**Wen, Shizhu** National Tribology Laboratory, Tsinghua University, Beijing, China

# On Thin Film Lubrication

**Abstract**

*Various phenomena are revealed under EHL and micro-EHL conditions, such as the properties of the lubricant under high pressure, traction, and the load-bearing capacity of the lubricant film, and are discussed in the present paper. A new lubrication regime, thin film lubrication, has been discussed. The theoretical and practical significance of research on thin film lubrication is elaborated. Finally, the characteristics describing thin film lubrication and its main research directions are suggested.*

**Keywords**

thin film lubrication, EHL, roughness, asperities, boundary, micro-EHL, research

## INTRODUCTION

The elastohydrodynamic lubrication (EHL) theory, developed since the 1960s, deals with lubrication in concentrated contacts by combining classical hydrodynamic lubrication theory with elastic contact theory. With the development of the EHL theory, lubrication research has become a multi-disciplinary subject. A close relationship between lubrication and wear, the two main subjects in tribology, has been created.

Since EHL films are present under severe conditions, such as high pressure, high shear strain rate and a very short load time during which the fluid particles pass through the contact region, lubricants do not fully conform to the hypotheses of the classical lubrication theory. The influence of the thermal effect, surface roughness, transient state and the rheology of lubricants are very important for understanding the EHL mechanism.

Another important aspect of EHL research is material wear in terms of engineering. Although wear is generally believed to be caused by mechanical, thermal and/or chemical acts, lubricated wear, especially that caused by roughness, has become one of the most interesting areas in EHL and micro-EHL research. Criteria for lubricant film failure and models of lubricated wear have been proposed.

EHL theory is now developing in depth, thoroughly revealing the physical nature of lubricant films during the lubrication process. EHL films cover a large order of thickness from micrometre to nanometre. As Chang has pointed out in the preface to the present author's book:[1] 'From now on, the challenge is probably to reveal the cross effects and mutual influences between the EHL mechanism and other phenomena, such as thermal, contact and wear in the process of friction.' In short, EHL theory has entered the area of research on the micro-state of lubricated interfaces.

## THE BACKGROUND TO EHL

According to classic hydrodynamic lubrication theory, asperities will contact each other directly when the lubricant film thickness is less than the roughness and lubricant is squeezed out of the contact region. However, the research on EHL and, especially, on micro-EHL, shows that the direct contact of asperities could almost never happen. And, during the development of EHL work, some important phenomena, previously unknown, have been highlighted.

### Properties of the lubricant under high pressure

Between the engineering friction surfaces in concentrated contacts, such as in gears, rolling bearings and valves, pressure is usually in the range of 345 to 3450 MPa. Similar pressure can also be found in boundary lubrication.[2] Experiments show that some properties of lubricants will be changed greatly under such high pressure. For example, the viscosity of cycloalkane mineral oil under a pressure of 750 MPa will be 100,000 times higher than that under normal atmospheric pressure. If the viscosity exceeds $10^5$ Pa·s, the oil will gradually lose its liquid properties. If the viscosity reaches $10^{12}$ Pa·s, it will behave like a glass or elastoplastic solid, whose shear modulus is about $G=10^9$ Pa. The solidifying pressures of several laboratory formulated lubricants at room temperature have been measured as 172.3 to 488.9 MPa by Xu.[3] Jacobson's experiments show that when pressure reaches 1200 MPa, the time for mineral oil to solidify is about 9 μs, less than that for a lubricant particle to pass through a concentrated contact region.

To sum up, under EHL conditions, lubricants will not preserve their properties as under normal atmospheric pressure. The rheological properties of a lubricant must be considered.

## Traction of oil film

Much research on traction of EHL film was carried out in the 1970s in order to develop high efficiency drives with the help of the high viscosity lubricants, according to the viscosity-pressure relationship in EHL. A viscosity-pressure relationship, given by Barus, is as follows:

$$\eta = \eta_0 e^{\alpha p}$$

where $\eta_0$ is the viscosity under the normal atmospheric pressure, and $\alpha$, equal to $(5\text{–}30)\times10^{-3})$ 1/MPa, is a viscosity-pressure index.

The calculating viscosity in EHL can be up to $10^{19}$ Pa·s according to the Barus formula. Obviously, traction coefficient (shearing traction/normal pressure) might be very high if such a viscosity could be realised. However, traction coefficients obtained by experiment are usually in the range 0.04-0.1, not as high as predicted. Research shows that the difference between theory and practice is due to the fact that EHL lubricant film experiences both high pressure and high shear strain rate. The limit of shear stress of a lubricant makes the lubricant behave like a plastic solid at high shear strain rate. Therefore, it is not realistic to try to obtain high traction coefficients.[4]

The non-Newtonian property of lubricants is very important in elasto-hydrodynamic lubrication solutions. Since the limiting shear stress exists, slip will occur on the boundaries of surfaces when shear strain rate is high. A slip EHL solution will be different from a non-slip one.[5] **Figure 1** (see overleaf) gives an example with a constant limiting shear stress (though the limiting shear stress may vary with pressure) to show the effects of the limiting shear stress on EHL solutions. The distributions of pressure and film thickness with boundary slip ($p_H \geq 0.5$ GPa) are quite different from the non-slip one ($p_H$ = 0.49 GPa), as in **Figure 1a**. The slip film thickness in the central contact region becomes much lower than the non-slip one since pressure is constant in the region. After the shear stress of the lubricant reaches the limiting value, the shear stress is truncated (**Figure 1b**).

## The load-bearing capacity of lubricant film

In conventional EHL theory, film thickness is a weak function of load because the deformation of materials and the increase of lubricant viscosity both slow the decrease of film thickness. In recent years, Jacobson,[6] and Huang and Wen[7,8] have

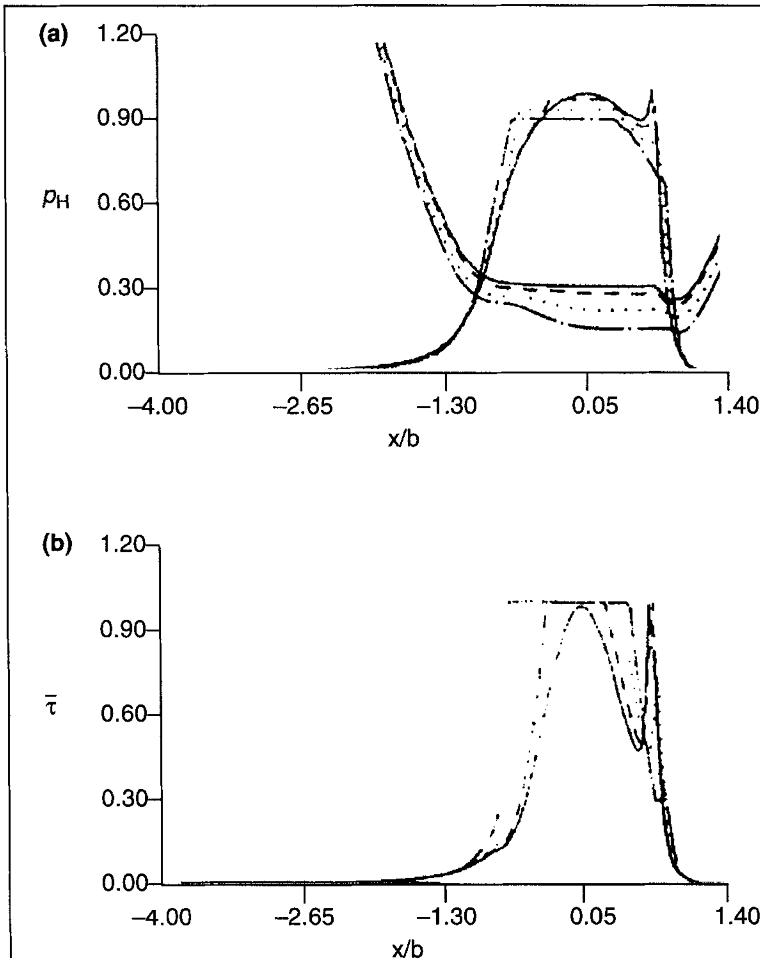



**Figure 1 Solutions of non-yield and yield EHL problems (from reference 5): (a) pressure and film thickness; (b) shear stress (see Nomenclature)**

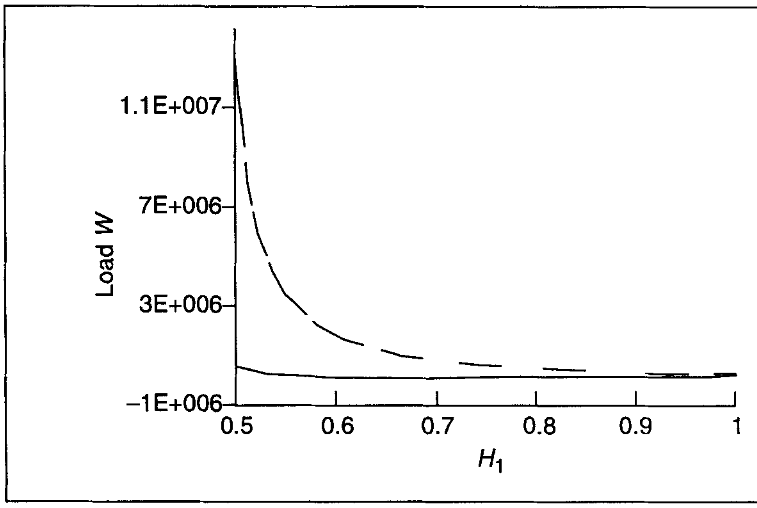
—— 0.49 GPa
- - 0.51 GPa
···· 0.53 GPa
—·— 0.55 GPa
$2(u_1-u_2)/(u_1+u_2) = 0.027$

studied the micro-EHL of rough surfaces in succession. Although different models and analysis methods are used, both studies show that local pressure caused by asperities is high enough to flatten surface roughness. Therefore, a full micro-EHL film may still exist despite roughness, and the lubricant cannot be squeezed out of the contact zone. Therefore, according to conventional EHL theory, an EHL film seems to have an infinite load-bearing capacity.

Obviously, an EHL film between surfaces in concentrated contacts has an enormous potential for load bearing, but the

**Figure 2 Load-bearing capacity ($H_4 = H_1 - 0.5$) (from reference 5) (see Nomenclature)**

—— yield
— — non-yield




load-bearing capacity cannot in fact be infinite. Huang and Wen[5] have further pointed out that the load-bearing capacity of a lubricant film will be greatly decreased if the limiting shear stress is taken into account. A solution of slider lubrication is given in **Figure 2** to show load-bearing capacity curves with and without the limiting shear stress.

## Starved and parched lubrication

In the numerical analysis of EHL, the starting point of a lubricant film influences the film thickness. If the starting point is far enough away from the centre of the contact region, the thickness will experience no obvious change. However, if the starting point is not far enough away, the thickness will be less than that in fully-flooded lubrication under the same load. The film thickness reflects the degree of lubricant supplication.

In recent years, another severe starvation state, 'parched' lubrication, has been studied. Under conditions of very little or no lubricant supply, lubricant film can also be formed since a very thin lubricant layer is adsorbed by the surfaces. On rolling bearings at high speed in a gyro, Kingsbury's experiments[9] show that a very thin fluid film, dependent on the lubricant layer remaining on the bearing surfaces, existed in the contact zone even if no lubricant is supplied. The experiments also show that satisfactory lubrication will be obtained if the deposited film has a thickness of 80-200 nanometres. The characteristics of fully-flooded, starved and parched lubrication have been analysed by Liu and Wen,[10] and a method of determining the three regimes of lubrication is given.

The above shows that lubrication films can exist in thicknesses of the order of micrometres, sub-micrometres and even nanometres.

## Minimum oil film thickness limit for application of EHL theory

Elastohydrodynamic lubrication similar to hydrodynamic lubrication is based on the Reynolds equation, a special form of the Navier-Stokes equation in viscous fluid mechanics. It belongs to continuous medium mechanics.

According to the film thickness formula, the EHL films of some realistic gear drives or rolling bearings would be very thin, even merely several dozen layers of lubricant molecules. Obviously, such thin films are not thick enough to assure the conditions of a continuous medium. Therefore, the EHL theory loses its basis in such situations, and two questions arise: one is how to determine the minimum limit of lubricant film thickness to which the EHL theory can be correctly applied; the other is how to evaluate the behaviour of thin film lubrication as EHL theory gradually loses its application basis. To answer the first question, the work of Johnston *et al.*[11] shows that the variation of film thickness with speed will deviate from that predicted by the EHL theory when film thickness is less than 10 nanometres. The molecule size of common lubricants is 0.5-3 nanometres, so a film thickness of 10 nanometres corresponding to 3-30 layers of molecules can be thought of as the minimum limit of EHL films. To answer the second question, more research needs to be carried out on the regime of lubrication of the order of sub-micrometre and nanometre.

Lubricant film thicknesses in engineering applications run from thicknesses of several layers of molecules to thousands of micrometres, which includes many different lubrication regimes. Different regimes have their own typical ranges of thicknesses. The Streibeck curve is widely used to distinguish between fluid film lubrication (including hydrodynamic lubrication and elasto-hydrodynamic lubrication) and boundary lubrication, the two familiar lubrication regimes. But for the regimes between these, research work has been scant and a common understanding has not yet been reached. It may be an important area to study in modern lubrication theory.

To sum up, the thorough development of EHL theory has revealed many phenomena and raised problems, which in turn leads to the exploration of a new regime of lubrication, the thin film lubrication regime, between the elastohydrodynamic lubrication and boundary lubrication.

## ON THIN FILM LUBRICATION

Although the concept of thin film lubrication is much more recent and is differently understood, much attention has been paid to it.

Obviously, problems in thin film lubrication are very complicated and difficult both in experiment and in theory. Therefore, the research work is still at its beginning stage. Some encouraging progress has been made in the measurement techniques of ultra-thin film lubrication and in experimental investigation of thin film lubrication of different lubrication mediums.

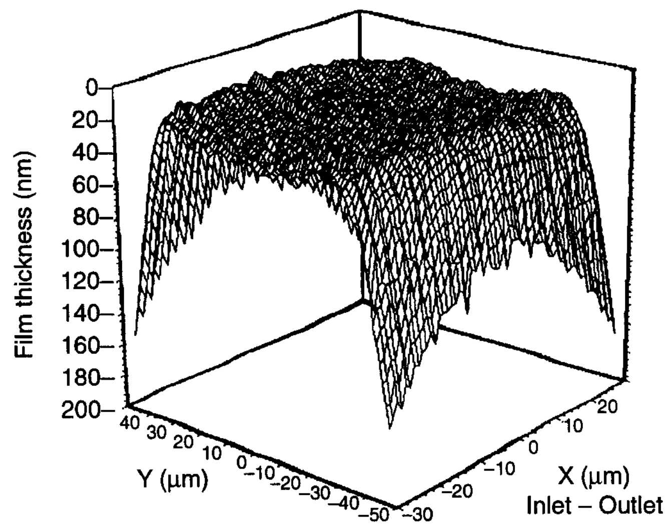
The measurement of thin lubricant films has been achieved in the order of nanometres by using improved optical interferometry of elastohydrodynamic film thickness, as achieved by Spikes *et al.*[12] At very low rolling speeds, lubricant films from 5 to 10 nanometres thick are formed in the point contact zone. The film thicknesses correspond to 3-30 layers of lubricant molecules. Subsequently, the apparatus has been used to conduct experiments with different lubricants. During research on the mechanism of anti-wear additives, a thin film of viscous lubricant has been observed. For example, ZDDP films are found to be only 10-50 nanometres thick.[11,13] In addition, the properties and behaviour of thin film lubrication of non-hydrocarbon-based lubricants[14] and emulsions[15] have also been evaluated. The optical interference technique and an optical total reflection film thickness measurement apparatus working in the order of nanometres have been developed by the National Tribology Laboratory of Tsinghua University. The precision and resolution of the measurement has been further improved. This has shown that full thin film lubrication between surfaces in point contacts can be formed at very low speeds (**Figure 3**, see overleaf).[16]

Further experiments show that the thin film lubrication has both similarities and differences with EHL. Variation of film thickness is proportional to velocity, but with a lower slope than that calculated according to the Dowson-Hamrock formula, **Figure 4** (see p. 283).

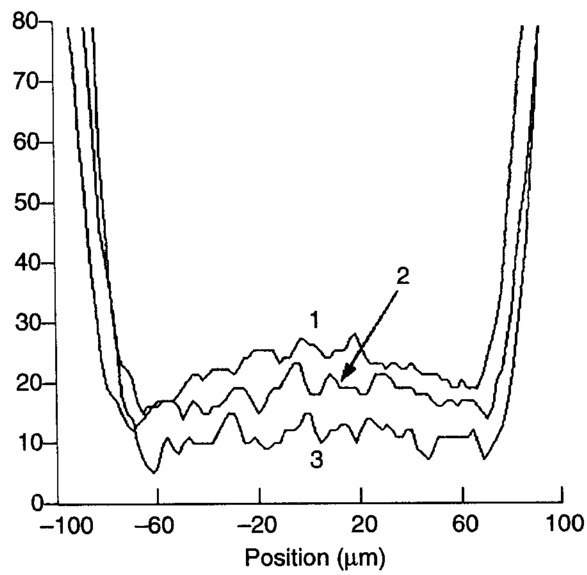
Research on thin film lubrication is important in developing lubrication and wear theory, and hence in modern science

**Figure 3 Film thickness in Hertz region. Temp: 25°C; load: 4 N**

(a) Film thickness in contact region
Lubricant 13602:
viscosity = 4.21 mPa·s/20°C
refractive index = 1.4616
Rolling speed = 18.6 mm/s




(b) Film thickness along A-A
Lubricant 13604:
viscosity = 17.4 mPa·s/20°C
refractive index = 1.4733
Rolling speed:
1 = 16.55 mm/s
2 = 11.53 mm/s
3 = 3.12 mm/s



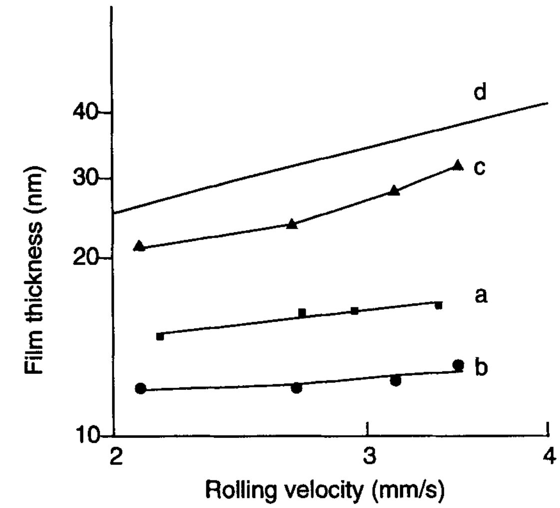



**Figure 4 Average film thickness at the centre (temp: 20°C; load: 4 N; ball: ø 20.4 mm)**

Oil a, viscosity = 190 mPa·s/20°C, RI = 1.4871
Oil b, viscosity = 84 mPa·s/20°C, RI = 1.4836
Oil c, viscosity = 36 mPa·s/20°C, RI = 1.4771
d = Dowson formula

and technology. For example, thin film lubrication is usually one of the key techniques contributing to the correct operation of some high-technology equipment and precision machinery. Other application examples can be found in machinery working in wet industrial conditions. Other areas include high temperature, anti-wear additives, ultra-low speed and rough surfaces.

Thin film lubrication is in the transitional regime between boundary lubrication and fluid lubrication. According to Dowson,[17] the four lubrication regimes are boundary, mixed, elastohydrodynamic and hydrodynamic. It is believed that when film is so thin that the contact gap between asperities is in the range of sizes of lubricant molecules, a boundary lubrication film would appear and the lubrication begins to transit to mixed lubrication.

Stearic acid is a typically oily additive whose molecule length is 3 nm. If a monomolecular layer is adsorbed on each surface, the film thickness will be 6 nm. Therefore, it can be supposed that 30 nm would be the film thickness going from boundary lubrication to mixed lubrication in so far as multi-molecule layers and other factors are considered. The thin film lubrication discussed in the present paper covers a wider area than the above. If rough surfaces are taken into account, the film thickness especially may change considerably because of the variation in the surface profile. Therefore, thin film lubrication should exist over a large range.

Research into thin film lubrication will be complex, because of the particular behaviour of thin lubricant film. First, thin film has the characteristics of both fluid film and boundary film, so the research involves fluid mechanics, rheology, elastic and plastic mechanics, thermodynamics, surface physics and chemistry, stochastic theory and molecular dynamics, among others. Secondly, since its thickness is very small, the geometric topography and physicochemical properties of surfaces are influencing factors which cannot be ignored. Thin film lubrication of rough surfaces is actually in the regime of mixed lubrication consisting of lubricant films of different thicknesses, whereas each different thickness has different properties. Thus the properties of thin film lubrication are the result of the aggregation and combination of the properties of the component films. Moreover, since the geometric topography of rough surfaces is random, the percentage of every component film changes with time during the process of friction, so that the properties of thin film lubrication are strongly random and time-dependent. All these make research on thin film lubrication complex.

Such research has only recently started. In addition to experiment investigations, some theoretical calculations have also been reported. The mechanism, characteristics, and failure criteria of thin film lubrication are not clearly known. The author believes that emphasis should be put on the following aspects of future research work:

- development of in-situ measurement techniques and precision measurement of the lubricant film in the order of nanometres to offer a reliable means for experimental research.
- properties and behaviour of the lubricant film, its relations to load, speed, temperature and surface pattern
- characteristics of mixed lubrication and laws relating to changes
- modelling of thin film lubrication and numerical calculation
- new lubricant media and surface engineering to improve the performance of thin film lubrication
- failure and its mechanism in thin film lubrication
- applications of thin film lubrication.

Since Tower discovered, in 1883, that the hydrodynamic pressure could be created between friction surfaces lubricated with lubricant, a century has passed. With research and application, lubrication theory has facilitated technical progress in machinery and become an important component branch of mechanical design. Today's highly developed science and technology will also promote thorough research on the subject of lubrication. It can be predicted that the theory of thin film lubrication will be gradually perfected, and open up the prospects for its wide application.

## Acknowledgement

The research work for the present paper was financially supported by the National Natural Science Foundation of China. The author would like to thank especially Dr. F.R. Kong, Dr. P. Huang, and PhD student J.B. Luo, for their contributions to the paper.

## References

1. Wen, S.Z., and Yang, P.R., *Elastohydrodynamic Lubrication*, Tsinghua University Press, 1992 (in Chinese).
2. Booser, E.R., *CRC Handbook of Lubrication*, CRC Press, Boca Raton, 1984.
3. Xu, H.P., 'The development of high pressure viscosity apparatus and performance of tailor-made lubricant oil under high pressure', Ph.D. thesis, 1991 (in Chinese).
4. Chang, H.S., Spikes, H.A., and Bunemann, T.F., 'The shear stress properties of ester lubricants in elastohydrodynamic contacts', *J. Synth. Lubr.*, **9**, 2, 91 (1992).
5. Huang, P., and Wen, S.Z., 'A new model of visco-plastic fluid lubrication for sliding problems', *Acta Tribologica of Romania*, **2**, 23-30 (1994).
6. Jacobson, B., 'Mixed lubrication', *Eurotrib '89*, Proc. of 5th Int. Cong. on Trib., Vol. 1, Finland, 1989.
7. Huang, P., and Wen, S.Z., 'Study on oil film and pressure distribution of micro-EHL', *Trans. ASME, J. of Trib.*, **114** (1992).
8. Huang, P., and Wen, S.Z., 'Sectional micro-elastohydrodynamic lubrication', *Trans. ASME, J. of Trib.*, **115** (1993).
9. Kingsbury, E., 'Parched elastohydrodynamic lubrication film thickness measurement in an instrument ball bearing', *Trib. Trans.*, **33** (1990).
10. Liu, J.H., and Wen, S.Z., 'Fully flooded, starved and parched lubrication of a point contact system', *Wear*, **159** (1992).
11. Johnston, G.J., Wayte, R., and Spikes, H.A., 'The measurement and study of very thin lubricant film in concentrated contacts', *Trib. Trans.*, **34** (1991).
12. Spikes, H.A., and Guangteng, G., 'Properties of ultra-thin lubricating film using wedged spacer layer optical interferometry', Proc. 14th Leeds-Lyon Symposium on Tribology, 1988.
13. Gunsel, S., Spikes, H.A., and Aderin, M., '*In-situ* measurement of ZDDP film in concentrated contacts', *Tribology 2000 International Colloquium*, Technische Akademie Esslingen, Esslingen, 1992.
14. Aderin, M., Johnston, G.J., Spikes, H.A., and Caporiccio, G., 'The elastohydrodynamic properties of some advanced non-hydrocarbon-based lubricants', STLE Preprint No. 91-TC-6D-6, STLE/ASME Tribology Conference, 1991.

15. Barker, D.C., Johnston, G.J., and Spikes, H.A., 'Behaviour of emulsion oil particles in concentrated contacts', *Proc. 17th Leeds-Lyon Symposium on Tribology*, 1991.
16. Luo, J.B., Huang, P., and Wen, S.Z., 'The study of thin film lubrication with relative optical interference intensity technique', to be published.
17. Dowson, D., 'Transition to boundary lubrication from elastohydrodynamic lubrication', in *Boundary Lubrication – An Appraisal of World Literature*, ASME, 1969.

## Nomenclature

| | | | |
|---|---|---|---|
| $B$ | width of the sliding block, m | | |
| $b$ | half line contact Hertzian width, m | | |
| $H$ | dimensionless film thickness, | $H = hR/b^2$ | (Fig. 1) |
| | | $H = h/h_m$ | (Fig. 2) |
| | $H_1$ is the dimensionless inlet film thickness and $H_4$ the outlet. | | |
| $h$ | film thickness, m | | |
| $h_m$ | the critical film thickness, m | $h_m = \eta_0(u_1-u_2)\tau_L$ | |
| $P$ | dimensionless pressure, | $P = p/p_H$ | (Fig. 1) |
| | | $P = p/\tau_L$ | (Fig. 2) |
| $p$ | pressure, Pa | | |
| $p_H$ | the maximum Hertzian stress, Pa | | |
| $R$ | equivalent radius, m | $1/R = 1/R_1 + 1/R_2$ | |
| $u$ | velocity, m/s. Where $u_1$ and $u_2$ are velocities of upper and lower surfaces. | | |
| $W$ | dimensionless load, | $W = 400 \times \int_{x1}^{x2} PdX$ | (Fig. 2) |
| $X$ | dimensionless coordinate, | $X = x/b$ | (Fig. 1) |
| | | $X = x/B$ | (Fig. 2) |
| $x$ | coordinate in the direction of velocity, m | | |
| $\eta_0$ | zero pressure lubricant viscosity, PaS | | |
| $\tau$ | shear stress | | |
| $\tau_L$ | limit shear stress, Pa | | |
| $\bar{\tau}$ | dimensionless shear stress, $\bar{\tau} = \tau/\tau_L$ | | |