FRICTION, WEAR AND LUBRICATION

MEMM 1343

FRICTION

Friction is the mechanical force which resists movement (dynamic or kinetic friction) or hinders movement (static friction) between sliding or rolling surfaces. This force of friction is also called "external friction". The work done by friction can translate into deformation, wear, and heat.

Internal friction as part of external friction when lubricants are used results from the friction between lubricant molecules; this is described as viscosity. The meaning of internal friction in material science is the force resisting motion between the elements making up a solid material while it undergoes deformation.

The causes of external friction are, above all, the microscopic contact points between two sliding surfaces; these cause adhesion, material deformation, and grooving.

Energy which is lost as friction can be measured as heat and/or mechanical vibration.

Lubricants should reduce or avoid the microcontact which causes the greatest part of external friction.

Basic idea that we need to know are:

Kinetic and static friction	Coefficient of friction
Stick – slip	Sliding and rolling friction

Kinetic and static friction

Different from static friction, kinetic friction occurs under conditions of relative motion.

The American Society for Testing and Materials (ASTM) defines the kinetic coefficient of friction as the coefficient under conditions of macroscopic relative motion of two bodies. The kinetic coefficient of friction, which sometimes is called the “dynamic coefficient of friction”, is usually somewhat smaller than the static coefficient of friction.

The static coefficient of friction is defined as the coefficient of friction corresponding to the maximum force that must be overcome to initiate macroscopic motion between two bodies (ASTM). The maximum value of static friction is sometimes referred to as “limiting friction”.

Coefficient of friction

Figure 1 defines the coefficient of friction as the dimensionless ratio of the friction force, F and the normal force, N. The proportionality between normal force and friction force is often given in dry and boundary friction, but not in fluid film lubrication.

The coefficient of friction is also known as the frictional coefficient or friction coefficient, and is symbolized by the Greek letter μ (or sometimes by f).

The coefficient of friction is not a material property, but is better described as a system property. It cannot be deduced by calculations, but must be determined by empirical measurements.

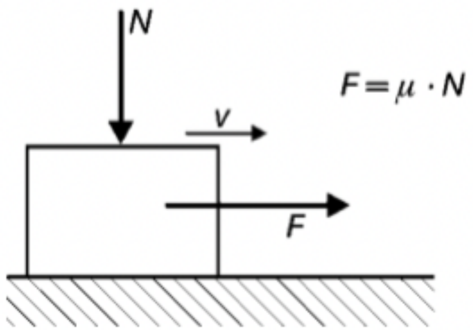
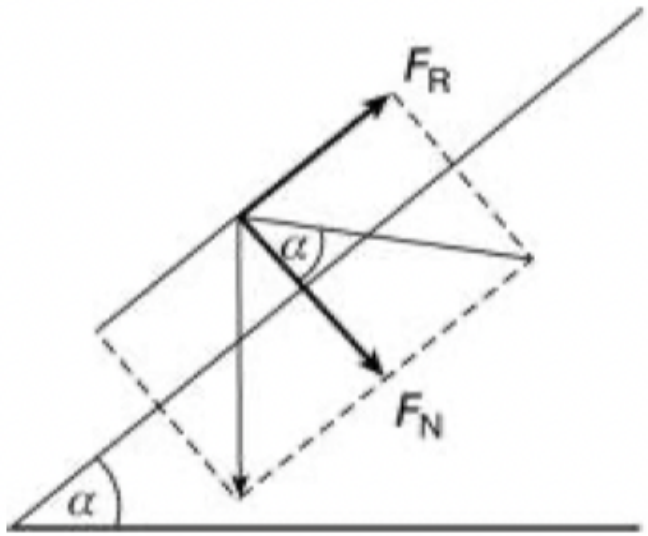


Figure 1: Coefficient of friction, μ.

$$\mu = \frac{F}{N}$$

For several combinations of material in a dry and clean state, and under same test procedure, the static coefficient of friction for steel against the most metals is found between 0.4 and 0.6. (The kinetic coefficient of friction relating to the various friction regimes is shown in Stribeck curve)

On occasion, the static coefficient of friction is defined in terms of the maximum angle before which one of the items will begin to slide. This is the "angle of friction', where α is the angle from horizontal and μ_{st} is the coefficient of static friction as shown in Figure 2.



$$\mu_{st} = \frac{F_R}{F_N}$$

$$\mu_{st} = \frac{\sin\alpha}{\cos\alpha}$$

$$\mu_{st} = \tan\alpha$$

Figure 2: The static coefficient of friction in terms of the maximum angle before which one of the items will begin sliding.

Stick-slip

Stick-slip is a special form of friction which often results from very slow sliding movements when the friction partners are connected to a system which can vibrate.

The process is influenced by the dependence of the coefficient of sliding friction on speed.

This generally occurs when the static coefficient of friction is larger than the kinetic coefficient of friction.

Stick-slip is normally encountered with machine tools which operate with slow feeds, and can cause chatter marks on components. The inclusion of special additives in lubricants, or special surface treatment of the bodies, can lead to stick–slip being reduced or even avoided, as shown in Figure 3.

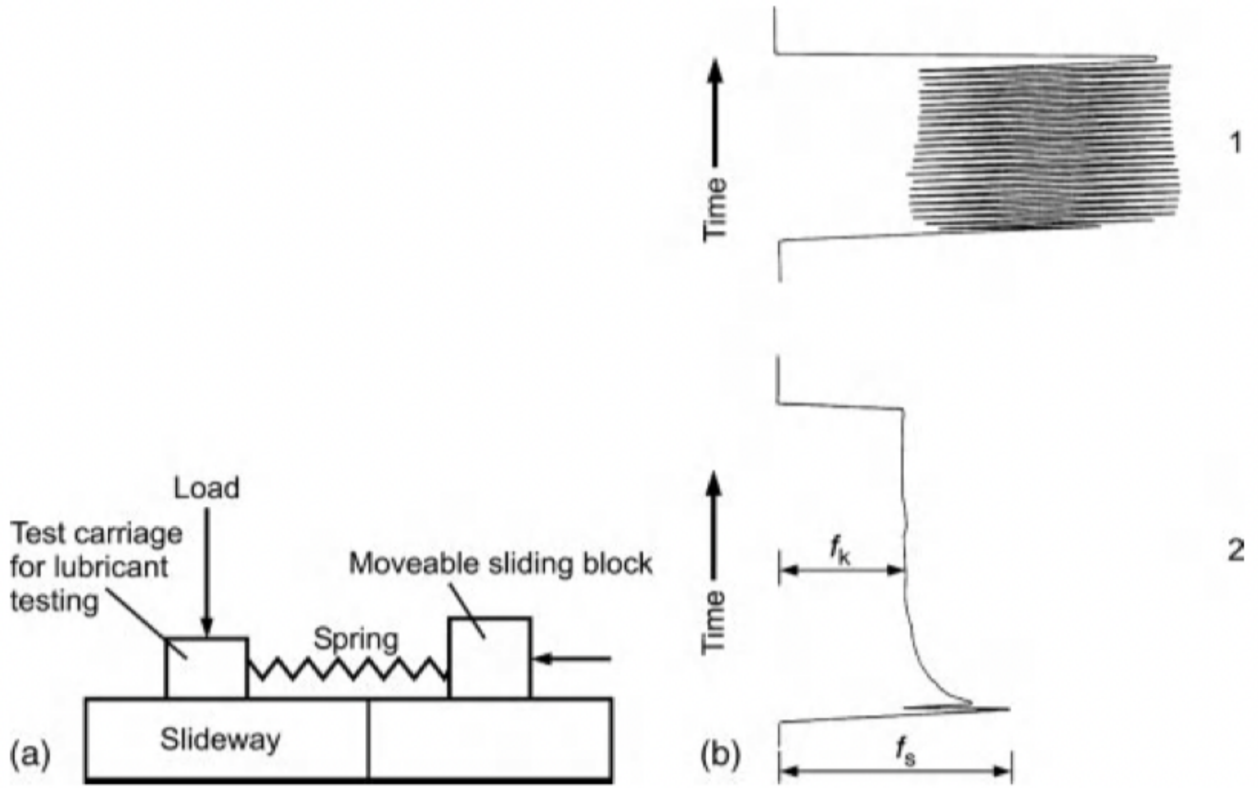


Figure 3: Stick-slip. (a) Test equipment for stick-slip; (b) Results of stick-slip behavior of two oils.
1: Oil with bad stick-slip behavior
2: Oil with good stick-slip behavior;
f_k : Relative kinetic coefficient of friction
f_s : Relative static coefficient of friction.

Sliding and rolling friction

Sliding friction is friction in a pure sliding motion with no rolling and no spin, as shown in Figure 4.

Rolling friction is the friction generated by rolling contact. In roller bearings, rolling friction mainly occurs between the rolling elements and the raceways, whereas sliding friction occurs between the rolling elements and the cage.

The main cause of friction in roller bearings is sliding in the contact zones between the rolling elements and raceways. It is also influenced by the geometry of the contacting surfaces and the deformation of the contacting elements.

Although the existing slip velocities are small (normally $< 1\%$), they may produce a major part of the total resisting in rolling.

In addition, sliding also occurs between the cage pockets and the rolling elements.

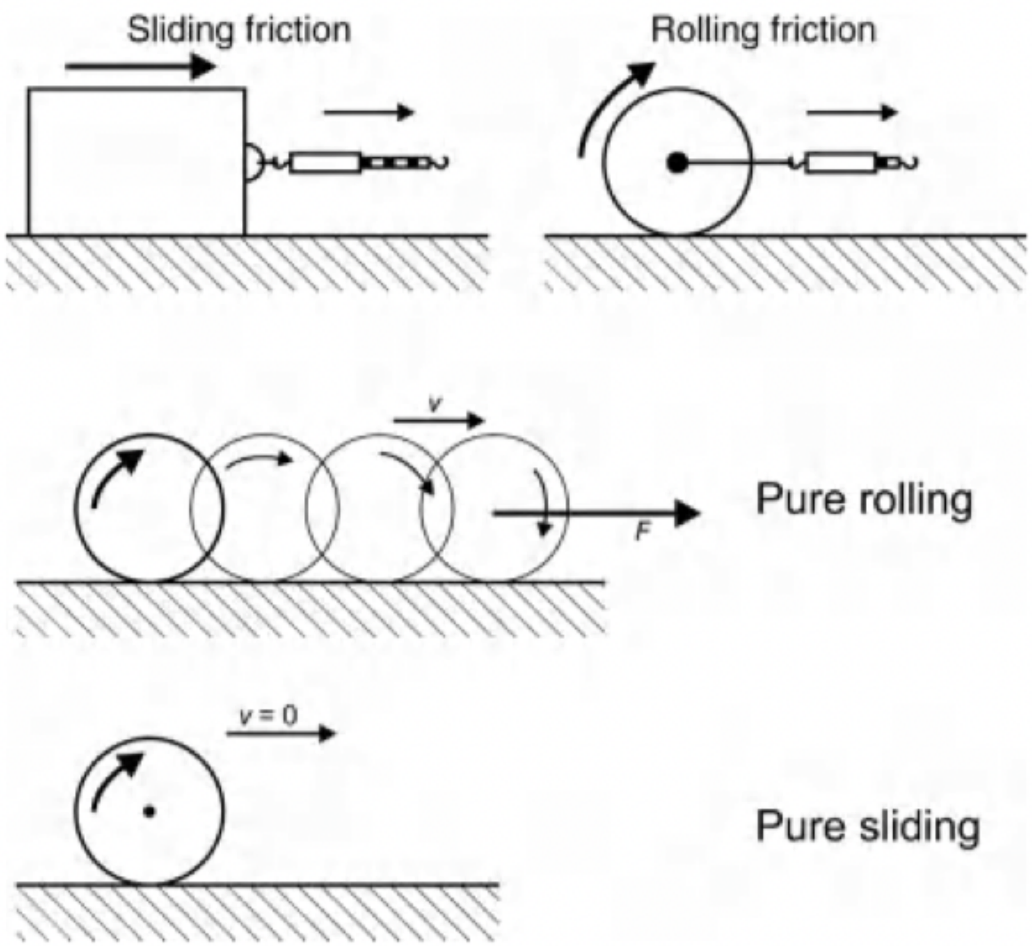


Figure 4: Sliding and rolling.

In the case when rolling motion and sliding motion combine to any significant extent, as for gear tooth meshing, a special terminology has been created.

The term “Walzen” (rolling, e.g., steel rolling), is used in Germany. Situations in which a high sliding/rolling ratio occur require totally different lubrication than does pure sliding.

Figure 4 and Figure 5 show this “rolling friction” during rolling and during gear meshing.

A special form of sliding friction is called spin, or spin friction; this can be illustrated by a drilling operation or a rotating cone on a plate.

In ball bearings, this type of friction can be seen when analysing the rotating movement of the balls round various axes.

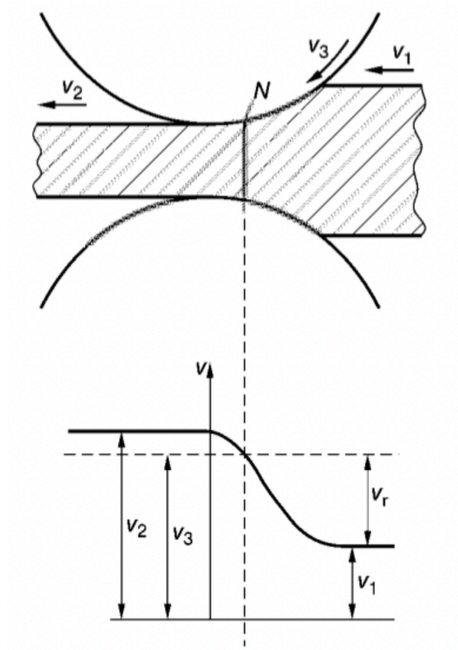


Figure 4: “Walzen” phenomenon, the mixing of rolling and sliding motions in rolling metal forming.

V1 : Initial speed of the sheet metal , V2 : Final speed of the sheet metal

V3 : Speed of the roller , V4 : Speed difference in the roll gap (sliding part)

N : Neutral point (non-slip, pure rolling)

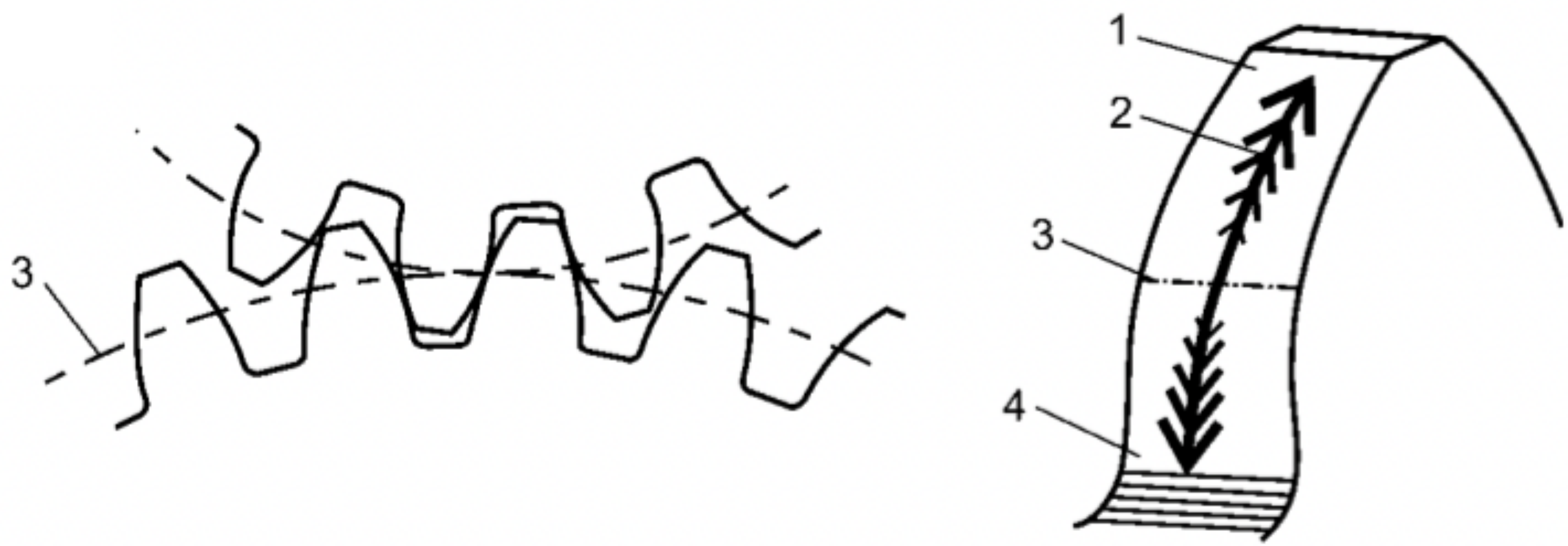


Figure 5: Engagement of gear teeth.

Points 1, 2 and 4 : High sliding/rolling ratio

Point 3 : Pitch circle (pure rolling, no slip).

FRICTION REGIMES: FRICTION AND LUBRICATION CONDITIONS

In tribological systems, different forms of contact can exist between contacting partners, and this results in different friction and lubrication conditions.

Solid Friction (Dry Friction)

Boundary Friction

Fluid Film Friction

Mixed Friction

Stribeck Curve

Solid Friction (Dry Friction)

This occurs when two solids have direct contact with each other, without a separating layer. If conventional materials are involved, the coefficients of friction and wear rates are high; lubrication technology attempts to eliminate this condition.

Boundary friction

The contacting surfaces are covered with a molecular layer of a substance, the specific properties of which can significantly influence the friction and wear characteristics. One of the most important objectives of lubricant development is to create such boundary friction layers in a variety of dynamic, geometric, and thermal conditions.

These layers are of major importance in practical applications when thick, long lasting lubricant films to separate two surfaces are technically impossible to achieve.

Boundary lubricating films are created from surface-active substances and their chemical reaction products. Adsorption, chemisorption, and tribochemical reactions also play significant roles.

Although boundary friction is often allocated to solid friction, the difference is of major significance to lubricant development and the understanding of lubrication and wear processes, especially when the boundary friction layers are formed by the lubricants.

Fluid Film Friction

In this form of friction, both surfaces are fully separated by a fluid lubricant film (full-film lubrication) which is formed either hydrostatically or, more commonly, hydrodynamically.

From a lubricants point of view, this is known as hydrodynamic or hydrostatic lubrication (Figure 6).

Liquid or fluid friction is caused by the frictional resistance, owing to the rheological properties of fluids.

If both surfaces are separated by a gas film, this is known as “gas lubrication”.

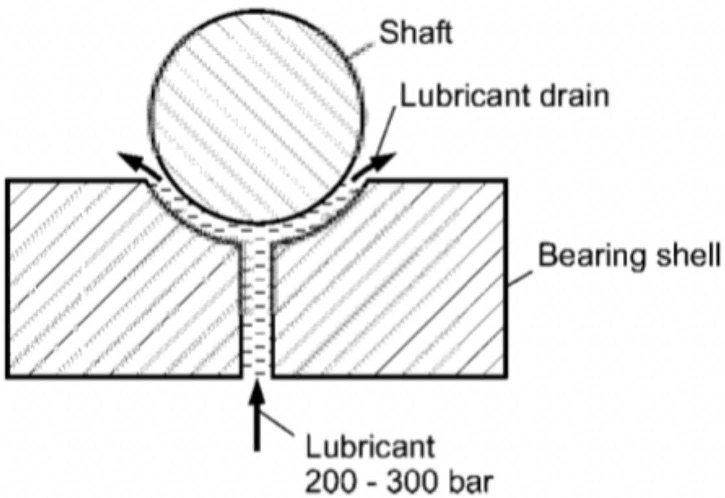


Figure 6: Hydrostatic lubrication as a form of fluid friction.

Stribeck Curve

The friction or lubrication conditions between boundary mixed and fluid friction are graphically illustrated by using the Stribeck diagram/curve, first developed in 1902 (Figure 7).

These curves are based on the starting-up of a plain bearing, the shaft and bearing shells of which are, when stationary, separated only by a molecular lubricant layer.

As the speed of revolution of the shaft increases (peripheral speed), a thicker hydrodynamic lubricant film is created; initially this causes a sporadic mixed friction which, nevertheless, causes a significant reduction in the coefficient of friction.

Then, as the speed continues to increase, a full, uninterrupted film is formed over the entire bearing faces, causing a sharp reduction in the coefficient of friction.

As the speed increases further, internal friction in the lubricating film adds to the external friction such that the curve passes a minimum coefficient of friction value and then increases, solely as a result of internal friction.

The lubricant film thickness shown in Figure 7 depends on the friction and lubrication conditions including the surface roughness,

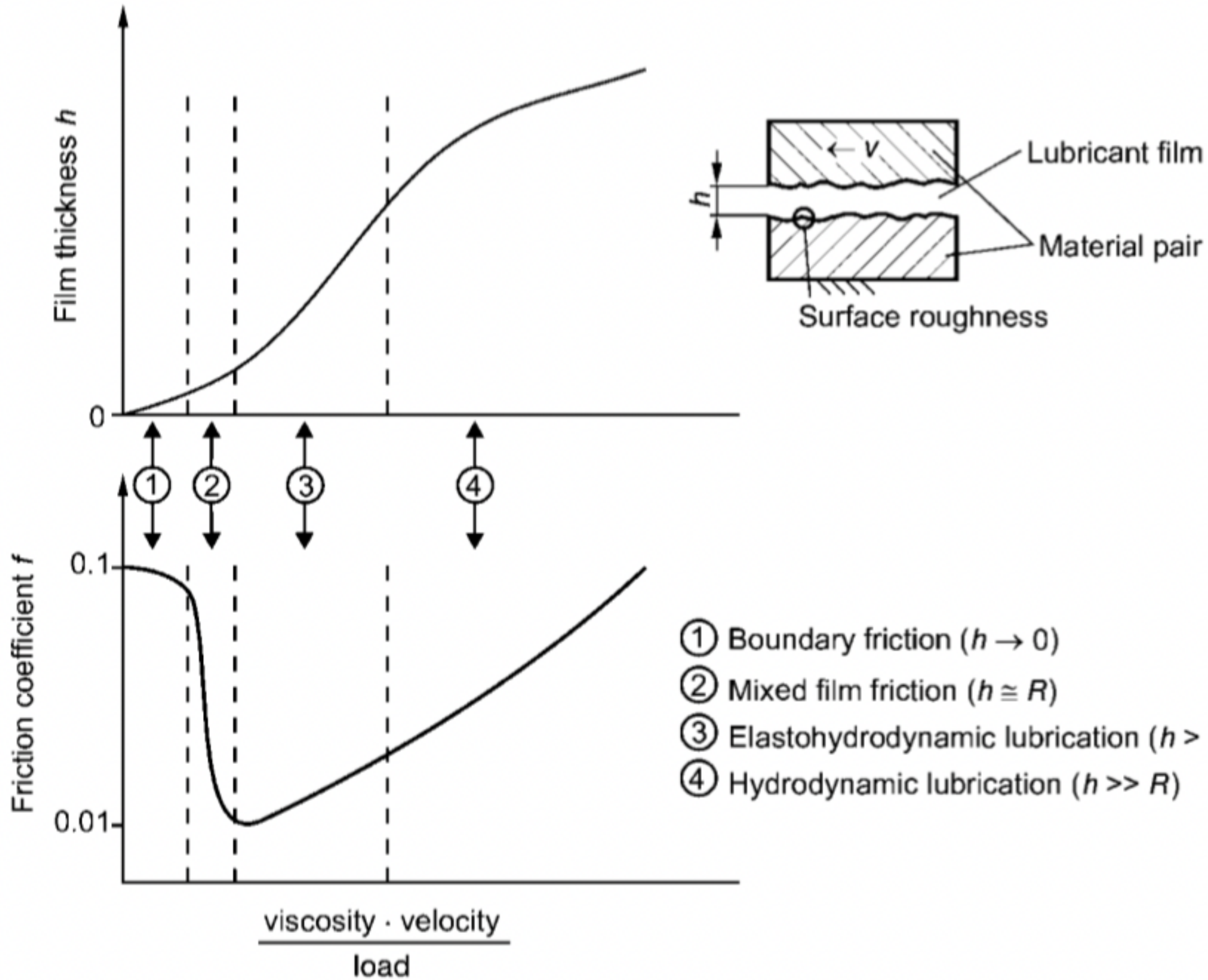


Figure 7: The Stribeck curve.

HYDRODYNAMIC AND ELASTOHYDRODYNAMIC LUBRICATION

The formation of a hydrodynamic fluid film is shown in Figure 8. Here, the lubricant is pulled into the conical converging clearance by the rotation of the shaft. The created dynamic pressure then carries the shaft.

Based on the Navier–Stokes theory of fluid mechanics, Reynolds created the basic formula for hydrodynamic lubrication in 1886. Several criteria remained excluded, however, especially the influence of pressure and temperature on viscosity.

Application of the Reynolds' formula led to theoretical calculations on plain bearings, where the only lubricant value was viscosity.

Main parameter to know are:

Elastohydrodynamic Lubrication (EHL) Regime

Thermoelastohydrodynamic Lubrication (TEHL)

Elastohydrodynamic Lubrication (EHL) Regime

Hydrodynamic calculations on lubricant films were extended to include the elastic deformation of contact faces (Hertzian contacts, Hertz's equations of elastic deformation) and the influence of pressure on viscosity.

This enables the application of these elastohydrodynamic (EHD) calculations to contact geometries other than that of plain bearings – for example, those of roller bearings and gear teeth.

Figure 8 shows the elastic deformation of the ball and raceway of a ball bearing, while Figure 3.9 shows an example of Hertzian contacts for various pairs with nonconverging lubricant clearance.

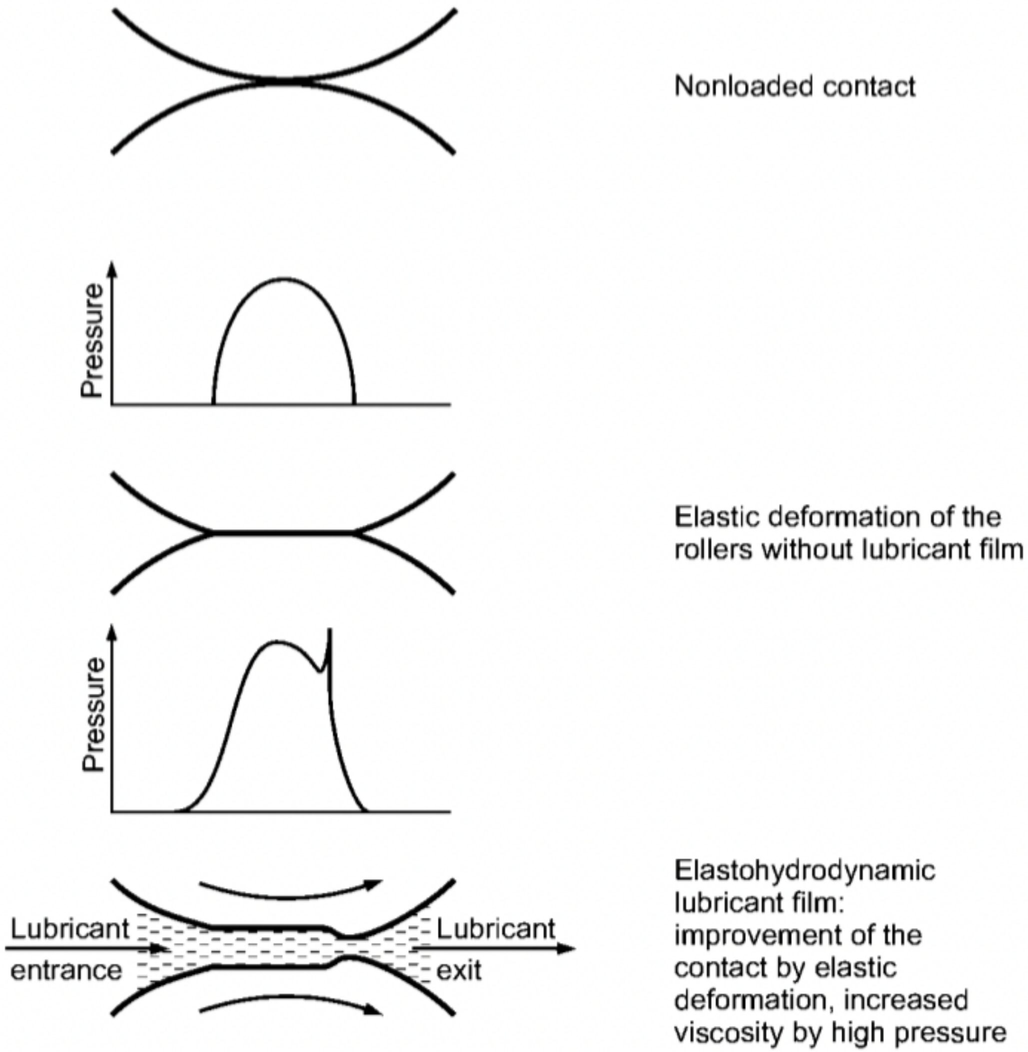


Figure 8: Improvement of hydrodynamic lubrication clearance between two rollers by Hertzian deformation [elastohydrodynamic (EHD) contact], pressure distribution in the Hertzian contact.

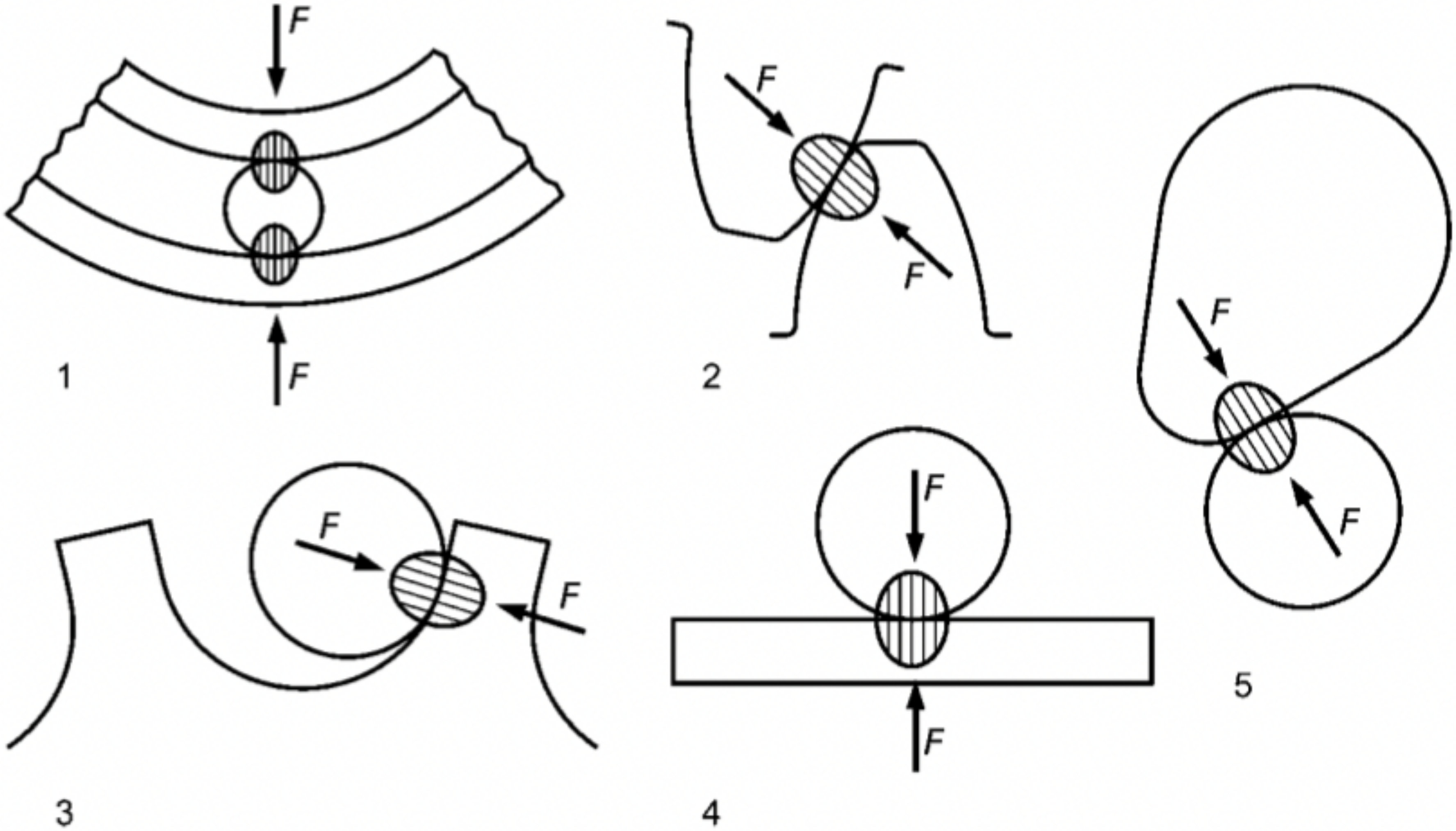


Figure 9: Hertzian contact for different pairs with nonconverging lubricant clearance. 1) Roller bearing, 2) Gear wheels, 3) Chain wheels, 4) Roller on flat path, 5) Cam lifter.

The EHL regime is relevant in lubricated nonconforming contacts, as are found in rolling element bearings, gears, and cam-tapped systems.

The fluid behaviours caused by motion of the surface can be calculated by using the Reynolds equation, similar to conforming contacts. In nonconforming contacts with a high elastic modulus, solids such as metals develops a very high pressure, so that the load is concentrated in a line or a point.

However, this has effects on the lubrication behaviour: typically, the high pressure causes the contacting surfaces to be elastically deformed, which in turn causes a large increase in the viscosity of the lubricant in the contact zone. This combined effect is known as “piezoviscous-elastic lubrication”.

In 1945, in order to solve the mathematical description of EHL, Hertel combined the solution of the three equations for fluid flow, elastic deformation of the contacting solids, and of the dependence of fluid viscosity on pressure. In this way, he obtained the following equation for the fluid film thickness in line contact (parallel cylinders in contact):

$$h_0 = 1.93 \times \frac{(\eta U \alpha)^{0.727} R'^{0.364} E'^{0.091}}{W_L^{0.091}}$$

where W_L is the applied load per unit length, U is the mean rolling speed, and R' is the reduced radius:

$$\frac{1}{R'} = \frac{1}{R_1} + \frac{1}{R_2}$$

and E' is the reduced elastic modulus:

$$\frac{2}{E'} = \frac{1 - v_1^2}{E_1} + \frac{1 - v_2^2}{E_2}$$

where η is the viscosity of fluid at atmospheric pressure, and α is the pressureviscosity coefficient of the fluid.

Later calculations of film thickness for EHL contacts were reported by Dowson and Higginson. For the ratio of the minimum film thickness and the normal film thickness, an approximate value of 0.75 was obtained.

Figure 3.10 demonstrates the calculations of film thickness of Dowson and Higginson, according to G. Knoll, for two contacting balls.

Based on different approaches, various research groups identified variant values of the exponents, with E_2-values ranging from 1.00 to 0.73.

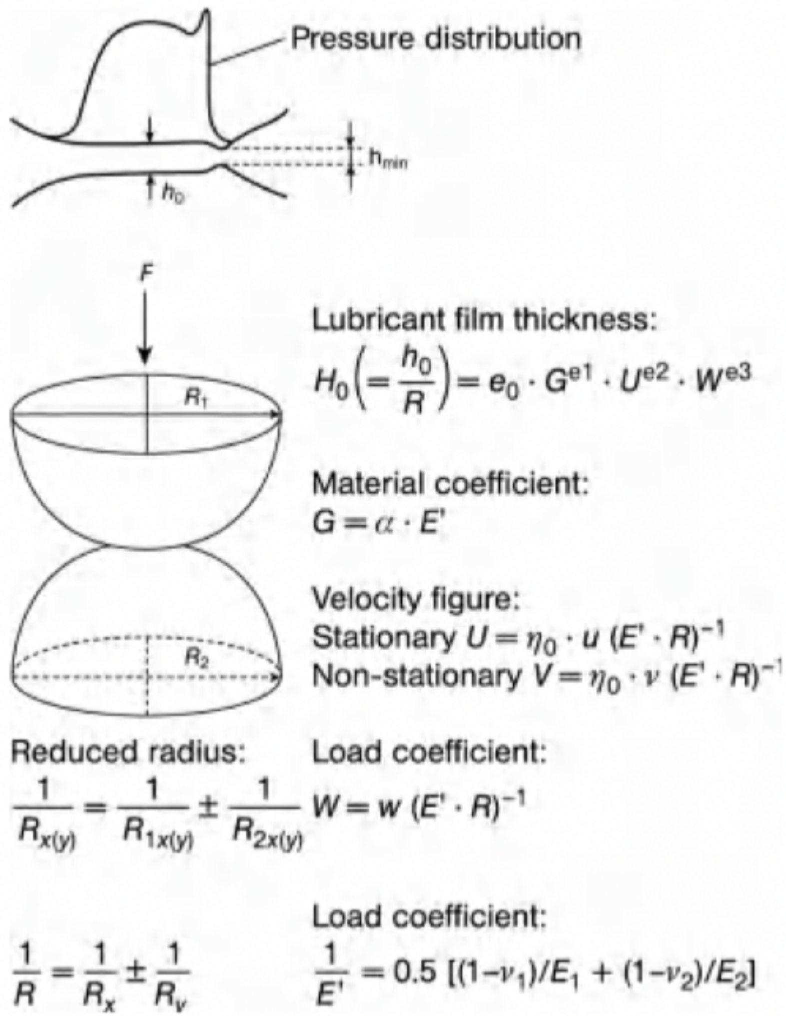


Figure 10: Calculation of film thickness of Dowson and Higginson (according to G. Knoll for two contacting balls).

Numerous studies have been conducted during the past years to predict the EHL conditions, by combining the Reynolds equation with the elasticity and 3-D energy equations, so as to obtain the pressure, film thickness, and temperature distribution within the lubricant film.

In addition, computer-controlled instruments have been developed for measuring the film thickness down to 1 nm and defining the conditions of EHL.

Yet, despite a more than 60-year history of EHL applications, further improvements in the regime have been demonstrated during recent years.

Thermoelastohydrodynamic Lubrication (TEHL)

TEHL theory solves the Reynolds equation, including the energy equation of the lubricant film. Calculation of the energy takes into consideration heat convection in all directions, heat conduction in the radial direction, compression, and heating caused by viscous and asperity friction.

TEHL theory has been applied, for example, in important areas of automotive engines, by using a model that includes a shear rate-dependent viscosity, and also in the simulation of lubrication conditions for the main crankshaft bearing of commercial automotive engines.

In metal-working processes – and especially in metal forming - the lubricant film thickness is formed by wedging before the workpiece enters the plastically deformed working zone. Although the geometric and pressure conditions of this zone are important parameters, in many processes a reasonable lubricant film can be generated hydrodynamically. This special lubrication condition, occurring under plastic deformation, is termed “plastohydrodynamic lubrication”.