /CPSIOX / XK1, XK0
COMMON /CO/ O
COMMON /CZERO/ IPTRANS, TRANS(6)
COMMON /CPR/ ETA2
DIMENSION GINF(21), ETA0(21), T(21), Y(21), DUDY(21), TG(21), OMEGA(21)
DIMENSION Q(21)
DIMENSION C(2, 3, 21), GC(21, 21)
DIMENSION ETA2(21)
DIMENSION DTDY(21), NEWT(21), TRACTCF(20), SLIP(20)
DIMENSION TJM1(21)
EXTERNAL PSI
REAL LOGETA, NEWT

PHYSICAL CONSTANTS AND DATA

9000 READ 9001, IRDP, IRDU, IRDT, PHIT, AH, CH
9001 FORMAT(4X, 15, I4, 4, F6.3, 10, F3.1, 6, XF4.2)
   IF (EOF(60)) 9999, 9002
9002 CONTINUE
   PUNCH 9003
9003 FORMAT(1)
   PHZ=1000.*FLOAT(IRDP)
   U=FLOAT(IRDU)
   TOIL=FLOAT(IRDT)+460.

* NP, NUMBER OF GRID POINTS ACROSS THE HALF-FILM THICKNESS
   NP=11
   NH = NP - 1
* NIP, NUMBER OF PRESSURE STEPS IN HALF CONTACT-LENGTH
   NIP=3
   NIP2=2*NIP
   Tw=TOIL

* COND, THERMAL CONDUCTIVITY OF THE LUBRICANT
   COND=1*778./3600.
* OILRHO, DENSITY OF THE LUBRICANT
* OILRHO=0.0325
  OILC SPECIFIC HEAT OF THE LUBRICANT
  OILC=4.778*12.
  CYLW=0.25
  CYLW CONTACT WIDTH OF THE DISKS
  R1, R2 RADII OF THE DISKS
  R1=R2=3.0
  E1, E2 ELASTIC MODULUS OF THE DISKS
  E1=E2=30*E+6
  POIS1, POIS2 POISSON'S RATIO FOR THE DISKS
  POIS1=POIS2=0.3
  ALPHA VISCOSITY PRESSURE COEFFICIENT FOR THE LUBRICANT
  ALPHA=1.04E-4
  BETA=5.1E7*ALPHA
  GAMMA=930*ALPHA
  DK THERMAL CONDUCTIVITY OF THE DISKS
  DK=217*778./3600.
  DRHO DENSITY OF THE DISKS
  DRHO=.283
  DC SPECIFIC HEAT OF THE DISKS
  DC=.109*778.*12.
  HERSA, HERSB CONSTANTS FOR THE HERSHEL VISCOSITY EQUATION
  HERSA=8.974
  HERSB=3.2
  NGRAPH REQUIRED NUMBER OF GRAPHS FOR EACH TEMPERATURE PROFILE
  NGRAPH=0
  MGRAPH REQUIRED NUMBER OF GRAPHS OF TRACTION COEF. VS SLIP
  MGRAPH=1
  PRNT=2HON GIVES ADDITIONAL OUTPUT FOR DEBUGGING PURPOSES
  PRNT=3HOFF

* INITIALIZATION AND BOUNDARY CONDITIONS

IC = 1
DTDY(1)=0.0
Y(1)=0.0
TRACT=0.0
FLASH=0.0
PI=3.1415927
DO 10 I=1*NH
  NEWT(I)=I**2*(1.-Y(I))+TW
  IF (Y(I)+1==Y(I)+1./FLOAT(NH)
10 CONTINUE

R=2*(R1*R2/(R1+R2))
E=2*(1.-POIS1*POIS1)/E1+(1.-POIS2*POIS2)/E2
B=4.*R/E*PHZ
FLASH=0.24/SQRT(P1*PHZ*DK*DRHO*DC*R/E)
TRANS(1)=B/Y/U/6.
TRANS(2)=B/Y/U/6.
TRANS(3)=TRANS(3)
TRANS(5)=TRANS(2)
TRANS(6)=TRANS(1)
DELXB=B/FLOAT(NH)

* CALCULATION OF LOAD

192
CALCULATION OF HALF-FILM THICKNESS

\[ \eta_{\text{ENT}} = \text{io.}^{(\text{HERSA} + \text{HERSB} \cdot \log_{10}(T_{\text{OIL}} - 460))} \cdot 1.45 \cdot 10^{-7} \]

\[ H = 1.6 \cdot \alpha^{0.6} \cdot \eta_{\text{ENT}}^{0.7} \cdot E^{0.03} \cdot R^{0.43} \cdot W^{0.13} \]

\[ H = 1.2 \cdot H \]

\[ H = \varphi_{\text{T}} \cdot H \]

\[ H = 0.5 \cdot H \]

FORMAT (*1 CONTROL \( \eta = *E15.8 \), \( \alpha = *E15.8 \), \( \eta_{\text{ENT}} = *E15.8 \)

FORMAT (*1 CONTROL \( H = *E15.8 \), \( \alpha = *E15.8 \), \( \eta_{\text{ENT}} = *E15.8 \),

SLIP LOOP

\[ \text{NSLIP} = 15 \]

DATA (SLIP(IU), IU=1,20) /5,1,2,3,4,5,6,7,8,10,15,20,30,40/ 

1,50,60,70,80,90/ 

DO 6000 IU=1,NSLIP 

DATA (TRACTCF(IP), IP=1,20) /20*0.0/ 

FLASH=FLASHK*TRACT*SLIP(IU) 

IF (FLASH.LT.0.5) FLASH=0.0 

TW=T_{\text{OIL}}+FLASH 

PRINT 6,FLASH 

U2U1=0.5*SLIP(IU) 

PRINT 8,IU,SLIP(IU) 

HERTZIAN PRESSURE LOOP

\[ \text{TRACT}=0.0 \]

DO 5999 IP=1,NIP2 

IPTRANS=IP 

XB=(2*FLOAT(IP)-1.)/2.*FLOAT(NIP) 

P=PHZ*SQRT(XB*(2.-XB)) 

DO 3 I=1,NP 

IF (IP.EQ.1) \( T_{\text{JM1}}(1) = T_{\text{OIL}} \)

IF (IP.GT.1) \( T_{\text{JM1}}(I) = \text{NEWT}(I) \)

3 CONTINUE

SOLVE MOMENTUM EQUATION

\[ \text{ITCOUNT}=0 \]

\[ \text{IT}=0 \]

\[ \text{ITC}=1 \]

\[ T_{\text{MAX}}=950. \]

\[ T(1)=951. \]

4 IF (PRINT.EQ.2) PRINT 44,IT 

44 FORMAT (*1 CONTROL IT=E15.8) 

IF (IT.EQ.0) GO TO 4970 

\[ \text{IT}=\text{IT}+1 \]

IF (IT.GT.1) \( T_{\text{MIN}}=T(\text{IT}-1) \)

\[ T_{\text{MIN}}=T_{\text{OIL}} \]

\[ \text{ITC}=0 \]

4970 DO 11 I=1,NP
IF (ITCOUNT .EQ. 0) TEMT = NEWT(I)
IF (ITCOUNT .GT. 0) TEMT = 0.5 * (T(I) + NEWT(I))
IF (ITCOUNT .GT. 200) GOTO 4982
IF (IP .LE. NIP .AND. TEMT .LT. TJM1(I)) 4980 4979
4979 IF (IP .GT. NIP .AND. TEMT .LT. TJM1(I)) 4982 4985
4980 TEMT = 0.5 * (TJM1(I) + T(I))
4985 IF (I .GT. ITP) GOTO 4987
* UPDATE TMAX AND TMIN
IF (TEMT .LT. T(I)) .AND. TMAX .LT. T(I) .AND. TMIN .GT. T(I) TMAX = T(I)
IF (TEMT .GT. T(I)) .AND. TMIN .LT. T(I) .AND. TMAX .GT. T(I) TMIN = T(I)
IF (PRTNT .EQ. 2) PRINT 4988 TMAX TMIN TEMT
4988 FORMAT (494 494 494)
* BOUND TEMPERATURE
IF (TEMT .GT. TMAX) OR (TEMT .LT. TMIN) TEMT = 0.5 * (TMAX + TMIN)
IF (PRTNT .EQ. 2) PRINT 4989 TEMT T(I)
4989 FORMAT (494 494)
* TEST FOR CONVERGENCE
IF (ABS(TEM-T(I)) .GT. 3) 4986 4987
4986 ITC = 1
4987 T(I) = TEMT
4990 CONTINUE
GINF(I) = 1.2 * P/(2.52 + 0.1333 * (T(I) - 492.)) - 1.45E4
IF (GINF(I) .LT. 1.) GINF(I) = 1.
* CALCULATION OF STEADY-STATE VISCOSITY
ETEXP = ALPHA * P + (BETA + GAMMA * P) * (680 - T(I)) / 680 * T(I)
ETA2(I) = 62 * EXP(ETEXP) * 1.45E-5
* CALCULATION OF TIME-DEPENDENT VISCOSITY
ETAO(I) = VISC(P, ETA2(I) * IVCODE)
11 CONTINUE
DUMMY = PSI0(0, 0)
CALL SECANT(XK0, XK1, TAU, PSI1, XK1, 001, 500, ITERM)
IF (ITEMR .EQ. 1) STOP
IF (PRTNT .EQ. 2) PRINT 41 TAU
41 FORMAT (415.8)
* SOLVE ENERGY EQUATION
* DO 5000 I = 1, NP
Q(I) = TAU * DDUDY(I) / COND * H
QC = 0.1RHO*OIRC/COND*H*U*(TCI - TJM1(I))/DELXB
Q(I) = Q(I) + QC
5000 CONTINUE
IF (PRTNT .EQ. 2) CALL PRI
DO 5010 I = 2, NP
CALL INTEG1(0, Y(I), Y, DTYD(I), NP, DTDY(I), GC, LLLL, IERROR)
5010 CONTINUE
IF (PRTNT .EQ. 2) CALL PRI
5012 FORMAT (* INTEGRATING O, IERROR = *13)
DO 5020 I = 2, NP
CALL INTEG1(0, Y(I), Y, DTYD(I), NP, NEWT(I), GC, LLLL, IERROR)
5020 IF (ITEMR .EQ. 0) PRINT 5022, IERROR
5022 FORMAT (* INTEGRATING DTDY, IERROR = *13)
XK7 = TW - NEWT(NP)
ITCOUNT = ITC + 1
NEWT(I) = 0
DO 5030 I = 1, NP
5030 NEWT(I) = NEWT(I) + XK7
194
GO TO 4

5032 DO 5033 I=1,NP
5033 NEWT(I)=T(I)
   IF (IVCODE*NE*0) PRINT 5034
5034 FORMAT(*0 MINIMUM VISCOSITY REDUCTION*)
   CALL PRI
   PRINT 99
   PRINT 5040:((NEWT(I)*I=1,NP)
5040 FORMAT(* NEW TEMP = *F7.2* I = #14)
   *
   *   CALCULATION OF TRACTION
   *
   TRACT=TRACT+8/FLOAT(NIP)*TAU
5999 CONTINUE
   TRACTCF(IU)=TRACT/W
   PRINT 6002,TRACTCF(IU)
6002 FORMAT(*CONTROL TRACTCF(IU) = *E15.8)
   *
   *   PLOT TEMPERATURE PROFILE
   *
   IF (NGRAPH*EQ*0) GO TO 6000
   DO 5052 NGR=1,NGRAPH
   PRINT 5050,SLIP(IU)
5050 FORMAT(*1(U2 - U1) = *E15.8)
   PRINT 99
   CALL STPLT1(1,NEWT,Y,NP,1H*111H) .CONTINUE
   PRINT 5051
5051 FORMAT(1HC92X*11HTEMPERATURE)
5052 CONTINUE
6000 CONTINUE
6003 PRINT 98
6003 PRINT 6001,(TRACTCF(IU),SLIP(IU)) IU=1,NSLIP
6001 FORMAT(* CONTROL TRACT = *E15.8* U2-U1 = *E15.8)
   PUNCH 6005,(SLIP(IU),TRACTCF(IU)) IU=1,NSLIP
6005 FORMAT(2F10.5)
   *
   *   PLOT TRACTION COEFFICIENT
   *
   IF (MGRAPH*EQ*0) GO TO 6013
   DO 6012 MGR=1,NGRAPH
   PRINT 98
   CALL STPLT1(1,SLIP,TRACTCF,NSLIP,1H*14,14HTRACTION COEF*)
6011 FORMAT(1HC100X,7HU2 - U1)
6012 CONTINUE
6013 CONTINUE
99 FORMAT(*1*)
99 FORMAT(*0*)
   GO TO 9000
9999 CONTINUE
STOP
END.
FUNCTION PSI(TAU)

* THIS FUNCTION SUBROUTINE SUPPLIES THE RHEOLOGICAL MODEL
* FOR THE LUBRICANT. IN THIS CASE, WE ARE USING THE
* BARLOW AND LAMB VISCOELASTIC MODEL WITH A LIMITING SHEAR STRESS.

FUNCTION PSI

--------

COMMON CG, GC, LLLL
COMMON /CPSI/ GINF, ETA, Y, T, U2U1, NP, DUDY, IC, TG, OMEGA, CH, AH
COMMON /CPSIO/ XK1, XK0
DIMENSION GINF(21), ETA(21), T(21), Y(21), DUDY(21), TG(21), OMEGA(21)
DIMENSION C(2, 3, 2), GC(21, 21)
REAL INTG
PRNT = 3HOFF
CONTINUE

IF (PRNT EQ 2H0) PRINT 200, TAU

FORMAT (*OTAU = *E15.8)
DO 210 I = 1, NP
TG(I) = TAU / GINF(I)
IF (TG(I) LT 0.25) GO TO 205
SLIP REGION
OMEGA(I) = (TG(I) - 0.2499963) * 1E6
GO TO 208
VISCOELASTIC REGION
OMEGA(I) = 55.2 * TG(I) * TG(I) + TG(I)
DUDY(I) = GINF(I) / ETAO(I) * OMEGA(I) * H
CONTINUE
GO TO (220, 230) I

CALL INTEG (0, 1, Y, DUDY, NP, INTG, C, GC, LLLL, I ERR)

IC = 2
GO TO 240

CALL INTEG2 (0, 1, Y, DUDY, NP, INTG, C, GC, LLLL, I ERR)

IF (I ERR NE 0) PRINT 241, I ERR, TAU

FORMAT (* I ERR = *I4** AT TAU = *E15.8)
PSI = ALOG10(INTG / U2U1)
IF (PRNT EQ 2H0) PRINT 250, INTG, PSI

FORMAT (* INTG = *E15.8** PSI = *E15.8)
TDUDY = 0.0
DO 310 I = 1, NP
ADJ = U2U1 / FLOAT(NP) / TDUDY
IF (ADJ GT 0.999) GO TO 390
DO 320 I = 1, NP

DUDY(I) = ADJ * DUDY(I)
CONTINUE
RETURN
ENTRY PSI

CHECK OMEGA = 3.7
XK1 = TAU = 0.25 * GINF(I)
DO 410 I = 1, NP
TG(I) = TAU / GINF(I)
OMEGA(I) = 55.2 * TG(I) * TG(I) + TG(I)
DUDY(I) = GINF(I) / ETAO(I) * OMEGA(I) * H
CALL INTEG (0, 1, Y, DUDY, NP, INTG, C, GC, LLLL, I ERR)

196
IF (IERR.NE.0) PRINT 241:IERR:TAU
IF (ALOG10(INTG/U2U1)) 20:20:30

* SLIP MODEL
20 XK0=10:*XK1
PSI=0.0
RETURN

* VISCOELASTIC MODEL
30 XK0=0.1:*XK1
PSI=0.0
RETURN

END
FUNCTION PSI(TAU)

* * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * *

FUNCTION PSI

---------- ---

* THIS FUNCTION SUBROUTINE SUPPLIES THE RHEOLOGICAL MODEL
* FOR THE LUBRICANT. IN THIS CASE WE ARE USING THE
* MAXWELL VISCOELASTIC MODEL WITH A LIMITING SHEAR STRESS.

* * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * *

COMMON C*GC*LLL
COMMON /CPSI/ GINF*ETA0*T*Y*H*U2U1*NP*DUDY*IC*OMEGA*CH
COMMON /CPSI OX XK1*XKO
DIMENSION GINF(21),ETA0(21),T(21),Y(21),DUDY(21),TG(21),OMEGA(21)
DIMENSION C(2,3,21),GC(21,21)
REAL INTG
PRNT=3HOFF
CONTINUE

; IF (PRNT=EQ#2HON) PRINT 200,TAU
FORMAT(*OTAU = *E15.8)
DO 210 1=1,NP
TG(I)=TAU/GINF(I)
IF (TG(I) LT 0.50) GO TO 205
SLIP REGION
OMEGA(I)=(TG(I)-0.499999)*1.E6
GO TO 208
VISCOELASTIC REGION
GINFK=0.5/TG(I)
OMEGA(I)=1./(GINFK+SQRT(GINFK*GINFK-1.))
DUDY(I)=GINF(I)/ETAO(I)*OMEGA(I)*H
CONTINUE
GO TO (220,230),1C
CALL INTEG (0.1.,Y,DUDY,NP,INTG,C*GC*LLL,IER)
1C=2
GO TO 240
CALL INTEG2(Y,DUDY,NP,INTG,C*GC*LLL,IER)
IF (IERR NE 0) PRINT 241,IERR,TAU
FORMAT(* IERR = *I4 AT TAU = *E15.8)
PSI=ALOG10(INTG/-U2U1)
IF (PRNT.EQ#2HON) PRINT 250,INTG,PSI
FORMAT(* INTG = *E15.8,* PSI = *E15.8)
TDDUDY=0.0
DO 310 I=1,NP
ADJ=U2U1*FLOAT(NP)/TDDUDY
IF (ADJ*GT*0.99) GO TO 390
DO 320 I=1,NP
TGG(I)=TAU/GINF(I)
GINFK=0.5/TG(I)
OMEGA(I)=1./(GINFK+SQRT(GINFK*GINFK-1.))

RETURN
ENTRY PSI0

* CHECK OMEGA =1
XK1=TAU*0.50*GINF(I)
DO 410 I=1,NP
TG(I)=TAU/GINF(I)
GINFK=0.5/TG(I)
OMEGA(I)=1./(GINFK+SQRT(GINFK*GINFK-1.))
410 DUDY(I) = GINF(I)/ETA0(I)*OMEGA(I)*H
        CALL INTEG (0.0, Y*Y, DUDY, NP, INTG, GC*GC, LLLL, IERR)
        IF (IERR .NE. 0) PRINT 241, IERR, TAU
        IF (ALOG10(INTG/U2U1)) 20, 20, 30

* SLIP MODEL
20 XK0 = 10.0*XXK1
   PSI = 0.0
   RETURN

* VISCOELASTIC MODEL
30 XK0 = 0.1*XXK1
   PSI = 0.0
   RETURN
END
SUBROUTINE RTMI(X, F, FCT, XLI, XRI, EPS, IEND, IER)

SUBROUTINE RTMI

PURPOSE
TO SOLVE GENERAL NONLINEAR EQUATIONS OF THE FORM FCT(X)=0
BY MEANS OF MULLER-S ITERATION METHOD.

USAGE
CALL RTMI (X,F,FCT,XLI,XRI,EPS,IEND,IER)
PARAMETER FCT REQUIRES AN EXTERNAL STATEMENT.

DESCRIPTION OF PARAMETERS
X - RESULTANT ROOT OF EQUATION FCT(X)=0.
F - RESULTANT FUNCTION VALUE AT ROOT X.
FCT - NAME OF THE EXTERNAL FUNCTION SUBPROGRAM USED.
XLI - INPUT VALUE WHICH SPECIFIES THE INITIAL LEFT BOUND
OF THE ROOT X.
XRI - INPUT VALUE WHICH SPECIFIES THE INITIAL RIGHT BOUND
OF THE ROOT X.
EPS - INPUT VALUE WHICH SPECIFIES THE UPPER BOUND OF THE
ERROR OF RESULT X.
IEND - MAXIMUM NUMBER OF ITERATION STEPS SPECIFIED.
IER - RESULTANT ERROR PARAMETER CODED AS FOLLOWS
IER=0 - NO ERROR,
IER=1 - NO CONVERGENCE AFTER IEND ITERATION STEPS
FOLLOWED BY IEND SUCCESSIVE STEPS OF
BISECTION,
IER=2 - BASIC ASSUMPTION FCT(XLI)*FCT(XRI) LESS
THAN OR EQUAL TO ZERO IS NOT SATISFIED.

REMARKS
THE PROCEDURE ASSUMES THAT FUNCTION VALUES AT INITIAL
BOUNDS XLI AND XRI HAVE NOT THE SAME SIGN. IF THIS BASIC
ASSUMPTION IS NOT SATISFIED BY INPUT VALUES XLI AND XRI, THE
PROCEDURE IS BYPASSED AND GIVES THE ERROR MESSAGE IER=2.

SUBROUTINES AND FUNCTION SUBPROGRAMS REQUIRED
THE EXTERNAL FUNCTION SUBPROGRAM FCT(X) MUST BE FURNISHED
BY THE USER.

METHOD
SOLUTION OF EQUATION FCT(X)=0 IS DONE BY MEANS OF MULLER-S
ITERATION METHOD OF SUCCESSIVE BISECTIONS AND INVERSE
PARABOLIC INTERPOLATION, WHICH STARTS AT THE INITIAL BOUNDS
XLI AND XRI. CONVERGENCE IS QUADRATIC IF THE DERIVATIVE OF
FCT(X) AT ROOT X IS NOT EQUAL TO ZERO. ONE ITERATION STEP
REQUIRES TWO EVALUATIONS OF FCT(X). FOR TEST ON SATISFACTORY
ACCURACY SEE FORMULAE (34) OF MATHEMATICAL DESCRIPTION.
FOR REFERENCE, SEE G. K. KRISTIANSEN, ZERO OF ARBITRARY
FUNCTION, BIT, VOL. 3 (1963), PP. 205-206.
C C PREPARE ITERATION
C
IER=0
XL=XLI
XR=XRI
X=XL
TOL=X
F=FCT(TOL)
IF(F)I,16+1
1 FL=F
X=XR
TOL=X
F=FCT(TOL)
IF(F)1,16+2
2 FR=F
IF(SIGN(1*FL)+SIGN(1*FR))25,3,25
C C BASIC ASSUMPTION FL*FR LESS THAN 0 IS SATISFIED.
C C GENERATE TOLERANCE FOR FUNCTION VALUES.
3 I=0
TOLF=100*EPS
C C START ITERATION LOOP
4 I=I+1
C C START BISECTION LOOP
DO '13 K=1*END
X=.5*(XL+XR>
TOL=X
F=FCT(TOL)
IF(F)5,16+5
5 IF(SIGN(1*F)+SIGN(1*FR))7,6,7
C C INTERCHANGE XL AND XR IN ORDER TO GET THE SAME SIGN IN F AND FR
6 TOL=XL
XL=XR
XR=TOL
TOL=FL
FL=FR
FR=TOL
7 TOL=F-FL
A=F*TOL
A=A+A
IF(A-FR*(FR-FL))8,9,9
8 IF(I=IEND)17,17,9
9 XR=X
FR=F
C C TEST ON SATISFACTORY ACCURACY IN BISECTION LOOP
TOL=EPS
A=ABS(XR)
IF(A-1)*11,11,10
10 TOL=TOL*A
11 IF(ABS(XR-XL)=TOL)12,12+13
12 IF(ABS(FR-FL)=TOLF)14,14+13
13 CONTINUE
C END OF BISECTION LOOP
C
C NO CONVERGENCE AFTER IEND ITERATION STEPS FOLLOWED BY IEND
C SUCCESSIVE STEPS OF BISECTION OR STEADILY INCREASING FUNCTION
C VALUES AT RIGHT BOUNDS. ERROR RETURN.
IERR=1
14 IF (ABS(FR)-ABS(FL)) > TOL 16, 16, 15
15 X=XL
  F=FL
16 RETURN
C
C COMPUTATION OF ITERATED X-VALUE BY INVERSE PARABOLIC INTERPOLATION
17 A=FR-F
  DX=(X-XL)*FL*(1.+F*(A-TOL)/(A*(FR-FL)))/TOL
  XM=X
  FM=F
  X=XL-DX
  TOL=X
  F=FCT(TOL)
  IF(F) 18*16*18
C
C TEST ON SATISFACTORY ACCURACY IN ITERATION LOOP
18 TOL=EPS
  A=ABS(X)
  IF (A-1) > TOL 20, 20, 19
19 TOL=TOL*A
20 IF (ABS(DX)-TOL) > TOL 21, 21, 22
21 IF (ABS(F)-TOL) 16, 16, 22
C
C PREPARATION OF NEXT BISECTION LOOP
22 IF (SIGN(1*F)+SIGN(1*FL)) 24, 23, 24
23 XR=X
  FR=F
  GO TO 4
24 XL=X
  FL=F
  XR=XM
  FR=FM
  GO TO 4
C END OF ITERATION LOOP
C
C ERROR RETURN IN CASE OF WRONG INPUT DATA
25 IERR=2
  RETURN
END
APPENDIX C

NOMENCLATURE

\( a \) rise parameter of the hyperbolic model

\( b = 4p R/E \), half-width of Hertzian contact

\( c \) limiting shear stress/limiting shear modulus ratio

\( c \) specific heat of the lubricant

\( c_m \) specific heat of the disk

\( D \) shear rate

\( E = \frac{1}{2}[1-(\nu_1^2/E_1)+[(1-\nu_2^2/E_2)]], \) effective modulus of elasticity of the disks

\( E_1, E_2 \) elastic moduli of the two disks

\( Ei(x) \) exponential integral

\( f \) fractional free volume

\( f_2 \) equilibrium free volume

\( \Gamma \) complex shear modulus

\( G \) high frequency limiting shear modulus

\( G = G_e / K \)

\( h \) half-parallel lubricant film thickness

\( h_0 \) minimum lubricant film thickness

\( k \) thermal conductivity of the lubricant

\( k_m \) thermal conductivity of the disk

\( K \) Oldroyd-Dyson parameter

\( K \) bulk modulus

\( K_o \) low frequency bulk modulus

\( K_\infty \) high frequency bulk modulus

\( K_r \) bulk modulus associated with molecular rearrangements in free volume

\( K_r \) complex relaxational modulus
$K_2$  high frequency value of $K_r$

$K_t$  torsional spring constant

$J$  inertia

$J^*$  complex compliance

$p$  pressure

$p_{\text{Hz}}$  maximum Hertzian pressure

$P$  maximum Hertzian pressure (on graphs)

$P_p$  pressure step

$P$  normal stress

$R = R_1 R_2/(R_1 + R_2)$, effective radius of the disk pair

$R_1, R_2$  radii of the disks

$s = \ln(\eta_2/\eta)$

$s_1 = \ln(\eta_2/\eta_1)$

$t$  time

$T$  temperature

$T_l$  lubricant inlet temperature (on graphs)

$T_b$  bulk temperature of the disk

$T_{\text{ent}}$  lubricant entrance temperature

$T_s$  mean surface temperature of the disk in the contact zone, "flash temperature"

$T_o$  reference temperature at which there is no free volume

$u, v$  velocity components in the fluid film

$U$  mean rolling speed

$U_1, U_2$  surface speeds of the disks

$U^*$  reference rolling speed at which $\bar{\eta} = \frac{1}{2} \bar{\eta}_{U=0}$

$v$  specific volume

$v_f$  free volume
v_i specific volume after elastic deformation only
v_o occupied volume
v_1 initial specific volume
v_2 equilibrium specific volume
w load per unit length of cylinder
x,y,z Cartesian coordinates
Z* complex shear mechanical impedance
\alpha viscosity-pressure coefficient
\gamma shear strain
\dot{\gamma} shear rate
\eta shear viscosity
\eta_{ent} viscosity of the lubricant at entrance conditions
\eta_f free volume viscosity
\eta_v volume viscosity
\eta_1 initial viscosity
\eta_2 equilibrium viscosity
\eta_i effective viscosity
\lambda Maxwell relaxation time
\lambda_f retardation time
\lambda_v volume relaxation time
\mu coefficient of friction
\mu* complex fluidity
\nu_{1,2} Poisson's ratio for the two disks
\xi slide/roll ratio
\rho density of the lubricant
\rho_m density of the disk
\tau shear stress

205
\( \dot{\theta}_T \)  \hspace{1cm} \text{film thickness thermal reduction factor}

\( \omega \)  \hspace{1cm} \text{angular frequency}

\( \Omega \)  \hspace{1cm} \text{non-dimensional shear rate}
REFERENCES


"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

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