# 9. Tribology

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The main subjects of this chapter are the tribological system, friction, wear and lubrication. Regarding the tribological system essential information on structure, real contact geometry, tribological loads, operating and loss variables are provided. Concerning friction, the different friction types, states and mechanisms are discussed. In the sections on wear a lot of details on types and mechanisms of wear, wear profiles and the determination of wear and the average lifetime are introduced. The sections on lubrication contain relevant expositions on the lubrication states, like hydrodynamic, elastohydrodynamic, hydrostatic, mixed and boundary lubrication and lubrication with solid lubricants. Lubricants like mineral, synthetic and biodegradable oils and additives, lubricating greases and solid lubricants are also considered, as are the properties of lubricants, like the dependence of oil viscosity on temperature, pressure and shear rate, and the consistency of lubricating greases.

## 9.1 Importance of Tribology

*Tribology* is the science and technology of interacting surfaces in relative motion. Tribology includes boundary-layer interactions both between solids and between solids and liquids and/or gases. Tribology encompasses the entire field of friction and wear, including lubrication [9.1].

Tribology aims to optimize friction and wear for a particular application case. Apart from fulfilling the required function, this means assuring high efficiency and sufficient reliability at the lowest possible manufacturing, assembly, and maintenance costs.

Friction and wear are frequently undesirable. While friction impairs the efficiency of machine elements, machines, and plants and thus increases the energy demand, wear diminishes the value of components and assemblies and can lead to the failure of machines and plants. On the other hand, many technical applications strive for high friction, e.g., brakes, clutches, wheels/

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rails, car tires/road, friction gears, belt drives, bolted joints, and press fits. To a limited extent, wear can also be advantageous in special cases, e.g., in breaking-in processes.

Friction and wear are not properties specific to the geometry or substance of only one of the elements involved in friction and wear, e.g., external dimensions, surface roughnesses, thermal conductivity, hardness, yield point, density or structure, but rather are properties of a system. The system's friction and/or wear behavior can already change seriously when one influencing variable of the tribological system is marginally modified.

Lubrication is employed to lessen friction and minimize wear or to prevent them entirely. In the case of circulatory lubrication, the lubricant can additionally remove wear particles and heat from the friction contact. Other important tasks of lubrication are preventing corrosion (rusting) and, in the case of grease lubrication, sealing the friction points.

## 9.2 Tribological System (Tribosystem)

Friction and wear occur within a tribological system (TS). To delimit a TS, a system envelope is appropriately placed around the components and materials directly involved in friction and wear, thus virtually isolating these from the remaining components. The materials and components involved in friction and wear are the elements of the TS and are characterized by their material and shape properties. A tribological system is described by the function to be fulfilled, the input variables (operating variables), the output variables, the loss variables, and the structure (Fig. 9.1).

Apart from desired *input variables*, undesired input variables, so-called *disturbance variables*, also arise. Together with the structure, they influence the *output* and *loss variables* of the TS.

The function of a TS is to use the system structure to convert input variables (e.g., input torque, input speed, input type of motion, and sequence of motions) into technically utilizable output variables (e.g., output torque, output speed, output motion) (Fig. 9.1).

### 9.2.1 Structure

The elements involved, their properties, and the interactions between the elements describe the structure of a TS. The basic structure of every TS consists of four elements: the base body, counterbody, interfacial medium, and ambient medium (Fig. 9.1). Table 9.1 displays some TS with different elements. While the base body and counterbody are found in every TS, the interfacial medium and, in a vacuum, even the ambient medium can be absent.

In transport and machining processes, the base body is constantly stressed by new material zones of the counterbody. Such systems are called an *open* TS. By contrast, the stressed zones of the base body and counter-

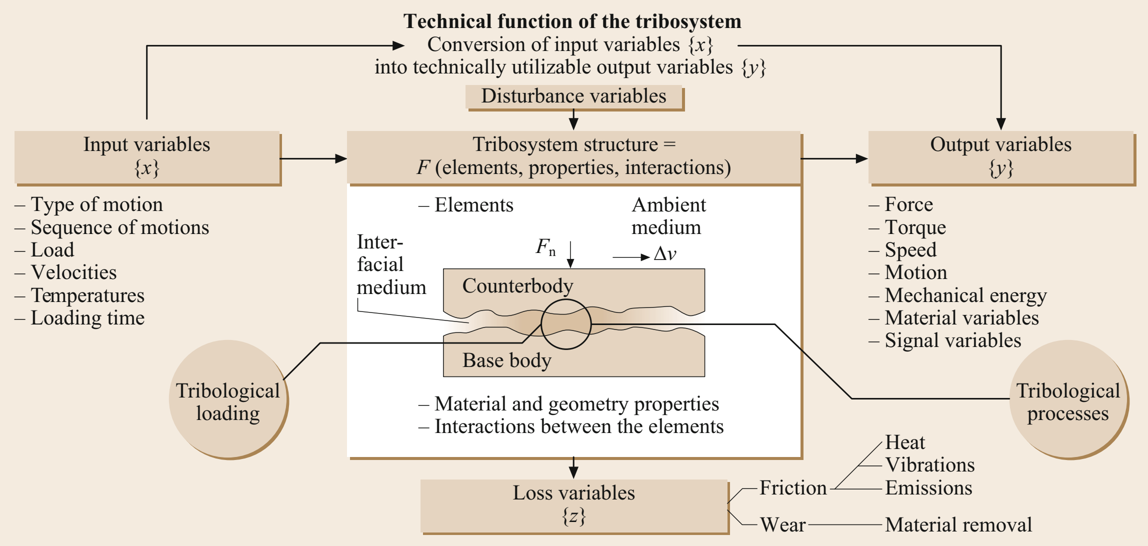


**Fig. 9.1** Expanded representation of a tribological system (after [9.2])

**Table 9.1** Examples of elements of tribological systems

| TS | Base body | Counterbody | Interfacial medium | Ambient medium | System type |
|---|---|---|---|---|---|
| Press and shrink joints | Shaft | Hub | – | Air | Closed |
| Sliding bearing | Journal | Bearing bush | Oil | Air | Closed |
| Mechanical face seal | Seal head | Seat | Liquid or gas | Air | Closed |
| Gear train | Pinion | Wheel | Gear oil | Air | Closed |
| Wheel/rail | Wheel | Rail | Moisture, dust, grease | Air | Open |
| Excavator bucket/ excavated material | Bucket | Excavated material | – | Air | Open |
| Turning tool | Cutting edge | Workpiece | Cutting lubricant | Air | Open |

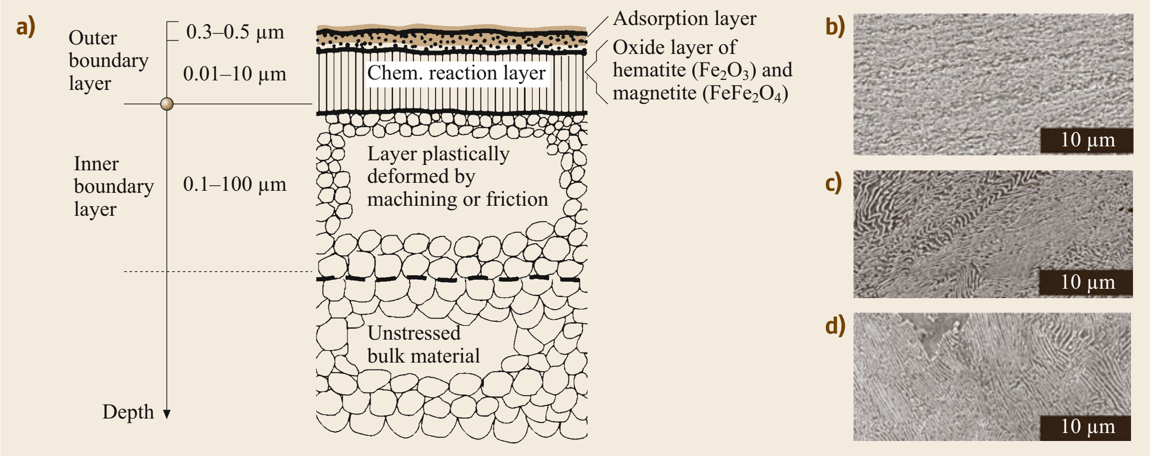


**Fig. 9.2a–d** Boundary-layer composition in metallic materials using a tribologically loaded steel rail: **(a)** cross-section diagram; **(b)** depth approximately 50 µm; **(c)** transition zone; **(d)** bulk material (after [9.3])

body in a *closed* TS are repeatedly in contact. Examples of open and closed systems can also be found in Table 9.1. The function in open systems mainly depends on the wear of the base body. The counterbody generates the load; as a rule, the wear on it is not of interest. By comparison, when systems are closed, the ability to operate depends on the wear of both friction bodies. The elements of the TS are characterized by a large number of *properties*, largely listed in Table 9.2.

A difference is made principally between geometric and material properties in the base body and counterbody, which are supplemented by physical variables. The interfacial medium and ambient medium can appear in different aggregate states, on which other important tribological properties depend. A difference in the material properties of the base body and counterbody is made between bulk material and the near-surface zone. The properties of the near-surface zone, e.g., structural composition, hardness, and chemical composition, are particularly important for the tribological processes. In addition, the surface roughness's play an important role.

Figure 9.2 shows a diagram of the possible composition of *boundary layers* in metallic materials. The undisturbed *basic structure* generally has attached to it a hardened composition of machined or deformed layers endowed with a structure that is fine-grained compared with the basic structure. A reaction layer and an adsorption layer lie on top of this. Taken together they are also called the *outer boundary layer*. Wear is generally acceptable as long as it occurs in the outer boundary layer.

As a rule, not only the near-surface zone's material structure but also its chemical composition differs from those of the bulk material. This is apparent in Fig. 9.3. The material beneath the surface already changes during manufacturing in terms of the concentration of elements present compared with the bulk material. Other considerable changes undergone by the near-surface zone's element concentrations are caused by breaking-in or occur after a short running time.

## 9.2.2 Contact Geometry

It is not only the input variables, the material properties and the physical variables of the base body and counterbody, the interfacial medium, and the ambient medium but also the contact areas that greatly influence friction

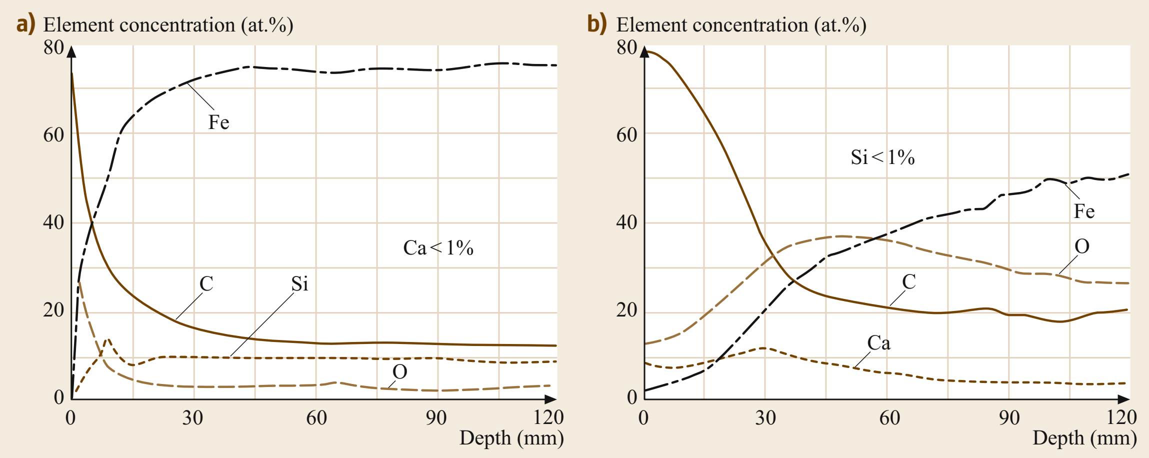


**Fig. 9.3a,b** Auger electron spectroscopy depth profiles as element concentration in the cylinder wall at top dead center in a diesel engine **(a)** before running in and **(b)** after 15 h running time (oil: SAE 15W-40) [9.4]

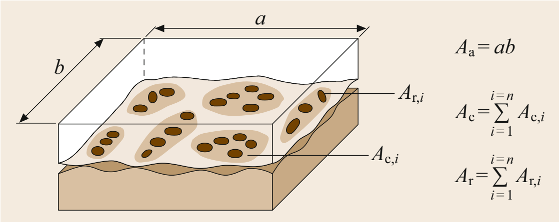


**Fig. 9.4** Various types of contact areas

and wear of the base body and counterbody and the tribological system's lubrication state.

In turn, the contact areas appearing during operation depend on the form of contact (Table 9.3), the input variables, the geometric properties of the base body and counterbody (Table 9.1), and the other system properties.

In the contact areas, a difference is made between the *nominal or apparent contact area* and the *real contact area* (Fig. 9.4).

The nominal or apparent contact area $A_a$ corresponds to the macroscopic contact area of the bodies in contact, e.g., of the contact area *ab* of a cuboid on one plane or the Hertzian contact area between a cylinder and a plane. Not only the apparent contact area but also the real contact areas $A_{r,i}$ play a key role when the asperities of the friction bodies meet in the apparent contact area.

The real contact areas result from the asperity contacts, which occur when the friction bodies are not completely separated by a lubricating film or in the cases of application in which no lubricant is used (dry friction). Not only the roughness but also the waviness on the friction body surfaces have to be taken into account when analyzing the real contact areas. The waviness causes so-called *contour areas* $A_{c,i}$ to form as well as the real contact areas on the asperity contacts within the contour areas.

As a rule, the sum of the real contact areas $A_r$, which is dependent on the roughness distributions and the separation of the two friction body surfaces, is substantially smaller than the apparent contact area ($A_r \approx 10^{-1}$–$10^{-4} A_a$). Hence the real contact pressures in the asperity contacts are substantially higher than the nominal pressure. While the nominal pressure displays elastic macromaterial behavior, plastic deformation may already have begun for a majority of the microcontacts (real contact areas). Results of calculations have yielded that the sum of the real contact areas is nearly proportional to the normal force $F_n$ [9.5]. In addition, as the normal force increases, the number of real individual contacts increases, while the real individual contact area $A_{r,i}$ remains roughly constant.

Apart from the real contact area, the *overlap ratio* $\varepsilon$ also plays an important role as well. It represents the ratio of the apparent contact area $A_a$ to a friction body's friction area $A_f$. Thus, for example, the nonrotating bush of a sliding bearing with bearing clearance has an overlap ratio of $\varepsilon = 1$, since the apparent contact area $A_a$ for the bearing bush corresponds to the friction area $A_f$. For the rotating shaft however, the friction area $A_f = \pi db$, with shaft diameter $d$ and bearing bush width $b$, is larger than the apparent contact area $A_a = db\gamma/2$, with contact angle $\gamma$, so that the overlap ratio is $\varepsilon < 1$. For a constantly loaded friction body, an overlap ratio $\varepsilon = 1$ means permanent contact, no cyclical mechanical loads (macroscopic), permanent frictional heat absorption, and limited microchemical reaction with the ambient medium. For the friction body concerned, an overlap ratio of $\varepsilon < 1$ leads to an intermittent contact, to cyclical mechanical loading, to intermittent frictional heat absorption, and to

**Table 9.2** Tribologically relevant properties of elements of the tribological system (Fig. 9.1)

| **1. Base body and counterpart** |
|---|
| 1.1 *Geometric properties* |
| ● External dimensions |
| ● Shape and position tolerances |
| ● Waviness |
| ● Surface roughnesses |
| 1.2 *Material properties* |
| 1.2.1 *Bulk material* |
| ● Strength |
| ● Hardness (macro, micro, and Martens hardness) |
| ● Structure, texture, microstructure phases (distribution, size, number type) |
| ● Young's modulus, Poisson's ratio |
| ● Residual stress |
| ● Chemical composition |
| 1.2.2 *Near-surface zone* |
| ● Hardness (macro, micro, and Martens hardness) |
| ● Surface energy |
| ● Metallurgical structures, texture, microstructure phases (distribution, size, number type) |
| ● Chemical composition |
| ● Young's modulus, Poisson's ratio |
| ● Residual stress |
| ● Boundary-layer thickness and structure |
| 1.3 *Physical variables* |
| ● Density |
| ● Heat conductivity |
| ● Coefficient of thermal expansion |
| ● Melting point |
| ● Specific thermal capacity |
| ● Hygroscopic properties |
| **2. Interfacial medium (lubricant)** |
| ● Aggregate state (solid, liquid, gaseous) |
| ● For *solid* interfacial medium |
|   – Hardness |
|   – Grain size distribution |
|   – Grain shape |
|   – Grain quantity, grain number |
|   – Number of components, mixing ratio |
|   – Chemical composition |
| ● For *liquid* interfacial medium |
|   – Viscosity depending on temperature, pressure and shear rate |
|   – Consistency |
|   – Wettability |
|   – Lubricant quantity and pressure |
|   – Chemical composition |
|   – Mixing ratio of components |
| **3. Ambient medium** |
| ● Aggregate state (solid, liquid, gaseous) |
| ● Heat conductivity |
| ● Chemical composition |
| ● Moisture |
| ● Ambient pressure |

tribochemical reactions with the ambient medium in the range $A_\mathrm{f}$–$A_\mathrm{a}$. When both friction bodies (the base body and counterbody) have an overlap ratio of $\varepsilon \approx 1$, the wear particles can remain in the contact area and hence adversely influence the further wear profile.

## 9.2.3 Tribological Loads and Interactions

Tribological loads in a TS are generated by the input and disturbance variables' action on the system structure. They chiefly include contact, kinematic, and thermal processes [9.2]. According to [9.1], the tribological load represents "the loading of the surface of a solid caused by contact and relative motion of a solid, liquid or gaseous counterbody." It is introduced via the real contact areas. Plastic deformation and wear can cause the real contact areas to change during TS operation.

When mechanical energy is converted by friction, energy dissipates, which makes itself noticeable by changing the thermal situation. Since the thermal behavior also continuously adapts to the new conditions as a result of wear, changes to the contact geometry, and resulting changes in the friction, dynamic rather than static influencing variables determine the tribological loading in a real contact.

The contact geometry, the processes occurring in the contact, and the thermal behavior of a TS are influenced by, among other things, the load, the motion conditions, the element properties, and the friction state.

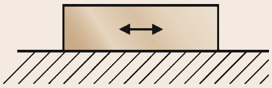
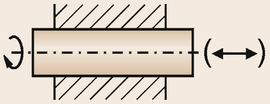
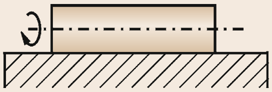
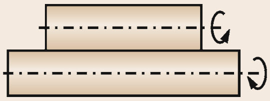
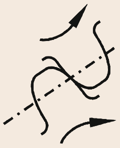
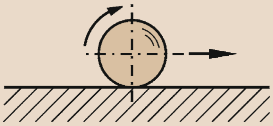
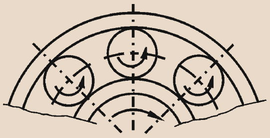
While the apparent contact area alone is decisive in fluid lubrication, according to [9.6], in mixed lubrication, i.e., when the dimensionless film thickness parameter

$$\Lambda = \frac{h_\mathrm{min}}{\left(R_\mathrm{q1}^2 + R_\mathrm{q2}^2\right)^{1/2}}\,, \tag{9.1}$$

with the minimum lubrication film thickness $h_\mathrm{min}$ and the root-mean-square (rms) surface roughnesses $R_\mathrm{q1}$ and $R_\mathrm{q2}$ of the base body and counterbody, is in the range $\Lambda < 3$, in boundary lubrication with $\Lambda < 1$ and for dry friction, both the apparent contact area and the real contact areas must be considered and allowed for (Fig. 9.4).

When there are contacts between the friction bodies, *interactions* occur in the real contact areas and in the near-surface zones. *Atomic/molecular* interactions occur on the one hand and *mechanical* interactions on the other. Whereas the former cause adhesion on solid–solid boundary layers or are extremely important technically in the form of physisorption and chemisorption on solid–fluid boundary layers, the latter lead to elastic and plastic contact deformations and to the development of the real contact areas.

**Table 9.3** Contact geometry (form of contact) of tribological systems

| Form of contact | Base body | Counterbody | Sketch | Example application |
|---|---|---|---|---|
| **Conformal** | | | | |
| Areal | Plane | Plane |  | Thrust bearings |
| | Hollow cylinder | Solid cylinder |  | Sliding bearings, round fittings, cylinder races |
| **Nonconformal** | | | | |
| Line contact | Plane | Cylinder |  | Thrust roller bearings |
| | Cylinder | Cylinder |  | Roller, roller bearings |
| | Pinion tooth | Gear tooth |  | Gears |
| Point contact | Plane | Ball |  | Thrust ball bearings |
| | Inner ring (circumferential direction) | Ball |  | Ball bearings |

The type of interaction that primarily occurs depends greatly on the friction state. Thus, when a lubricant is present the atomic/molecular interaction can be disregarded more often than the mechanical.

Friction and wear in a given TS ultimately depend on the interactions between the elements. The friction state, the effective mechanisms of friction and wear, and the contact state can be used to describe the interactions.

The tribological loads occurring in the real contact areas produce *tribological processes*. These subsume the dynamic physical and chemical mechanisms of friction and wear and boundary-layer processes that can be attributed to friction and wear.

## 9.2.4 Operating Variables (Input Variables)

According to [9.1], the operating variables are: the type of motion, the time sequence of motions of the elements contained in the system structure, and a number of technical-physical load parameters, which act on the system structure when the function is executed. The operating variables originate from:

- Type of motion and time sequence of motions
- Load
- Velocities
- Temperatures
- Loading time.

The type of motion can frequently be attributed to one of the basic types of motion *sliding*, *rolling*, *spin*, *impact* or *flowing* or can be composed from these. The time sequence of motions can occur regularly, irregularly, back and forth, or intermittently. The sequence of motions frequently also consists of different components. As a rule, the normal force $F_n$ is decisive for the load.

Both the relative velocity between the friction bodies and the entraining velocity of the lubricant in the contact and the slippage as a ratio of the relative velocity to the average circumferential velocity play a role for the velocities. The friction body temperatures and the effective contact temperature produced in operation are critically important for the temperature variables. It is normally not possible to measure the contact temperatures. Apart from these desired input variables, which as a rule are specified by a technical function, *disturbance variables* such as vibrations or dust particles must be considered under certain circumstances.

### 9.2.5 Output Variables (Useful Variables)

The TS provides output variables for subsequent utilization. These useful variables reflect the performance of a function of the TS. The useful variables can differ over extremely wide ranges depending on the main task of the TS. In an energy-determined system, for example, the following output variables may be desired:

- Force
- Torque
- Velocity
- Motion
- Mechanical energy.

Particular material or signal variables could be interesting as useful variables in a material- or signal-determined TS.

### 9.2.6 Loss Variables

The loss variables of a TS are essentially represented by friction and wear. While friction leads to losses of force, torque or energy, wear means a progressive loss of material.

The energy losses produced when there is friction are converted into heat for the most part. This process is irreversible and is called energy dissipation. Along with the conversion of friction into heat and the generation of wear particles, the tribological process generates other tribologically induced loss variables such as vibrations that frequently become apparent through sound waves, photon emission (triboluminescence), electron, ion emission, etc.

## 9.3 Friction

Friction can be ascribed to the interactions between bodies' material zones that are in contact or moving relative to one another; it counteracts relative motion. External and internal friction are differentiated. When friction is external, the different friction bodies' material zones are in contact; when friction is internal, material zones that are in contact belong to one friction body or the interfacial medium.

A number of parameters can characterize friction. Thus, depending on the application, friction is characterized by the friction force $F_\mathrm{f}$, the friction torque $M_\mathrm{f}$ or the coefficient of friction $f$. Instead of $f$ the symbol $\mu$ is also frequently used for the coefficient of friction. The coefficient of friction $f$ is formed from the ratio of the friction force $F_\mathrm{f}$ to the normal force $F_\mathrm{n}$

$$f = \frac{F_\mathrm{f}}{F_\mathrm{n}} \,. \tag{9.2}$$

The work of friction or friction energy $W_\mathrm{f}$ is used to calculate the frictional heat or the amount of deformation of the friction force in solid friction. It is calculated as

$$W_\mathrm{f} = F_\mathrm{f} s_\mathrm{f} \,, \tag{9.3}$$

with the friction distance $s_\mathrm{f}$. The friction power $P_\mathrm{f}$ is of interest for an energy balance or efficiency calculation. The friction power is a power loss and, disregarding signs, the following applies

$$P_\mathrm{f} = F_\mathrm{f} \Delta v \,, \tag{9.4}$$

with the relative velocity $\Delta v$. (The power loss is frequently defined negatively.)

### 9.3.1 Types of Friction

Friction can be classified according to various features. Types of friction are distinguished depending on the type of relative motion between the friction bodies. Figure 9.5 presents the most important types of friction with sample applications. There are three main types of friction:

- Sliding friction
- Rolling friction
- Spin friction.

Apart from these three kinematically defined types of friction, there can be overlaps (mixed forms), namely:

- Sliding-rolling friction (rolling friction)
- Sliding-spin friction
- Rolling-spin friction.

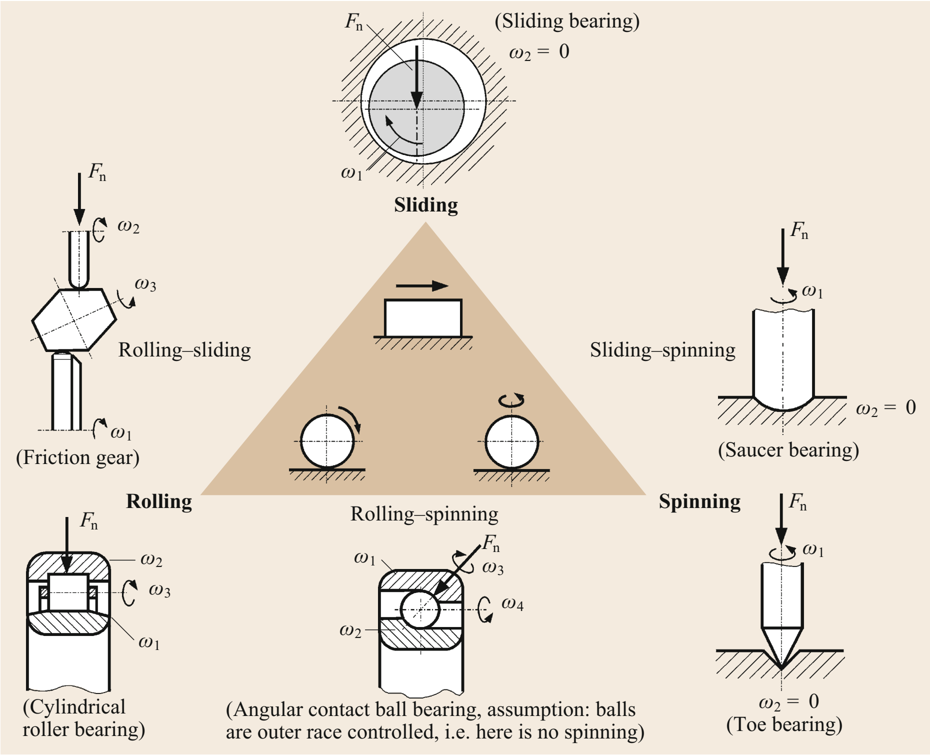


**Fig. 9.5** Types of dynamic friction

Along with the types of friction shown in Fig. 9.5, another type of friction is impact friction, which applies when a body strikes another body perpendicular or oblique to the contact surface and possibly withdraws again. The angular contact ball bearing is a machine element in which sliding and rolling and spin friction appear.

## 9.3.2 Friction States

Various friction states can be defined if friction is classified as a function of the aggregate state of the material zones involved. To illustrate this, Fig. 9.6 presents different states of friction based on the Stribeck curve using a radial sliding bearing as an example. Generally, the following friction states are differentiated:

- Solid friction
- Fluid friction
- Gas friction
- Mixed friction.

In *solid friction* the friction acts between material zones that exhibit solid properties and are in direct contact. If the friction occurs between solid boundary layers with modified properties compared with the bulk material, e.g., between reaction layers, then this is *boundary-layer friction*. If the boundary layers on the contact surfaces each consist of a molecular film coming from a lubricant, then this is called *boundary friction*. In boundary friction, the lubricant's hydrodynamic effect can be disregarded because the velocity is very low and/or only a very small quantity of lubricant, insufficient to fill the lubrication gap, is present.

*Fluid friction* is internal friction in a lubricating film with fluid properties and is present in a hydrodynamically, elastohydrodynamically or hydrostatically produced lubricating film.

*Gas friction* is internal friction in a lubricating film with gas properties and is present in an aerodynamically or aerostatically produced lubricating film.

*Mixed friction* is any mixed form of the friction states. The term is primarily used in the simultaneous presence of boundary and liquid friction.

## 9.3.3 Friction Mechanisms

Solid friction can be ascribed to interactions between the elements. As already addressed before, there are essentially two different types of interaction, i.e., atomic/molecular and mechanical. *Kragelski* [9.7] speaks of friction's "dual nature". Hence, the friction mechanisms can be divided into two groups. Generally a difference can initially be made between the following four fric-

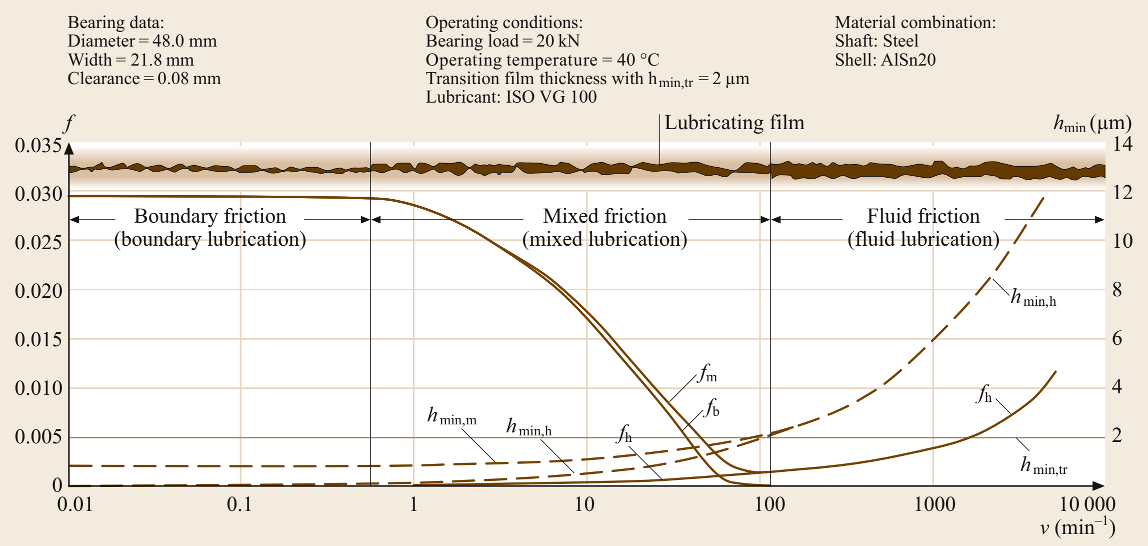


**Fig. 9.6** Stribeck curve, minimum lubrication film thickness, and friction states in a radial sliding bearing ($f_m$ friction coefficient in mixed friction, $f_b$ friction coefficient in boundary friction, $f_h$ friction coefficient in fluid friction, $h_{min,tr}$ minimum lubrication film thickness during the transition from fluid to mixed friction, $h_{min,h}$ minimum lubrication film thickness during fluid friction, $h_{min,m}$ minimum lubrication film thickness during mixed friction)

tion mechanisms, compiled schematically in Fig. 9.7:

- Adhesive bond shearing
- Plastic deformation
- Abrasion
- Hysteresis losses in elastic deformation.

Adhesion is an atomically/molecularly based friction mechanism. Its frictional effect is based on the bonds formed atomically or molecularly in the real contact areas that separate again when relative motion occurs, as a result of which energy loss occurs.

Deformation, abrasion, and hysteresis can be classified as mechanically based friction mechanisms. The action of friction in deformation and abrasion can above all be ascribed to the displacement of overlaps of asperities. Hysteresis is based on internal friction and has a damping effect. Different friction mechanisms frequently appear at the same time. The friction mechanisms that are principally acting depends on the state of friction.

## 9.3.4 Coefficient of Friction

Table 9.4 reproduces ranges of friction coefficients in various types and states of friction [9.8]. It should however be recalled that friction does not represent a constant property of a material or a combination of material properties but rather depends on the operating variables and the system structure, i.e., on the load and the elements involved in the friction process with their properties and interactions.

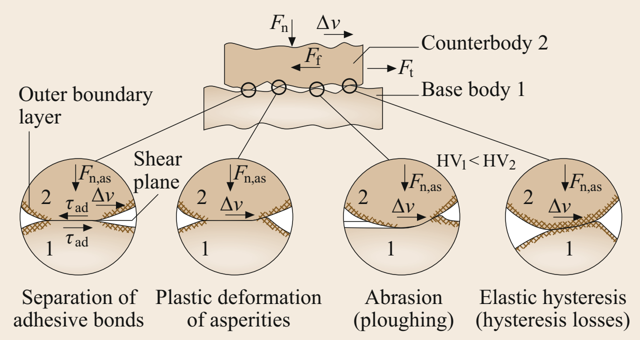


**Fig. 9.7** Basic friction mechanisms viewed microscopically ($F_n$ normal force on apparent contact area, $F_f$ friction force between base body and counterbody, $F_t$ tangential force, $F_{n,as}$ normal force on asperity contact, $\Delta v$ relative velocity, $\tau_{ad}$ shear stress for shearing an adhesive bond, HV Vickers hardness)

**Table 9.4** Coefficients of friction for different types and states of friction

| Type of friction | Friction state | Coefficient of friction $f$ |
|---|---|---|
| Sliding friction | Solid friction | 0.1–1 |
| | Boundary friction | 0.1–0.2 |
| | Mixed friction | 0.01–0.1 |
| | Fluid friction | 0.001–0.01 |
| | Gas friction | 0.0001 |
| Rolling friction | (Grease lubrication) | 0.001–0.005 |

## 9.4 Wear

As soon as the base body and counterbody come into contact, i.e., when the lubrication film thickness becomes too small or lubricant is unavailable, wear occurs. Wear is a progressive loss of material from the surface of a solid, brought about by mechanical causes, i.e., by contact and relative motion of a solid, fluid or gaseous counterbody [9.1]. Signs of wear are small detached wear particles, material removal from one friction body to the other, and material and shape changes of the tribologically loaded material zone of one or both friction partners.

### 9.4.1 Types and Mechanisms of Wear

Wear processes can be classified into different types according to the type of tribological load and the materials involved, e.g., sliding wear, fretting wear, abrasive wear, and material cavitation. Wear is caused by a number of mechanisms, the following four being especially important:

- Surface fatigue
- Abrasion
- Adhesion
- Tribochemical reaction.

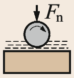
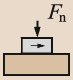
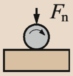
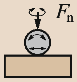
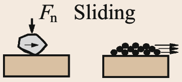
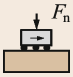
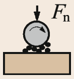
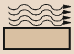
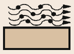
Table 9.5 provides a breakdown of wear according to types of wear and wear mechanisms based on [9.1].

Figure 9.8 presents a chart of the effective wear mechanisms. The wear mechanisms can occur individually, successively or concomitantly.

Surface fatigue manifests itself through cracking, crack growth, and detachment of wear particles, brought about by alternating loads in near-surface zones of the base body and counterbody.

In *abrasion* microcuttings, fatigue due to repeated ploughing, and fracture of the base body caused by the

**Table 9.5** Types and mechanisms of wear (after [9.1])

| **System structure** | **Tribological load (types of motion and simplified symbols)** | **Type of wear** | **Effective wear mechanism** | | | |
|---|---|---|---|---|---|---|
| | | | **Surface fatigue** | **Abrasion** | **Adhesion** | **Tribochemical reactions** |
| Solid<br>● Interfacial medium (complete solid separation)<br>● Solid | Sliding<br>Rolling<br>Bouncing<br>Impacting  | – | × | | | × |
| Solid<br>● Solid (solid friction, boundary lubrication, mixed lubrication) | Sliding  | Sliding wear | × | × | × | × |
| | Rolling<br>Revolving  | Rolling wear | × | × | × | × |
| | Oscillating  | Fretting wear | × | × | × | × |
| Solid<br>● Solid and particles | Sliding  | Sliding abrasion (three-body abrasion) | | × | | × |
| | Sliding  | Sliding abrasion (three-body abrasion) | × | × | | × |
| | Rolling  | Rolling abrasion (three-body abrasion) | × | × | | × |
| Solid<br>● Fluid | Flowing<br>Vibrating  | Material cavitation (cavitation erosion) | × | | | × |
| Solid<br>● Fluid and particles | Flowing  | Particle erosion (erosion wear) | × | × | | × |

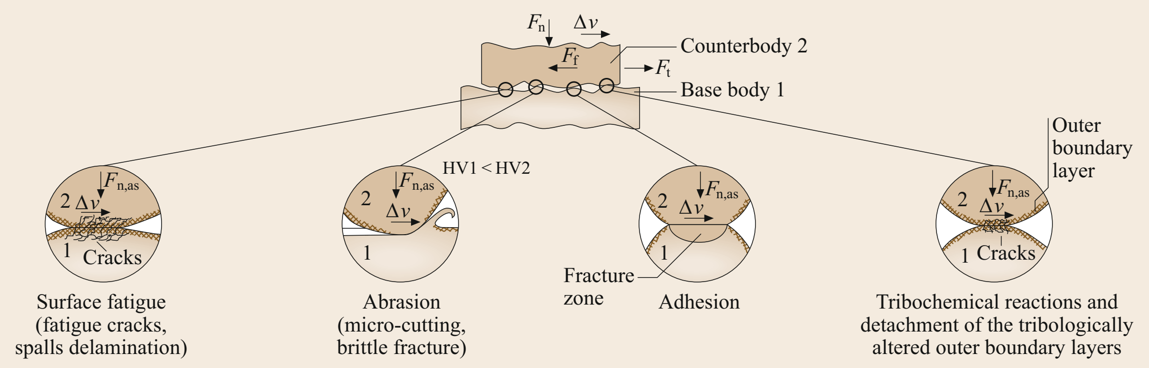


**Fig. 9.8** Basic wear mechanisms viewed microscopically ($F_n$ normal force on apparent contact surface, $F_f$ friction force between base body and counterbody, $F_{n,as}$ normal force on asperity contact, $\Delta v$ relative velocity, HV Vickers hardness)

**Table 9.6** Typical wear phenomena caused by the main wear mechanisms (after [9.1])

| Wear mechanism | Wear phenomenon |
|---|---|
| Adhesion | Scuffing or galling areas, holes, plastic shearings, material transfer |
| Abrasion | Scratches, grooves, ripples |
| Surface fatigue | Cracks, pitting |
| Tribochemical reactions | Reaction products (layers, particles) |

counterbody's hard asperities or by hard particles in the interfacial medium lead to wear.

In *adhesion*, after possibly extant protective surface layers have been broken through, atomic bonds (microwelds) form above all on the plastically deformed microcontacts between the base body and counterbody. If the strength of the adhesive bonds is greater than that of the softer friction partner, material eventually detaches from the deformed surface of the softer friction partner and is transferred to the harder one. The transferred material can either remain on the harder friction partner or detach, or even return.

In *tribochemical reactions*, friction-induced activation of loaded near-surface zones causes elements of the base body and/or counterbody to react chemically with elements of the lubricant or ambient medium. Compared with the base body and counterbody, the reaction products exhibit changed properties and, after reaching a certain thickness, can be subject to brittle chipping or even exhibit properties reducing friction and/or wear.

Apart from the types and mechanisms of wear, *wear phenomena* are also extremely interesting for interpreting the result of wear (Table 9.6). These mean the changes of a body's surface layer resulting from wear and the type and shape of the wear particles accumulating. Light or scanning electron microscope images can present this extremely clearly.

## 9.4.2 Wear Profiles and Measurable Variables

Estimating the service life of components necessitates knowing the wear profile over the loading time and/or the rate of wear (amount of wear per loading time). According to [9.1, 8], different wear profiles are frequently generated depending on the effective wear mechanism (Fig. 9.9).

Three phases are distinguished: break-in, steady state, and failure. During breaking-in, increased wear, so-called break-in wear, with a degressive profile can occur and, for example, switch over into a long-lasting state with a constant increase of the amount of wear (constant rate of wear) until failure is announced by a progressive increase of wear (Fig. 9.9a).

If surface fatigue takes effect as the primary mechanism of wear, then measurable wear after break-in frequently only becomes noticeable after a certain incubation period during which microstructural changes, cracking, and crack growth commence. Wear particles only detach after the incubation period (Fig. 9.9b). A negative amount of wear is even occasionally measured at the beginning of the wear process. This is caused by material transfer from the wear partner (Fig. 9.9c).

In principle, the profile of the amount of wear $W$ over the loading time $t$ can be progressive, linear or digressive (Fig. 9.9d).

A difference is made between directly measurable wear variables, such as linear, planimetric, volumetric and mass amount of wear, and related measurable wear variables (wear rates), such as rate of wear, wear–distance ratio, and wear–throughput ratio. As a rule, the amount of wear can be measured. It is advisable to use specific or relative amounts of wear in comparative wear tests whenever the operating variables or the

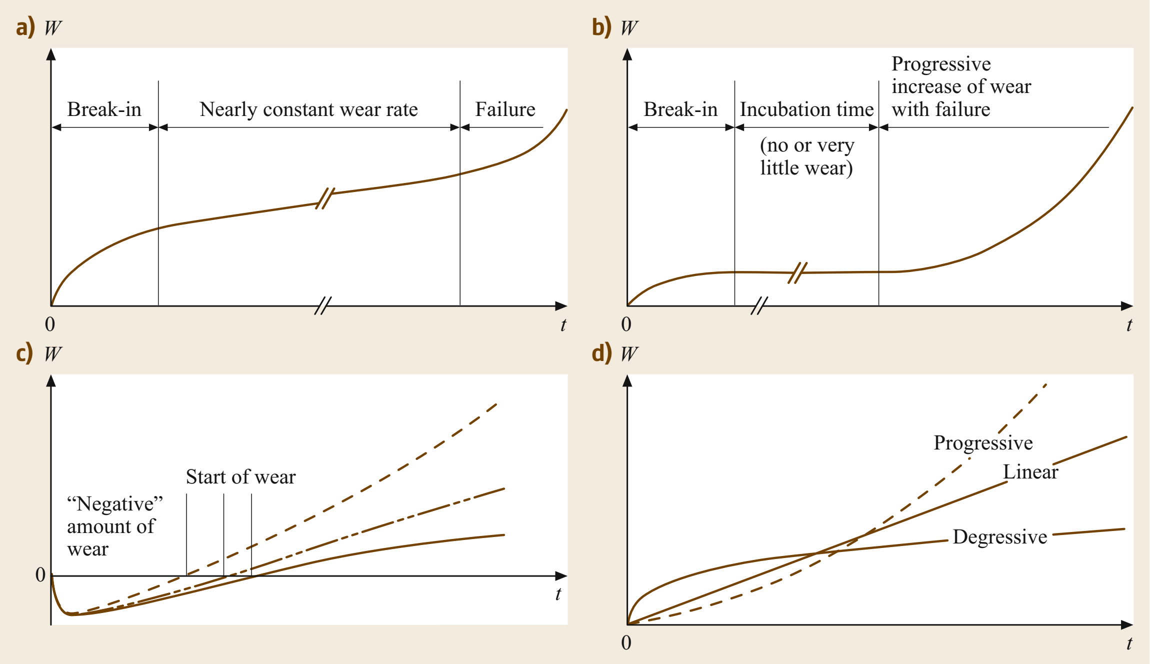


**Fig. 9.9a–d** Amount of wear as a function of loading time: **(a)** typical wear profile with a nearly constant wear rate; **(b)** wear profile with an incubation time; **(c)** wear profile with negative amount of wear; **(d)** different wear profiles

properties of the elements involved in the wear cannot be kept constant or are intentionally changed.

### 9.4.3 Determination of Wear and Lifetime

In unlubricated tribological systems and in systems operated in the boundary or mixed lubrication range, practically usable information about friction and wear can frequently only be found through experimentation since the physical and chemical mechanisms occurring here frequently cannot be described sufficiently exactly theoretically so that a precise calculation of friction and wear is often impossible.

However, mathematical equations for an approximate preliminary determination of friction and wear provide the energy balance of the friction process. If friction is approached as an energetic problem, the energy balance forms the basis for determining the friction and wear [9.9]. The friction force $F_\mathrm{f}$ and the friction coefficient $f$ can be determined from the friction energy $W_\mathrm{f}$

$$\begin{aligned} W_\mathrm{f} &= W_\mathrm{el,hys} + W_\mathrm{pl} + W_\mathrm{abr} + W_\mathrm{ad} \\ &= fF_\mathrm{n}s_\mathrm{f} = fF_\mathrm{n}\Delta vt\,, \end{aligned} \qquad (9.5)$$

where $W_\mathrm{el,hys}$ is the proportion of the work of friction from the hysteresis during elastic deformation, $W_\mathrm{pl}$ is the work of friction from plastic deformation, $W_\mathrm{abr}$ is the work of friction caused by abrasion, $W_\mathrm{ad}$ is the work of friction caused by separation of adhesive bonds, $f$ is the friction coefficient, $F_\mathrm{n}$ is the normal force, $s_\mathrm{f}$ is the friction distance, $\Delta v$ is the relative velocity between the friction bodies, and $t$ is the friction time [9.9].

The action time of a friction pair usually depends on the wear. The anticipated lifetime $L_\mathrm{h}$ under specific operating conditions is frequently determined by calculation. A function-based maximum allowable wear is assumed. Thus, the lifetime $L_\mathrm{h}$ corresponds to the action time for which the friction pair can be safely operated.

Wear properties as a function of the friction time are presented in Fig. 9.10. The development of wear with its scatter as a function of the friction time is shown for various operating conditions. The wear height $h_\mathrm{w}$ serves as the gauge of wear. The function-based wear limit is characterized by the maximum allowable wear height $h_\mathrm{w,al}$. As a rule, a wear profile exists, which is composed of a break-in process and a steady state (Fig. 9.10, cases 2 and 3). In addition, Fig. 9.10 also shows the distribution densities $f(t)$ for the lifetimes $L_\mathrm{h}$ defined by $h_\mathrm{w,al}$.

Despite a large allowable wear height, case 1 is not practicable since the friction pair here fails early and

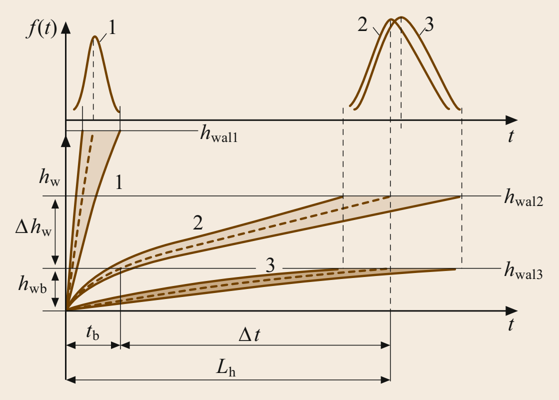


**Fig. 9.10** Courses of wear height $h_\mathrm{w}$ and distribution density $f(t)$ as a function of the friction time $t$ when allowable wear height values have been reached for different high stresses and different frictional states (after [9.10])

does not reach a steady state. Case 2 shows a normal progression in pure solid friction. More favorable conditions exist in case 3 (e.g., mixed friction), but the allowable wear height is small.

The average lifetime (action time) for the median of the distribution density results from the expression

$$L_\mathrm{h} = t_\mathrm{b} + \Delta t = t_\mathrm{b} + \frac{h_\mathrm{w,al} - h_\mathrm{w,b}}{I_\mathrm{h} \Delta v}\,, \tag{9.6}$$

with the linear wear intensity $I_\mathrm{h}$ formed from the ratio of the wear height $h_\mathrm{w}$ to the friction distance $s_\mathrm{f}$ ($I_\mathrm{h} = \Delta h_\mathrm{w}/\Delta s_\mathrm{f}$), the break-in time $t_\mathrm{b}$, and the break-in wear height $h_\mathrm{w,b}$. If it is assumed that the wear volume $V_\mathrm{w}$ and the work of friction $W_\mathrm{f}$ are proportional ($W_\mathrm{f} \propto V_\mathrm{w}$), and the value $e_\mathrm{w}$ as a proportionality factor is introduced as the wear energy density, as a result of which the basic wear equation becomes $W_\mathrm{f} = e_\mathrm{w} V_\mathrm{w}$, and if the probabilistic nature of real operation is allowed for (through the introduction of $L(\gamma)_\mathrm{h}$ as $\gamma$ percentage lifetime, $x$ as the quantile (random variable) of the standardized normal distribution, and $v$ as an empirical coefficient of variation that represents a measure of the scatter of the wear rate), then the following expression can be set up for the lifetime [9.10]

$$L(\gamma)_{\mathrm{h}1,2} = t_{\mathrm{b}1,2} + \frac{(h_{\mathrm{w,al}1,2} - h_{\mathrm{w,b}1,2})\, e_{\mathrm{w}1,2}}{\alpha_{\mathrm{f}1,2}\, \tau_\mathrm{f} \Delta v\, (1 - x v_{1,2})}\,, \tag{9.7}$$

with the friction shear stress $\tau_\mathrm{f} = f F_\mathrm{n}/A_\mathrm{a}$, where $A_\mathrm{a}$ stands for the apparent contact area, and the energy proportion factor $\alpha_{\mathrm{f}1,2}$ is used to quantify the share of the work of friction induced in friction body 1 or 2.

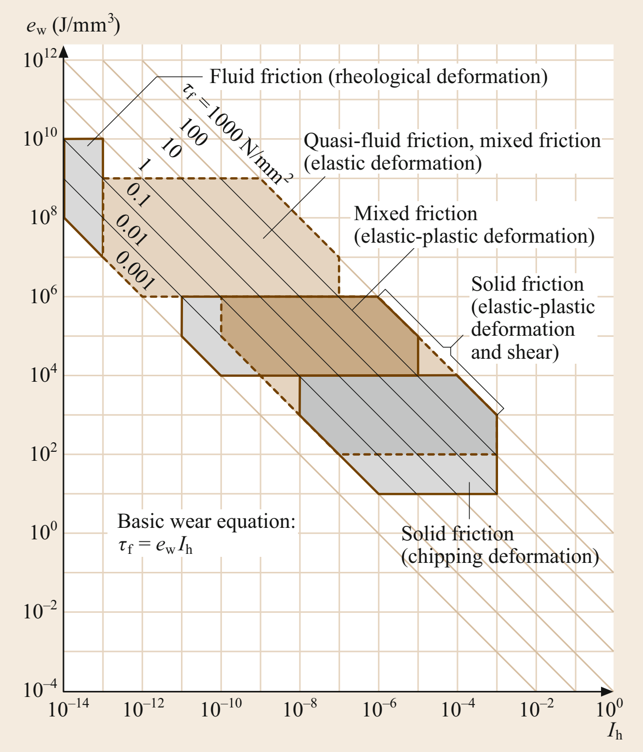


**Fig. 9.11** Wear energy density $e_\mathrm{w}$ as a function of the linear intensity of wear $I_\mathrm{h}$ and the friction shear stress $\tau_\mathrm{f}$ (after [9.9])

For a survival probability of $\gamma = 50\%$, when there is a standardized normal distribution, the quantile is $x = 0$, whereas $x = -1.28$ applies for $\gamma = 90\%$. The coefficient of variation depends on the operating conditions (common values based on [9.11] are $v = 0.2{-}0.8$; when there is fatigue wear, $v = 0.2{-}0.4$).

Determining the lifetime by using (9.7) not only requires knowledge of the break-in time $t_\mathrm{b}$, the allowable wear height $h_\mathrm{w,al}$, the break-in wear height $h_\mathrm{w,b}$, the energy proportion factor $\alpha_\mathrm{f}$, the normal force $F_\mathrm{n}$, the apparent contact surface $A_\mathrm{a}$, the relative velocity $\Delta v$, and the statistical variables quantile $x$ and coefficient of variation $v$ but above all also information about the friction coefficient $f$ and the wear energy density $e_\mathrm{w}$ appearing in the contact.

The friction coefficient can be determined experimentally or estimated through calculations [9.3, 12]. Values for the wear energy density $e_\mathrm{w}$ are either determined in tests or taken out of [9.9–11]. Figure 9.11 presents the nomogram of the basic wear equation $W_\mathrm{f} = e_\mathrm{w} V_\mathrm{w}$, with ranges for typical friction and wear states for evaluating and classifying the tribological behavior of friction pairs.

## 9.5 Fundamentals of Lubrication

Lubrication completely or partially separates the surfaces of the friction bodies by selectively introducing an interfacial medium (lubricant) that minimizes friction and wear. Most lubricants are fluids (mineral oils, synthetic oils, water, etc.), yet they can also be solid, e.g., for use in dry sliding bearings (polytetrafluoroethylene (PTFE), graphite, molybdenum disulfide ($MoS_2$), etc.). Greases are also applied, e.g., in ball bearings and sliding bearings and occasionally in gears too. Gases (air) are also employed, e.g., in gas-lubricated bearings. Figure 9.12 reproduces the areas of application of different lubricants. Powders, lubricating varnishes, and pastes can be classified as solid lubricants [9.13].

Similar to friction states, different lubrication states can also be defined (Table 9.7), to be precise: fluid film lubrication, mixed lubrication, boundary lubrication, and solid lubrication (lubrication with solid lubricants and surface coatings).

### 9.5.1 Fluid Film Lubrication

Fluid film lubrication, i.e., the complete separation of the frictional surfaces by a lubricating film, can be achieved by hydrodynamic, elastohydrodynamic or hydrostatic lubrication (Table 9.8). A load-dependent pressure is developed in the lubricating film, as a result of which the load can be carried.

### 9.5.2 Hydrodynamic Lubrication

The following conditions must be met to produce hydrodynamic lubrication. A viscous lubricant that adheres to both the moving and the fixed friction body must be used (adhesive effect of the lubricant). Furthermore, the friction body surfaces must be converging and the lubricant must be dragged into the converging gap. The entraining velocity depends on the velocity at which the lubricant is dragged into the contact and is frequently confused with the relative velocity. The latter is decisive for friction, while the entraining velocity is essential for load-carrying capacity.

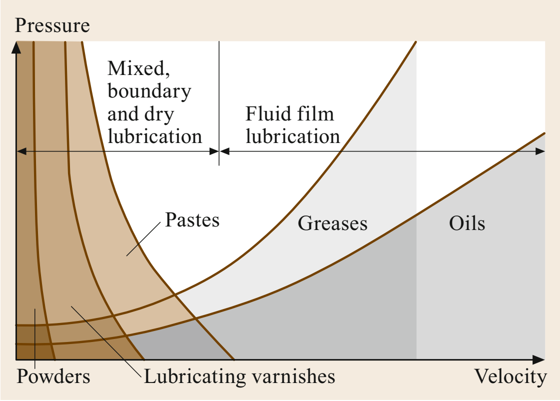


**Fig. 9.12** Areas of lubricant application (after [9.13])

The lubricant adhering to the surface is dragged into the lubrication gap by the friction bodies moving relative to the minimum lubrication film thickness. Since the gap converges in the direction of motion, the quantity of oil dragged by the friction bodies accumulates in front of the narrowest gap cross section and cannot be completely transported by the moving friction bodies' drag effect alone through the minimal cross section.

Thus, overpressure is inevitably produced in the converging zone in between the sliding surfaces, which grows to be just large enough that the difference between the quantities supplied and discharged by the drag flow are squeezed out of the lubricant gap as a result of compressive flows (satisfaction of the condition of continuity). The overpressure produces the friction contact's load-carrying capacity in equilibrium with the external load.

Figure 9.13 presents examples of velocity distributions at various points in the lubrication gap and the pressure distribution for an inclined plane bearing pad. The level of the average pressure developed (normally $< 7\,\mathrm{MPa}$) is usually not high enough to cause significant elastic deformation of the surfaces. The

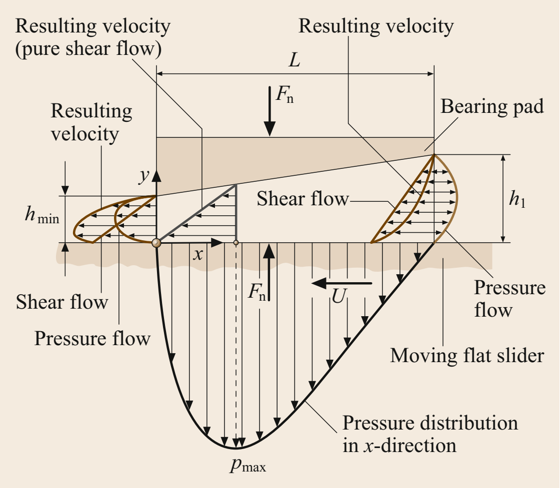
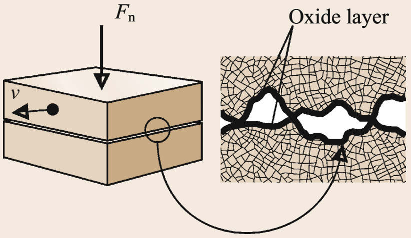
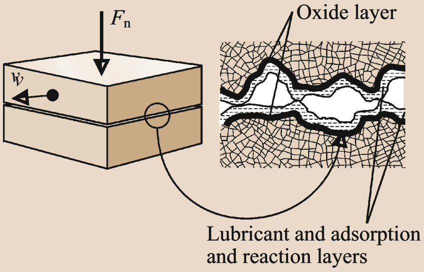
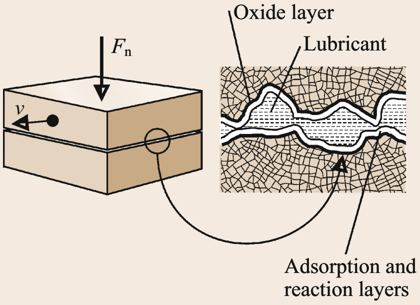
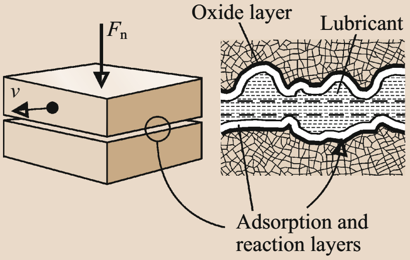


**Fig. 9.13** Velocity distributions and pressure distribution in a bearing pad

**Table 9.7** Friction and lubrication states

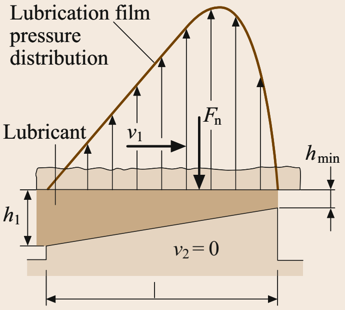
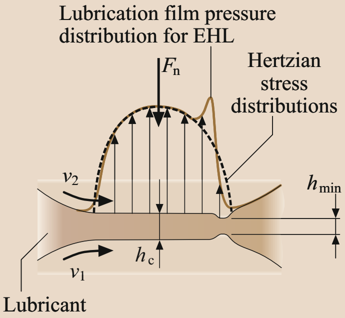
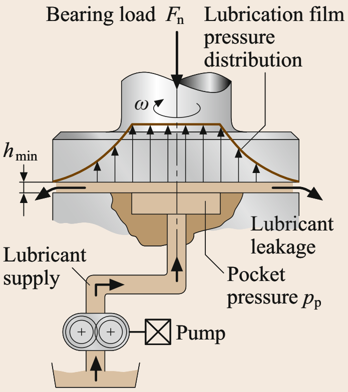
| | **Friction/lubrication states** |
|---|---|
|  | 1. Solid friction/no lubrication<br>● Direct contact of friction partners<br>● Formation of oxide reaction layers and adsorption of gases<br>● High wear rates probable, risk of seizing<br>● Friction coefficients (reference values): $f \approx 0.35$ to $> 1$<br>● Special cases<br>*Boundary-layer friction*<br>Friction between solid boundary layers with modified properties compared with the bulk material (e.g., oxide layers)<br>*Friction of pure metallic surface*<br>Direct contact of pure metallic surfaces (e.g., in machine or scuffing processes) |
|  | 2. Boundary friction/boundary lubrication<br>● Surfaces of the friction partners covered with a thin friction minimizing lubrication film<br>● Physisorption, chemisorption, and tribochemical reaction with additives from the lubricant form friction-minimizing, lightly shearing layers on the surfaces<br>● Lower wear rates than for solid friction<br>● Friction coefficients (reference values): $f \approx 6\times10^{-2}$–$2\times10^{-1}$ |
|  | 3. Mixed friction/mixed lubrication<br>● Lubrication film not thick enough to separate surfaces from each other completely; consequence: roughness contacts<br>● Load is partially carried by lubricating film through hydrodynamic effect and partly by the roughness contacts<br>● As in boundary friction, use of additives in lubricant is also important to generate friction-minimizing absorption and reaction layers on the surfaces<br>● Wear rates are smaller, the larger the hydrodynamic part of the load-carrying capacity<br>● Friction coefficients (reference values): $f \approx 10^{-3}$–$10^{-1}$ |
|  | 4. Fluid friction/fluid film lubrication<br>● Friction partners are completely separated from each other by a fluid film that can be generated, hydrodynamically or hydrostatically<br>● Virtually wear-free operation<br>● Friction coefficients (reference values): $f \approx 6\times10^{-4}$–$5\times10^{-3}$ |

hydrodynamic pressure is dependent on the friction bodies' geometry, the inclination of the surfaces to each other, the viscosity of the lubricant, the load, and the velocity of the lubricant dragged into the converging lubrication gap.

Table 9.8 specifies the correlation between the minimum lubrication film thickness $h_{\min}$ and the entraining velocity, the viscosity of the lubricant, and the load. It is evident that the minimum lubrication film thickness increases as the entraining velocity increases and the viscosity becomes greater, and that it decreases as the load grows, although only with an exponent of 0.5 in each case. The minimum lubrication film thickness is normally greater than 1 µm.

Above all, it is the lubricant's viscosity that determines the lubricating effect in hydrodynamic lubri-

**Table 9.8** Different types of fluid film lubrication

| | |
|---|---|
|  | **Hydrodynamic** [9.14]<br>For a given geometry<br>$h_{min} \approx \sqrt{\frac{\overline{v}\eta}{F_n}} = (\overline{v}\eta)^{0.5} F_n^{-0.5}$,<br>$\overline{v} \; = \frac{v_1+v_2}{2}$ entraining velocity,<br>$\eta$ average viscosity of the lubricant in the lubricating film,<br>$F_n$ load |
|  | **Elastohydrodynamics** [9.6]<br>a) *Hard material surfaces (hard EHL)*<br>Elliptical contact surfaces, given geometry<br>$h_{min} \approx (\overline{v}\eta_0)^{0.68} \alpha^{0.49} E^{*-0.117} F_n^{-0.073}$,<br>$\frac{h_{min}}{h_c} \approx 0.56$<br>$\overline{v} \; = \frac{v_1+v_2}{2}$ entraining velocity,<br>$\eta_0$ viscosity of the lubricant at gap entry at $p = 0$,<br>$\alpha$ viscosity–pressure coefficient,<br>$\frac{1}{E^*} = \frac{1}{2}\left(\frac{1-v_1^2}{E_1} + \frac{1-v_2^2}{E_2}\right)$ reduced modulus of elasticity,<br>$E_1$ and $E_2$ Young's modulus of friction bodies 1 and 2,<br>$v_1$ and $v_2$ Poisson's ratio of friction bodies 1 and 2,<br>$F_n$ load<br>b) *Soft material surfaces (soft EHL)*<br>Elliptical contact surfaces, given geometry<br>$h_{min} \approx (\overline{v}\eta_0)^{0.65} E^{*-0.44} F_n^{-0.21}; \quad \frac{h_{min}}{h_c} \approx 0.77$ |
|  | **Hydrostatic** [9.14]<br>For a given geometry and constant lubricant volumetric flow ($\dot{V} = \text{const.}$)<br>$h_{min} \approx \sqrt[3]{\frac{\eta}{F_n}} = \eta^{0.33} F_n^{-0.33}$,<br>$F_n$ load,<br>$\eta$ average viscosity of the lubricant in the lubricant cap |

cation. Friction is only generated by the shear of the viscous lubricant. Apart from the viscosity, the relative velocity between the moving bodies' lubrication gap surfaces and the lubrication gap height are decisive for friction. If both surfaces slide with the same sliding velocity, the relative velocity is $\Delta v = 0$, i.e., friction does not occur. However, when the sliding conditions are identical, the entraining velocity in the lubrication gap can be high (e.g., in the case of radial sliding bearings, in which the shaft and bushing rotate with the same speed in the same direction) so that the tribological system then has a high load-carrying capacity and a friction force of zero.

### Elastohydrodynamic Lubrication

Elastohydrodynamic lubrication (EHL) is a form of hydrodynamic lubrication in which elastic deformations of the lubricated surfaces become significant. The essential prerequisites for hydrodynamic lubrication, such as a converging lubricating film, the lubricant entraining velocity in the converging gap, and a viscous lubricant between the surfaces, are also important in EHL. Elastohydrodynamic lubrication is normally connected with nonconformal surfaces (Table 9.3). There are two different forms of EHL: for hard surfaces (hard EHL) and soft surfaces (soft EHL).

**Hard EHL.** Elastohydrodynamic lubrication for hard surfaces (hard EHL) refers to materials with high moduli of elasticity, e.g., metals. Both the elastic deformations and the viscosity's pressure dependence are equally important in this type of lubrication. The maximum occurring lubricating film pressure is typically between 0.5 and 4 GPa. The minimum lubrication film thickness normally exceeds 0.1 µm. When the loads are those normally occurring in nonconformal contacts of machine elements, the elastic deformations in hard EHL exhibit values several magnitudes greater than the minimum lubrication film thicknesses. Furthermore, the lubricant viscosity in the lubrication film can change by a magnitude of 3–4 or more, depending on the lubricant, pressure, and temperature.

The minimum lubrication film thickness $h_{\min}$ is a function of the same parameters as in hydrodynamic lubrication, but these must be augmented by the effective Young's modulus $E^*$ and the lubricant's viscosity–pressure coefficients $\alpha$. Table 9.8 indicates that, in the relationship for the minimum lubrication film thickness, the exponent for the normal load in hard EHL is approximately seven times smaller than it is in hydrodynamic lubrication. This means that, in contrast to hydrodynamic lubrication, the load only marginally influences the lubrication film thickness in hard EHL. The reasons are to be found in the increase of the contact area as the load increases in hard EHL, as a result of which a larger lubrication area is provided to bear the load. The exponent for the lubricant entraining velocity in hard EHL is greater than in hydrodynamic lubrication. Typical applications for hard EHL are toothed gears, rolling element bearings, and cam–follower pairs.

**Soft EHL.** Elastohydrodynamics for soft surfaces (soft EHL) refers to materials with low moduli of elasticity, e.g., rubber. In soft EHL, sizeable elastic deformations occur even at low loads. The maximum occurring pressures in soft EHL are typically 1 MPa, in contrast to 1 GPa for hard EHL. This low lubricating film pressure only negligibly influences the viscosity during the flow through the lubrication gap. The minimum lubrication film thickness is a function of the same parameters as in hydrodynamic lubrication with the addition of the effective modulus of elasticity $E^*$. The minimum lubricating film thickness in soft EHL is typically 1 µm. Applications for soft EHL are seals, artificial human joints, tires and nonconformal contacts in which rubber is used.

A common feature that hard and soft EHL exhibit is the generation of coherent lubricating films as a result of local elastic deformations of the friction bodies and thus the prevention of interactions between asperities. Hence, only the lubricant's shear generates frictional resistance to motion.

#### Hydrostatic Lubrication

In hydrostatic lubrication of friction bodies, a pocket or recess is incorporated in one friction body's loaded surface into which a fluid is forced from outside at constant pressure. A pump outside the bearing generates the lubricant pressure. Hence, the lubricant pump and the lubricating pocket into which the lubricant is fed under pressure are the most important features of hydrostatic lubrication. The lubricating pocket is normally positioned opposite the external load. The load-carrying capacity of a contact with hydrostatic lubrication is also assured when surfaces are not moving. When the volumetric flow of lubricant into the lubricating pocket is constant, the minimum lubrication film thickness is proportional to the cube root of the ratio of the average lubricant viscosity in the lubrication gap and the load, i.e., the minimum lubrication film thickness is less dependent on the viscosity and the load than is the case in hydrodynamic lubrication.

Hydrostatic lubrication is mainly used: where the friction partners' surfaces do not have any metallic contact, i.e., wear may not occur, not even when ramping up and ramping down a machine or at low speed; where as low a friction coefficient as possible must be produced at low speeds; and where, as a result of less effective lubricant entraining velocities in the lubrication gap, the wedge effect cannot produce any bearing lubricating film hydrodynamically.

### 9.5.3 Boundary Lubrication

In *boundary lubrication*, the friction bodies are not separated by a lubricant, the hydrodynamic lubricating film effects are negligible, and there are extensive asperity contacts. The physical and chemical properties of thin surface films of molecular thickness control the lubricating mechanisms in the contact. The base lubricant's properties are of little importance. The coefficient of friction is on the whole independent of the lubricant's viscosity. The frictional characteristic is determined by the properties of the solids involved in the

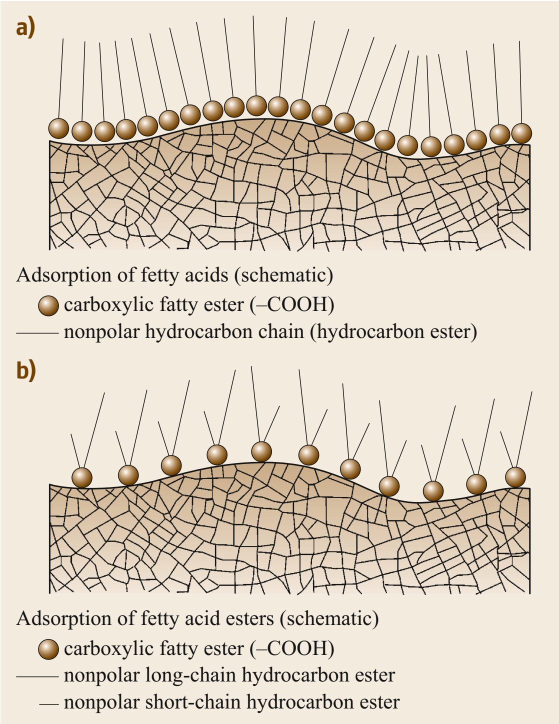


**Fig. 9.14** Adsorption on metal surfaces (after [9.15])

friction process and the boundary layers forming on the material surfaces, which primarily depend on the lubricant's properties, particularly the lubricant additives, as well as the material surfaces' properties. These boundary layers are formed by physisorption, chemisorption, and/or tribochemical reaction. The thickness of the surface boundary layer varies between 1 and 10 nm, depending on the molecule size.

In *physisorption*, additives contained in the lubricant (e.g., antiwear (AW) additives) such as saturated and unsaturated fatty acids, natural and synthetic fatty acid esters, and primary and secondary alcohols are adsorbed on the tribologically loaded surfaces. Such materials have in common a high dipole moment because of at least one polar group in the molecule (Fig. 9.14).

The coverage of the surfaces follows the laws of adsorption and is dependent on temperature and concentration. A prerequisite for the adsorption of polar groups is that the material surface exhibits a polar character so that van der Waals bonds can form. This is usually attained for metallic materials by oxide films forming on the surfaces. Problems can arise for ceramic materials however. While, for example, adsorption of fatty acids with polar end groups occurs easily on aluminum oxide with ionic bonding so that friction is diminished from a certain chain length onward, adsorption apparently does not occur on silicon carbide with covalent bonds so that the friction coefficient is not influenced [9.2].

In *chemisorption*, molecules are bonded to the surface. Substantially more-stable boundary layers develop because chemical bonds with greater bonding force are formed on the surface (e.g., the reaction of stearic acid with iron oxide when water is present, as a result of which metallic soap forms as iron stearate). Chemisorbed layers have excellent lubricating properties up to their melting point. At medium loads, temperatures, and velocities, they produce a sustained reduction of friction.

*Tribochemical reactions* between elements of the lubricant and the metallic material surface form reaction layers that generally have more thermal and mechanical load-carrying capacity than layers formed by physisorption or chemisorption. For this, chlorine, phosphorous or sulfur compounds are added to the lubricants as additives (extreme-pressure (EP) additives). The effectiveness of such EP additives depends on the speed of reaction layer formation, which is influenced by the reaction's activation energy, the surface temperature, and the concentration of the additive. EP active ingredients themselves, or their thermal cleavage products, react with the surface metallic oxide in very short time ($10^{-6}$–$10^{-7}$ s), forming a reaction layer that adheres well and shears easily (Fig. 9.15).

While only very little or no wear appears in hydrodynamic and elastohydrodynamic lubrication because there are no asperity contacts, the number of asperity contacts grows in boundary lubrication and thus so does the rate of wear as the load increases. However, compared with unlubricated conditions, the rates of wear in boundary lubrication are substantially lower.

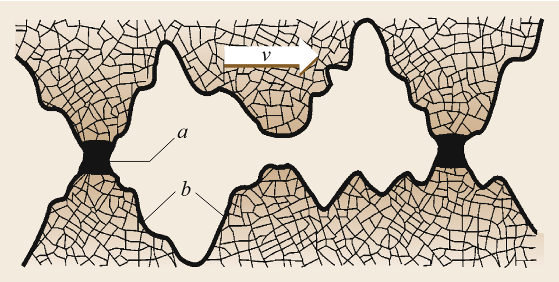


**Fig. 9.15** Formation of a protective layer by EP active ingredients of sulfur compounds. *a*: metal-sulfide compound; *b*: metal surface (after [9.15])

### 9.5.4 Mixed or Partial Lubrication

If the load in hydrodynamically or elastohydrodynamically lubricated machine elements grows too large or the lubricant entraining velocity in the lubrication gap is too small, the lubricating film becomes too thin so that asperity contacts appear at some points. This is mixed friction, in which the states of boundary friction and fluid friction coexist. The lubricating film is no longer coherent because individual surface asperities of the paired parts penetrate it and cause direct contact of the sliding surfaces. In partial or mixed lubrication, the normal load is partly carried by hydrodynamic film pressures and partly by contact of the frictional surfaces at the asperity contacts. The transition from hydrodynamic or elastohydrodynamic lubrication to partial lubrication does not happen suddenly when the load is increased but rather the share of the load carried by the hydrodynamic pressure buildup decreases gradually, while the share carried by the asperities grows in equal measure.

### 9.5.5 Lubrication with Solid Lubricants and Surface Coatings

In lubrication with solid lubricants and in coatings, the most important measures for improving friction, preventing scuffing or seizing, and minimizing wear are applying metallic (e.g., indium) or nonmetallic layers (e.g., synthetic resin coatings), forming reactionlayers through chemical transformation (e.g., phosphatizing), and using solid lubricants. Nonmetallic layers also include layers obtained by physical vapor deposition (PVD) or chemical vapor deposition (CVD) from the gas phase; the layer has a thickness in the micrometer range and can be made of, for example, titanium nitride (TiN), titanium aluminum nitride (TiAlN), chromium nitride (CrN) or tungsten carbide/carbon (WC/C).

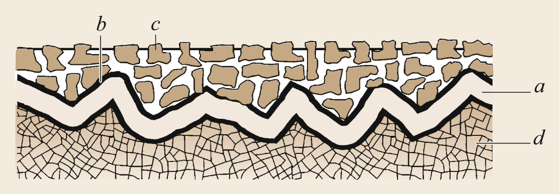


**Fig. 9.16** Solid lubrication made of lubricating varnish: *a*: phosphate layer (3 μm); *b*: binder; *c*: solid lubricant; *d*: base metal (after [9.15])

Among the solid lubricants, crystalline solids with layer lattices (lamina structure) appear particularly suitable. Their structure is characterized by lamellae that can be displaced slightly with respect to each other along glide planes. Typical representatives are graphite (C) and molybdenum disulfide ($MoS_2$). Certain plastics such as PTFE are also used as solid lubricants.

Apart from these, coating layers on metals, which are adhesive, act as lubricants, and are firm to the touch, so-called lubricating varnishes, also play a role (Fig. 9.16). They have a large proportion of solid lubricants and can be used as dry films over a wide range of temperatures.

## 9.6 Lubricants

Gaseous, fluid, consistent or solid lubricants are used in mechanical engineering. Gaseous lubricants are utilized, for example, in high-speed, lightly loaded machines (e.g., ultracentrifuges, gas pumps for nuclear power plants) or in equipment technology. The most important group of substances for fluid lubricants are the mineral oils, as well as synthetic, animal, and vegetable oils and even water in special applications. Greases and adhesive lubricants are consistent lubricants. Greases consist of mineral or synthesis oils that are thickened with, for instance, soaps. There are bitumen-based adhesive lubricants as well as bitumen-free adhesive spray lubricants. These are principally used to lubricate larger, open-toothed gears. Solid lubricants are frequently introduced into fluid or consistent carrier substances. Solid lubricants are used in pure form only under special operating conditions. Among others substances, solid lubricants include graphite, molybdenum disulfide, PTFE, etc.

During operation, pressure and shear stresses, among others, act on a lubricant at different temperatures. The lubricant comes into contact with gases (e.g., air), liquids (e.g., water), and solids (e.g., metals, sealing materials, wear particles). In addition, contact with human skin cannot be ruled out. This results in a number of requirements that a good lubricant should have. For a particular application, it should have an appropriate viscosity–temperature and viscosity–pressure behavior, a low pour point, and preferably not be volatile, exhibit high-temperature, shear, and oxidation stability, and be hydrolysis and radiation resistant in special cases. Beyond this, a good lubricant should be compatible with the constructional materials used and be nontoxic and not cause any disposal problems.

## 9.6.1 Lubricating Oils

Among the fluid lubricants, a difference is made between mineral oils and synthetic oils.

### Mineral Oils

Mineral oils are obtained from naturally deposited crude oil with the aid of distillation and refining. Crude oils chiefly consist of hydrocarbons and organic oxygen, sulfur, and nitrogen compounds. The exact composition of a crude oil depends on its origin (provenance). When mineral oils are manufactured, systematic selection of the base oil and control of the manufacturing process can produce particular compositions and thus influence desired properties. Mineral oils are mixtures of hydrocarbons, which can be subdivided as a function of their structure into open-chain hydrocarbons (paraffins) and cyclic hydrocarbons (naphthalenes, aromatics), which can each be saturated or unsaturated.

Mineral oils' performance characteristics change as their working life increases. A primary reason for diminished performance is gradual oxidization of hydrocarbon chains, with aromatics reacting more than naphthenes and naphthenes more than paraffins. As a result, sludge-like deposits can form, which impede the oil supply by clogging oil feed lines and filters. Moreover, the formation of organic acids can be boosted, which can cause corrosion of machine parts. This can be prevented in part by admixing additives (e.g., antioxidants, detergent, and dispersant agents). More information on their effect and the use of additives can be found in Sect. 9.6.1, *Additives*.

### Synthetic Oils

Synthetic-base lubricating oils are produced by chemical synthesis from chemically defined structural elements (e.g., ethylene). Their development has made it possible to systematically satisfy even extreme requirements (e.g., lubricant temperature $> 150\,°\mathrm{C}$). According to their chemical composition, synthetic lubricants are subdivided into synthetic hydrocarbons, which only contain carbon and hydrogen (e.g., polyalphaolefines (PAO), dialkylbenzenes (DAB), polyisobutenes (PIB)), and synthetic fluids (e.g., polyglycols, carboxylic acid esters, phosphoric acid esters, silicon oils, polyphenyl ethers, fluorine-chlorine-carbon oils). Typical characteristics of synthetic oils are provided in Table 9.9 and a comparison of the properties of synthesis oils with those of mineral oil is presented in Table 9.10.

Synthetic oils have a number of advantages over mineral oils. They have better resistance to aging (thermal and oxidative stability) and thus their useful life is three to five times longer. They exhibit a more favorable viscosity–temperature behavior (with a significantly lower dependence of viscosity on temperature), display better flow properties at low temperatures and lower volatility at high temperatures, can cover applications operating at a substantially expanded range of temperature, and are radiation and flame resistant. Moreover, synthetic lubricants can be used to obtain

**Table 9.9** Characteristics of various synthetic oils (after [9.16])

| Property | Mineral oil | PAO | Polyglycol (water insoluble) | Ester | Silicon oil | Alcoxifluorine oil |
|---|---|---|---|---|---|---|
| Viscosity at 40 °C ($mm^3/s$) | 2–4500 | 15–1200 | 20–2000 | 7–4000 | 4–100 000 | 20–650 |
| Use for oil-sump temperature (°C) | 100 | 150 | 100–150 | 150 | 150–200 | 150–220 |
| Use for oil-sump temperature (°C) | 150 | 200 | 150–200 | 200 | 250 | 240 |
| Pour point (°C) | −20$^{a}$ | −40$^{a}$ | −40 | −60$^{a}$ | −60$^{a}$ | −30$^{a}$ |
| Flash point (°C) | 220 | 230–260$^{a}$ | 200–260 | 220–260 | 300$^{a}$ | – |
| Evaporation losses$^{b}$ | ○ | + | ○ to − | + | +$^{a}$ | ++$^{a}$ |
| Water resistance$^{c}$ | + | + | +$^{a,d}$ | + to ○ | + | + |
| $\eta$–$T$ behavior$^{c}$ | ○ | + to ○ | + | + | ++ | + to ○ |
| Pressure–viscosity coefficient ($10^8\,m^2/N$)$^{e}$ | 1.1–3.5 | 1.5–2.2 | 1.2–3.2 | 1.5–4.5 | 1.0–3.0 | 2.5–4.4 |
| Suitable for high temperatures (≈ 150 °C)$^{c}$ | ○ | + | + to ○$^{a}$ | +$^{a}$ | ++ | ++ |
| Suitable for high load$^{c}$ | ++$^{f}$ | ++$^{f}$ | ++$^{f}$ | + | −$^{a}$ | + |
| Compatibility with elastomers$^{c}$ | + | +$^{a}$ | ○$^{g}$ | ○ to − | ++ | + |
| Price ratios | 1 | 6 | 4–10 | 4–10 | 40–100 | 200–800 |

[a] dependent on type of oil;
[b] very low, ++; low, +; moderate, ○; moderate to high, ○ to −;
[c] excellent, ++; good, +; moderate to good, + to ○; moderate, ○; moderate to poor ○ to −; poor, −;
[d] difficult to separate since density is identical;
[e] measured up to 2000 bar, amount is dependent on type of oil and the viscosity;
[f] with EP additives;
[g] inspect when coating

**Table 9.10** Comparison of the properties of natural and synthetic base oils for lubricants (after [9.15])

| Property | A | B | C | D | E | F | G | H | I | J | K |
|---|---|---|---|---|---|---|---|---|---|---|---|
| Viscosity–temperature behavior (viscosity index, VI) | − | + | ○ | + | + | ○ | −− | −− | ++ | ++ | − |
| Low-temperature performance (pour point) | −− | ○ | ++ | ++ | + | ○ | −− | − | ++ | + | ○ |
| Oxidation stability (aging test) | − | −− | + | ○ | + | −− | + | ○ | + | + | ++ |
| Thermal stability (heating under absence of oxygen) | − | − | − | ○ | + | ○ | ++ | − | + | ○ | + |
| Volatility (evaporating loss) | − | ○ | ○ | ++ | ++ | ○ | ○ | + | + | ○ | ○ |
| Finish compatibility (effect on coatings) | ++ | − | ++ | − | − | − | − | −− | ○ | − | ○ |
| Water resistance (hydrolysis test) | ++ | −− | ++ | − | − | ○ | ++ | − | ○ | − | + |
| Antirust properties (corrosion test) | ++ | ++ | ++ | − | − | ○ | − | − | ○ | −− | + |
| Seal compatibility (swelling behavior) | ○ | − | ++ | − | − | ○ | ○ | −− | ○ | ○ | − |
| Flame resistance (ignition temperature) | −− | −− | −− | − | − | − | − | ++ | ○ | − | ++ |
| Additive solubility (dissolving of larger concentrations) | ++ | ○ | + | −− | ○ | − | + | ○ | −− | ○ | − |
| Lubricity (load-carrying ability) | ○ | ++ | ○ | + | + | ○ | ++ | ++ | −− | − | ++ |
| Biodegradability (degradability test) | − | ++ | ○ | ++ | ++ | ++ | −− | + | −− | − | −− |
| Toxicity | ○ | ++ | ++ | ○ | ○ | + | ○ | −− | ++ | − | + |
| Miscibility (formation of a homogeneous phase) | ++ | ++ | ++ | + | + | −− | ○ | − | −− | − | −− |
| Price ratio to mineral oil | 1 | 3 | 4 | 7 | 8 | 8 | 350 | 7 | 65 | 25 | 350 |

Weighting: 1: ++; 2: +; 3: ○; 4: −; 5: −−
A – Mineral oil (solvent neutral);
B – Rape oil;
C – Polyalphaolefin;
D – Carboxylic acid ester;
E – Neopentyl polyol esters;
F – Polyalkylenglycol (polyglycol);
G – Polyphenyl ether;
H – Phosphoric acid ester;
I – Silicon oil;
J – Silicate ester;
K – Fluorine-chlorine-carbon oil (chlorotrifluoroethylene)

specific frictional properties, e.g., lower friction coefficients to minimize power loss in ball bearings or gears, or higher friction coefficients to increase the transmittable torque in friction gears.

On the other hand, synthetic lubricants often cannot be used as universally as mineral oils since they have been developed for specific properties. In addition, they are more strongly hydroscopic (water attracting), display only slight air release characteristics (risk of foaming), mix poorly or not at all with mineral oils, are toxic to a large extent, and are characterized by poor compatibility with other materials (risk of chemical reaction with seals, paints, and nonferrous metals) and by poor solubility for additives. They are not always available, most notably in certain viscosity classes, and they frequently cost substantially more. Table 9.11 details examples of typical areas of application of synthetic oils.

### Biodegradable Oils

Environmentally compatible lubricating oils are increasingly being used, for example, in motor vehicles and equipment in water protection areas and in hydraulic engineering, in vehicles for agriculture and forestry, and in openly running gears with loss lubrication (excavators, mills). They are readily and rapidly degradable, have a low water hazard class, and are toxicologically harmless. Their base substances have to be degraded in a degradability test (e.g., CEC L-33-T-82) by a defined amount within a specified time and the additives used (up to a maximum of 5%) should be potentially degradable. Native oils and native base synthetic esters as well as fully synthetic esters and polyglycols are used. Native oils (e.g., rape oil and natural esters) are unsuitable for high temperatures (>70 °C) and additionally have low thermal stability and resistance to aging. The synthetic oils suitable for continuous high temperatures are often used as hydraulic oils in agricultural and forestry machines. Polyglycols are used, for example, as readily biodegradable oils in water engineering.

### Additives

Additives are substances that either give new characteristics to mineral, synthesis or vegetable oils or enhance already existing positive properties. The quantity of ad-

**Table 9.11** Examples of use of the most important synthetic lubricants (after [9.17])

| Product group | Examples of use |
|---|---|
| PAO (synthetic hydrocarbons) | ● High-performance oils for diesel engines<br>● Multigrade engine oils<br>● Gear lubrication at high thermal stress<br>● Compressor oils |
| Carboxylic acid esters | ● Aircraft engine oils<br>● Fuel economy oils (low-friction engine oils)<br>● Base oil for high- and low-temperature greases<br>● Applications requiring good and fast biodegradability |
| Neopentyl polyol esters | ● Applications similar to those for carboxylic acid esters but especially wherever oxidation stability and better additive solubility are required |
| Polyalkylglycols (polyglycols) | ● Metalworking fluids<br>● Gear oils (worm gears)<br>● Hydraulic fluids (flame resistant)<br>● Lubricant for compressors and pumps |
| Polyphenyl ethers | ● High-temperature lubricants (up to 400 °C)<br>● Applications requiring resistance to ionizing radiation ($\gamma$ rays and thermal neutrons) |
| Phosphoric acid esters | ● Plasticizers<br>● Flame-resistant hydraulic oils<br>● Safety lubricants for air and gas compressors<br>● EP additives |
| Silicone oils | ● Special lubricants for high temperatures<br>● Base oil for lifetime lubricating greases (e.g., for clutch release bearings for motor vehicle clutches, starters, brakes, and axle components) |
| Silicate esters | ● Hydraulic oils for lower temperatures<br>● Heat exchange fluids |
| Fluorine-chlorine-carbon oils | ● Lubricants for oxygen compressors and for pumps for aggressive fluids |

ditive used differs greatly. Thus, circulating or hydraulic oils may only contain 0.1%, whereas special engine and gear oils may contain up to 30% additives.

All properties of lubricants cannot be changed by additives. However, by using additives a clear improvement in lubrication can be obtained by modifying some properties. Thus, for example, heat dissipation, viscosity–density properties, and temperature resistance cannot be influenced by additives. Improvements brought about by additives are obtained for low-temperature performance, aging stability, viscosity–temperature properties, and corrosion protection. Only additives can attain good cleaning performance, favorable dispersion behavior, antiseizing properties, and foam inhibition.

Additives have to be matched to the base oil in terms of quantity and composition and the presence of other additives since they respond differently to the base oil and are not mutually compatible in every case. For example, there are antagonistic effects between viscosity index improves and antifoam additives, between detergent/dispersant additives and antiwear, antiseizing, and antifoam additives, and between corrosion inhibitors and antiwear and antiseizing additives [9.15].

A difference can be made between additives that form surface layers and those that change the properties of the lubricant itself. Additives forming surface layers act as a lubricating film above all when there is insufficient lubrication, as a result of which friction is reduced and the load-carrying capacity of sliding–rolling pairs is improved. Among others, this group of additives includes antiwear (AW) additives, extreme pressure (EP) additives, and friction modifiers. Adding additives that form surface layers also has drawbacks though. Thus, lubricants with additives oxidize faster than normal mineral oils and corrosive acids and insoluble residues frequently form. Hence these additives should only be used when necessitated by the operating conditions. Additives that modify lubricants influence, for example, foaming behavior, corrosion behavior, sludging, and pour point. Table 9.12 provides an overview of the most important types of additives and their applications.

During operation, the effectiveness of some additives can decrease (exhaustion) since reaction with the

**Table 9.12** Additives, typical types of additives, applications, and active mechanisms (after [9.18])

| Additive | Types of additive | Application | Active mechanisms |
|---|---|---|---|
| AW additive | Zinc dialcyldithiophosphates, tricresylphosphates | Decrease of inordinate wear metal surfaces | Reaction with metal surfaces produces layers that are plastically deformed and improves the contact pattern |
| EP additives | Sulfurized greases and olefines, chlorohydrocarbons, lead salts of organic acids, aminophosphates | Prevention of microwelding between metal surfaces at high pressures and temperatures | Reaction with metal surfaces produces new bonds with lower shear resistance than the base metal. There is constant shearing off and reformation |
| Friction modifiers | Fatty acids, fat amines, solid lubricants | Reduction of friction between metal surfaces | Highly polar molecules are absorbed on metal surfaces and separate the surfaces, solid lubricants form friction-reducing surface film |
| Viscosity index improvers | Polyisobutylenes, polymethylacrylates, polyacrylates, ethylenepropylene, styrene maleic acid esters, copolymers, hydrogenated styrene-butadiene-copolymers | Reduction of dependence of viscosity on temperature | Polymer molecules are strongly balled in cold oil (poor solvent) and take on greater volume in warm oil (good solvent) by unballing. This produces a relative thickening in oil |
| Pour point depressants | Paraffin-alkylated naphthalenes and phenols, polymethylacrylates | Decrease of pour point of the oil | Encasing prevents the agglomeration of paraffin crystals |
| Detergent additives | Normal or alkaline calcium, barium or magnesium-sulfinates, phenates or phosphonates | Reduction or prevention of deposits in engines at high operating temperatures | Reaction with the oxidation products controls the formation of coating and sludge. Products are produced that are oil soluble or suspended in oil |
| Dispersant additives | Polymers such as nitrogenous polymethylacrylates, alkyl succunimides and succinate esters, high molecular weight amines and amides | Prevention or delay of the development and deposition of sludge at low operation temperatures | Dispersants have a pronounced affinity for impurities and encase these with oil soluble molecules that suppress the agglomeration and deposition of sludge in the engine |
| Oxidation inhibitors | Inhibited phenols, amines, organic sulfides, zinc dithiophosphates | Minimization of the formation of resin, coating, sludge, acid, and polymer-like compounds | Reducing the organic peroxides ends the oxidation chain reaction. reduced oxygen intake by the oil decreases the acid formation. Catalytic reactions are prevented |
| Corrosion inhibitors | Zinc dithiophosphates, sulfurized terpenes, phosphorized, sulfurized olefines | Protection of bearing and other metal surfaces against corrosion | Acts as an anticatalyst; film forms on metal surfaces as protection against attacks from acids and peroxides |
| Rust inhibitors | Amine phosphates, sodium, calcium, and magnesium sulfates, alkyl succinic acid, fatty acids | Protection of ferrous surfaces against rust | Metal surfaces prefer to adsorb polar molecules and they serve as a barrier against water neutralization by acids |
| Metal deactivators | Triarylphosphate, sulfur compounds, diamines, dimercaptothiadiazop deriviatives | Suppression of the catalytic influence on oxidation and corrosion | A protective film is adsorbed an metal surfaces, which inhibits the contact between the bases metal and the corrosive substance |
| Foam inhibitors | Silicon polymers, tributylphosphates | Protection of the development of stable foam | Attacking the oil film surrounding every air bubble reduces the boundary surface stress. As a result smaller bubbles coalesce into larger bubbles that rise to the surface |
| Adhesion improvers | Soaps, polyisobutylenes and polyacrylate polymers | Increase of the oils's adhesive ability | Viscosity is increased. Additives are viscous and sticky |
| Emulsifier | Sodium salts of sulfonic acids and other organic acids, fat amine salts | Emulsification of oil in water | Adsorbing the emulsifier in the oil/water boundary surface reduces boundary surface stress, as a result of which one fluid disperses into another |
| Demulsifier | Anionic sulfon acid compounds (dinonylnaphthalinsulfonate) | Demulsification of water | A boundary layer develops between water and oil form substances active in the boundary surface |
| Bactericide | Phenols, chlorine compounds, formaldehyde derivatives | Increase of the emulsion's working life, prevention of unpleasant odors | The growth of microorganisms is prevented or delayed |

materials or atmospheric oxygen causes their concentration to drop. Once the concentration of the additive falls below a certain value, an oil change is necessary.

## 9.6.2 Lubricants (Lubricating Greases)

Consistent lubricants have a flow limit. No movement occurs below a shear stress that is specific to the lubricant. Only when this flow limit has been exceeded does the viscosity drop from a virtually infinitely high to a measurable value.

Lubricating greases consist of three components: a base oil (75–96 wt%), a thickener (4–20 wt%), and additives (0–5 wt%). Suitable thickeners can be dispersed both in mineral oils and in synthetic or vegetable oils so that consistent lubricants are produced. By far, most greases are manufactured using soaps (metallic salts from fatty acids) as thickeners. Thus, fatty acids are dissolved in the base oil at relatively high temperatures and a suitable metal hydroxide (e.g., hydroxides of sodium, lithium, and calcium or to a lesser extent barium and aluminum) is added subsequently. Long-chain fatty acids come from vegetable or animal oils and can be hydrogenated. Occasionally, not only long-chain fatty acids but also short-chain acids such as acetic, propionic, benzoic acid, etc. are used. Then so-called complex soaps are produced [9.15]. Most soap compounds form a fibrous matrix of interlocking particles, which retains the base oil (Fig. 9.17).

By contrast, aluminum soaps contain a spherical gel structure. The grease's lubricating action is based on the base oil being dispensed slowly and sufficiently in operation under load. The delivery of the base oil depends strongly on the temperature. The lubricating grease releases less and less oil as the temperature drops and the grease becomes stiffer and stiffer (consistency). Beyond a certain temperature limit, this eventually leads to insufficient lubrication in the friction contact. As the temperature increases, more and more oil is released. Simultaneously, the grease ages and oxidizes faster and the deterioration products produced have an adverse effect on the lubrication. A standard value is to cut the grease's working life and thus the relubrication interval in half for every 15 °C rise in temperature above approximately 70 °C. Below 70 °C, the grease's working life and consequently the relubrication interval can be extended unless the temperature drops below the lower limit. The type of base oil, its viscosity, and the additives it contains are decisive for the lubricating properties.

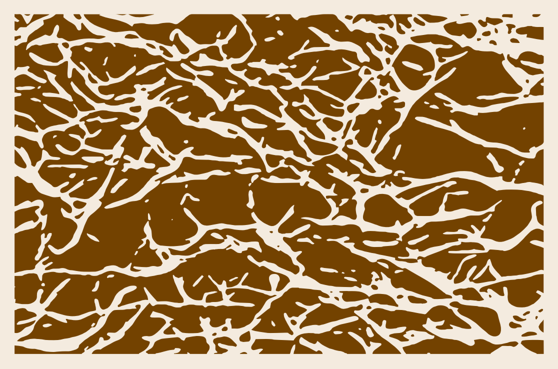


**Fig. 9.17** Fiber structure of a grease with soap thickener [9.19]

Lubricating greases are predominantly used at low speeds since the lubricant results in lower frictional heat transportation than oil lubrication. The relevant temperature ranges are generally from −70 to +350 °C. Lubricating greases also frequently have the job of protecting lubricating points from the infiltration of water and dirt and keeping out small quantities of dirt without disrupting the function. Tables 9.13 and 9.14 present performance the characteristics and applications of different lubricating greases.

Additives mainly serve to improve particular performance characteristics of the base oils. They must be uniformly distributed and be dissolved. Additives can improve the following properties of greases in particular: oxidation stability, corrosion protection, water resistance, adhesiveness, and antiwear properties.

The greatest care is necessary when mixing different types of lubricating greases since not all types of lubricating greases are mutually compatible (Table 9.15). Thus, for example, sodium soap grease is incompatible with nearly all other lubricating greases with the exception of barium complex soap grease. Lithium soap grease is incompatible with sodium soap grease, aluminum complex soap grease, and bentonite grease. In turn, bentonite grease is incompatible with all other types of grease.

## 9.6.3 Solid Lubricants

Solid lubricants are used especially whenever fluid and consistent lubricants cannot provide the lubricating action required. This is frequently the case under the following operating conditions: low sliding speeds, oscillating motions, high specific loads, high or low operating temperatures, extremely low ambient pressures (vacuum), and aggressive ambient atmospheres. Solid lubricants are also used to improve particular properties of fluid and consistent lubricants, i.e., as additives, for example, to minimize friction and wear and to guar-

**Table 9.13** Performance characteristics of mineral-oil-based lubricating greases [9.15]

| Characteristic | | Sodium | Lithium | Calcium | Calcium complex | Bentonite |
|---|---|---|---|---|---|---|
| Thickener form | | Fiber | Fiber | Fiber | Fiber | Platelets |
| Fiber length (µm) | | 100 | 25 | 1 | 1 | 0.5 |
| Fiber diameter (µm) | | 1 | 0.2 | 0.1 | 0.1 | 0.1 |
| Short description | | Long fibered | Medium fibered | Short fibered | Short fibered | Short fibered |
| **Properties** | | | | | | |
| Drop point (°C) | | 150–200 | 170–220 | 80–100 | 250–300 | ≈ 300 |
| Operating temperature | Upper (°C) | +100 | +130 | +50 | +130 | +150 |
| | Lower (°C) | –20 | –20 | –20 | –20 | –20 |
| Water resistance | | Not stable | Good | Very good | Very good | Good |
| Mechanical stability[a] (0.1 mm) | | 60–100 | 30–60 | 30–60 | < 30 | 30–60 |
| Corrosion protection[b] | | Good | Very poor | Poor | Poor | Good |
| **Use** | | | | | | |
| Suitability for ball bearings | | Good | Very good | Variable | Variable | Very good |
| Suitability for sliding bearings | | Good | Good | Variable | – | – |
| Primary use | | Low-viscosity gear grease | Multipurpose grease | – | Multipurpose grease | High-temperature grease |
| Price | | Medium | High | Low | Very high | Very high |

[a] Difference in penetration after 60 and 100 000 double strokes
[b] Can be noticeably improved by additives

**Table 9.14** Areas of application of synthetic lubricating greases [9.15]

| | Mineral oil (benchmark) | PAO | Ester oils | Silicon oils | Alkoxyfluorine oils |
|---|---|---|---|---|---|
| Upper limit of application (°C) | 150 | 200 | 200 | 250 | 250 |
| Lower limit of application (°C) | –40 | –70 | –70 | –75 | –30 |
| Lubrication of metals | ++ | ++ | +++ | ––– | – |
| Lubrication of plastics | ○ | ++ | ○ | +++ | +++ |
| Hydrolysis resistance | ++ | ++ | ○ | +++ | +++ |
| Chemical resistance | + | + | –– | ++ | +++ |
| Elastomer compatibility | ○ | + | ○ | +++ | +++ |
| Toxicity | – | + | + | +++ | +++ |
| Flammability | ––– | ––– | + | ++ | +++ |
| Radiation resistance | –– | –– | – | + | ++ |

+++ excellent; ++ very good; + good; ○ moderate; – adequate; –– limited; ––– poor

**Table 9.15** Compatibility of types of lubricating grease [9.15]

| Grease type | Na | Li | Ca | Ca complex | Ba complex | Al complex | Bentonite |
|---|---|---|---|---|---|---|---|
| Na grease | | – | – | – | + | – | – |
| Li grease | – | | + | + | + | – | – |
| Ca grease | – | + | | – | + | – | – |
| Ca complex | – | + | + | | + | – | – |
| Ba complex | + | + | + | + | | + | – |
| Al complex | – | – | – | – | + | | – |
| Bentonite | – | – | – | – | – | – | |

+ compatible; – incompatible

antee antiseizure performance. Solid lubricants in the form of powders, pastes or lubricating varnishes contribute directly to the build up of the lubricating film on the one hand or improve the lubricating properties in oils, greases or bearing materials on the other hand.

Substances with a layer lattice structure (graphite, the sulfides $MoS_2$ and $WS_2$), selenides ($WSe_2$), organic substances (PTFE, amides, imides), soft nonmetals (lead sulfide, iron sulfide, lead oxide, and silver iodide), soft nonferrous metals (gold, silver, lead, copper, and indium), and reaction layers on the surface (oxide, sulfide, nitride, and phosphate layers) are used as solid lubricants. Graphite needs water to adhere and to reduce shear strength (low friction) and hence is unsuitable for use in a dry atmosphere or vacuum. Molybdenum disulfide adheres well to all metal surfaces with the exception of aluminum and titanium. It is a highly suitable solid lubricant for temperatures up to 350 °C but costs more than graphite. Polytetrafluoroethylene (PTFE or Teflon) exhibits a low friction factor at low speeds and high loads and is suitable for temperatures from −250 to +250 °C.

Their high proportion of solid lubricants (graphite, molybdenum disulfide or PTFE) distinguishes lubricating varnishes from decorative industrial varnishes. They can be used as a dry film at temperatures between −180 and +450 °C. Lubricating varnishes with oil-resistant binders can also be used in oily systems and are suitable, for example, for bypassing the critical break-in phase without damage or for shortening the break-in time.

## 9.6.4 Properties of Lubricants

The properties of fluid and consistent lubricants are specified by data predominantly determined with standardized test procedures. It should be noted that not all results from laboratory tests are meaningful for lubrication applications. Along with viscosity, other properties of lubricants are density, specific heat, thermal conductivity, pour point, flashpoint, fire point, foaming behavior, compatibility with sealing materials, and consistency in the case of lubricating grease. Furthermore, aging resistance is vital since it characterizes the abatement of the lubricity and thus the useful life of lubricating oils and determines the oil change interval (Fig. 9.18).

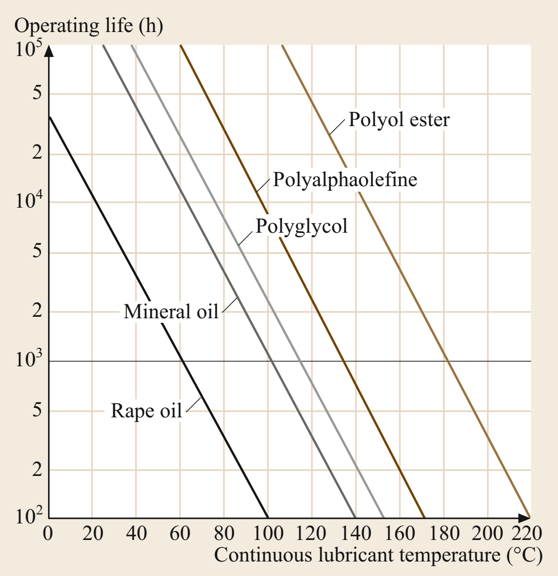


**Fig. 9.18** Influence of the lubricant continuous temperature on the approximate useful life of mineral-oil-based and synthetic lubricants (after [9.17])

### Viscosity

One of a lubricant's most important rheological properties is its viscosity. A fluid's dynamic (or absolute) viscosity $\eta$ is a measure of the amount it resists a relative motion. The dynamic viscosity $\eta$ is defined as the shear force $F$ required to realize the relative movement between two planes, acting in the direction of the lubricant flow between the two planes and related to the friction area $A$ and the velocity gradient $\mathrm{d}v/\mathrm{d}y$ between the planes (Fig. 9.19).

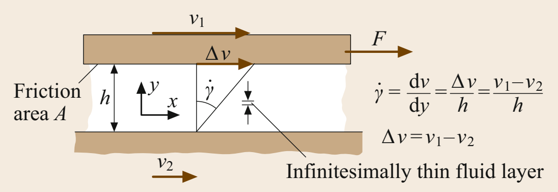


**Fig. 9.19** Gap flow and velocity gradient for a Newtonian fluid

Since the shear force per friction area corresponds to the shear stress $\tau$ and the velocity gradient to the local shear strain rate $\dot{\gamma}$ (also called shear rate), the following relationship applies

$$\eta=\frac{F/A}{\mathrm{d}v/\mathrm{d}y}=\frac{\tau}{\dot{\gamma}}\,, \qquad (9.8)$$

with $\dot{\gamma}=\mathrm{d}v/\mathrm{d}y=\Delta v/h$, where $\Delta v$ stands for the relative velocity between the two friction bodies and $h$ is the lubrication gap height. The absolute value of the shear force $F$ equals that of the frictional force $F_\mathrm{f}$.

Fluids that can be characterized with (9.8) at constant temperatures and pressures are also called *Newtonian fluids*. Many common fluids, especially those with relatively simple molecular structures, fall into this group (e.g., undoped mineral oils, synthetic fluids, vegetable oils, water, and gases). The unit for dynamic viscosity $\eta$ is $\mathrm{N\,s/m^2}=\mathrm{Pa\,s}$. Engineering generally uses mPa s however, which corresponds to cP (centipoise), which was common earlier.

The viscosity is measured with commercially available viscometers, which are standardized as rotation, capillary, falling ball, and falling rod viscometers. Rotation and falling ball viscometers determine the dynamic viscosity $\eta$, while the capillary viscometer used most determines the ratio of the dynamic viscosity $\eta$ and the density $\rho$. This ratio is known as the kinematic viscosity $\nu$, so that the following applies

$$\nu = \frac{\eta}{\rho} \,. \qquad (9.9)$$

The kinematic viscosity $\nu$ is a mathematical value, i.e., it is not a material property. Its units are $\mathrm{m^2/s}$. Usually however $\mathrm{mm^2/s}$ is used, corresponding to cSt (centistokes), which was used earlier. The kinematic viscosity is the basic of the standardization of oils. Table 9.16 shows the 20 ISO viscosity groups (ISO VG) by ISO 3448. The basic of the viscosity groups is the middle kinematic viscosity $\nu_\mathrm{m}$ at 40 °C. Oils of a viscosity group can vary ± 10% around the middle viscosity.

The viscosity of substances that do not behave like a Newtonian liquid is dependent on the temperature, pressure, shear rate, and mean molecular weight. In addition, shear strains are dependent not only on the instantaneous shear rate but also on the past shear history (the lubricant's *memory properties*).

**Table 9.16** Viscosity groups by ISO 3448

| ISO VG | Middle kinematic viscosity at 40 °C ($\mathrm{mm^2/s}$) | Minimum kinematic viscosity at 40 °C ($\mathrm{mm^2/s}$) | Maximum kinematic viscosity at 40 °C ($\mathrm{mm^2/s}$) |
|---|---|---|---|
| 2 | 2.2 | 1.98 | 2.42 |
| 3 | 3.2 | 2.88 | 3.52 |
| 5 | 4.6 | 4.14 | 5.06 |
| 7 | 6.8 | 6.12 | 7.48 |
| 10 | 10 | 9.00 | 11.0 |
| 15 | 15 | 13.5 | 16.5 |
| 22 | 22 | 19.8 | 24.2 |
| 32 | 32 | 28.8 | 35.2 |
| 46 | 46 | 41.4 | 50.6 |
| 68 | 68 | 61.2 | 74.8 |
| 100 | 100 | 90.0 | 110 |
| 150 | 150 | 135 | 165 |
| 220 | 220 | 198 | 242 |
| 320 | 320 | 288 | 352 |
| 460 | 460 | 414 | 506 |
| 680 | 680 | 612 | 748 |
| 1000 | 1000 | 900 | 1100 |
| 1500 | 1500 | 1350 | 1650 |
| 2200 | 2200 | 1980 | 2420 |
| 3200 | 3200 | 2880 | 3520 |

### 9.6.5 Dependence of Viscosity on Temperature and Pressure

The viscosity of lubricating oils greatly depends on the operating temperature. As the temperature increases, the viscosity of the lubricating oil decreases considerably.

To design lubricated tribological contacts it is important to know the viscosity at the operating temperature since this decisively influences the lubrication film thickness between the surfaces to be separated. The lubricant's viscosity–temperature behavior ($\eta$–$T$ behavior) is determined metrologically with viscometers and is frequently specified by simple power and exponential equations. The following equation from Vogel has proven to be valuable for lubricating oils used in the field

$$\eta(T) = A \exp\left(\frac{B}{C+T}\right) . \qquad (9.10)$$

In this equation, $\eta$ stands for the dynamic viscosity in Pa s, $A$, $B$, and $C$ are values specific to lubricants that have to be determined for every lubricant, and $T$ is the operating temperature in °C.

Figure 9.20 shows an example for the viscosity–temperature behavior for liquid industrial lubricants in accordance with ISO 3448. The selection of a logarithmic scale on the ordinate axis and a hyperbolic scale on the abscissa gives the viscosity curves a straight gradient. This makes it possible to determine the $\eta$–$T$ behavior with only two measurements.

The viscosity index (VI) is also frequently used in practice to specify viscosity–temperature behavior. The viscosity index is a measure of the slope of the *straight lines* of viscosity–temperature compared with a reference lubricant. The higher the VI, the more favorable the $\eta$–$T$ behavior. A high VI is characterized by a relatively mild change of viscosity as temperature changes and a low VI by a relatively intense change. Oils that hardly tend to thicken at low temperatures and that do not become low viscosity too fast at high temperatures, i.e., oils with a high VI, should be preferred when machine parts to be lubricated have to operate over a wide range of temperatures. Common paraffin basic oils exhibit a VI of 90—100, synthetic lubricants have a VI of approximately 200 and above.

The viscosity of lubricating oils increases as the pressure rises. The viscosity's pressure dependence only really becomes noticeable at high pressure though. The influence of the pressure decreases as the temperature increases. The viscosity of mineral oil increases more strongly as the pressure increases as the viscosity–temperature curve becomes steeper. The viscosity–pressure behavior can be approximated by

$$\eta(p) = \eta_0(T) \exp\left[\alpha\,(p - p_\mathrm{u})\right] . \qquad (9.11)$$

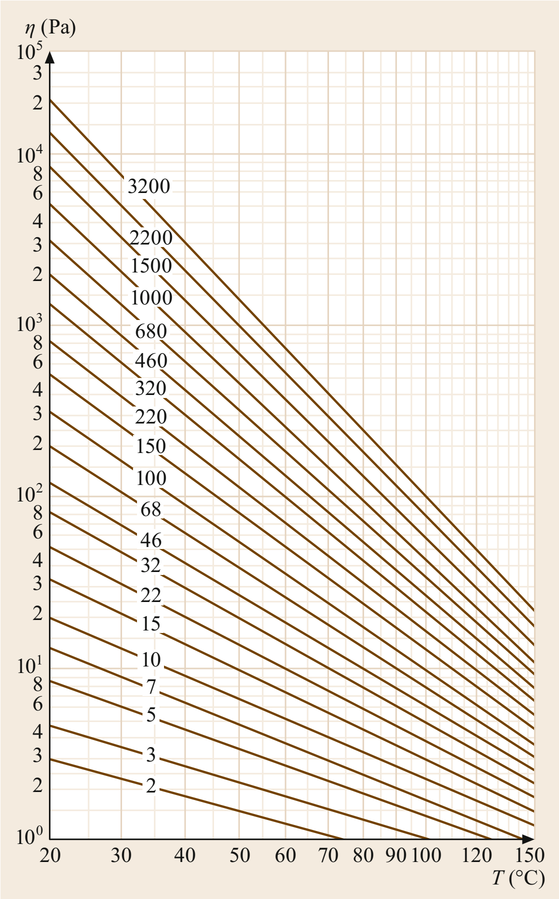


**Fig. 9.20** Dependence of the dynamic viscosity $\eta$ on the temperature $T$ for the ISO VGs (Table 9.16) at density $\rho = 900\,\mathrm{kg/m^3}$ and VI = 95 in accordance with ISO 3448

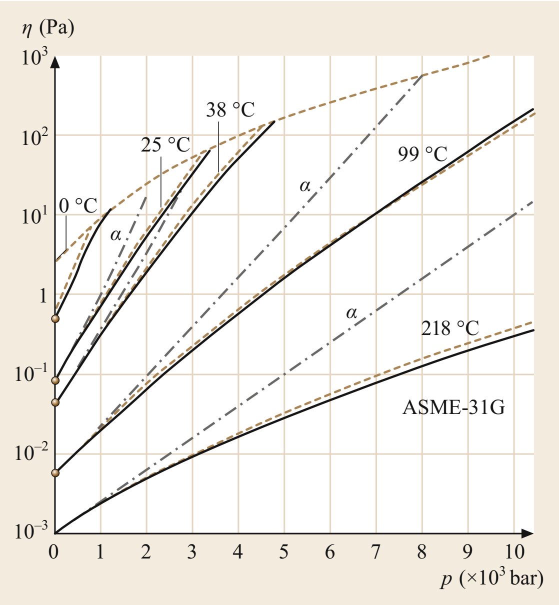


**Fig. 9.21** Viscosity of the lubricating oil ASME-31G as a function of pressure and temperature [9.21] (ASME-31G is roughly equivalent to a lubricating oil ISO VG 46). *Dash-dotted lines*: calculations based on (9.11), *dashed lines*: calculations based on (9.12), *continuous lines*: ASME measurements

In the equation, $\eta_0(T)$ is the viscosity at 1 bar and the corresponding operating temperature, $\alpha$ is the viscosity–pressure coefficient, and $p_\mathrm{u}$ is the ambient pressure; $\alpha$ is dependent on the base oil (mineral or synthetic) and the additives (Tables 9.12 and 9.17).

Reference [9.1] provides an expression that simultaneously reproduces the dependence of the dynamic viscosity $\eta$ on the state variables pressure $p$ and temperature $T$

$$\eta(T,p) = A\exp\left[\frac{B}{C+T}\left(\frac{p-p_\mathrm{u}}{2000}+1\right)^{\left(D+E\frac{B}{C+T}\right)}\right]. \tag{9.12}$$

The dependence of the dynamic viscosity on the temperature is represented by the coefficients $A$, $B$, and $C$ (Vogel equation) and the dependence on the pressure is described by the coefficients $D$ and $E$. Tests are employed to determine the coefficients $A$–$E$. Figure 9.21 presents the viscosity of a lubricating oil as a function of pressure and temperature.

**Table 9.17** Viscosity–pressure coefficient $\alpha$ and examples of increasing viscosity for different lubricants [9.20]

| Type of oil | $\alpha_{25\,°\mathrm{C}}(\times 10^8\,\mathrm{m^2/N})$ | $\frac{\eta_{2000\,\mathrm{bar}}}{\eta_0}$ at 25 °C | $\frac{\eta_{2000\,\mathrm{bar}}}{\eta_0}$ at 80 °C |
|---|---|---|---|
| Paraffin basic mineral oils | 1.5–2.4 | 15–100 | 10–30 |
| Naphthene basic mineral oils | 2.5–3.5 | 150–800 | 40–70 |
| Aromatic solvent extracts | 4.0–8.0 | 1000–200 000 | 100–1000 |
| Polyolefines | 1.3–2.0 | 10–50 | 8–20 |
| Ester oils (diester, dendritic) | 1.5–2.0 | 20–50 | 12–20 |
| Polyether oils (aliph.) | 1.1–1.7 | 9–30 | 7–13 |
| Silicon oils (aliph. subst.) | 1.2–1.4 | 9–16 | 7–9 |
| Silicon oils (arom. subst.) | 2.0–2.7 | 300 | – |
| Chlorinated paraffin (depending on level of halogenation) | 0.7–5.0 | 5–20 000 | – |

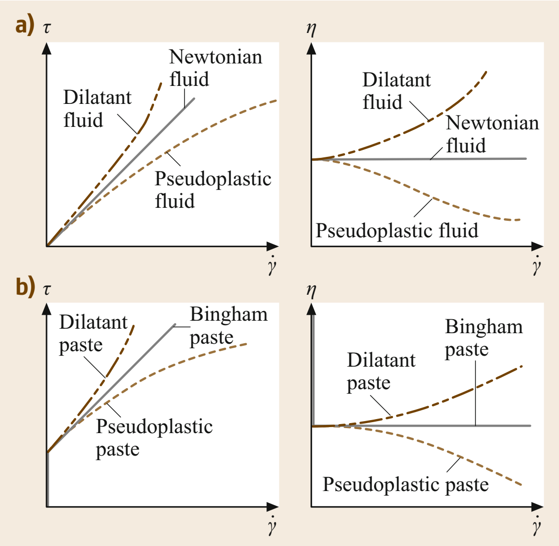


**Fig. 9.22a,b** Typical flow curves of different lubricants. **(a)** Newtonian, dilatant and pseudoplastic fluid; **(b)** Bingham paste, dilatant and pseudoplastic paste

## 9.6.6 Dependence of Viscosity on Shear Rate

When the rheological properties are independent of time, the flow properties of viscous lubricants can be easily described. Then the shear stress $\tau$ in the lubricant is a simple function of the local shear rate $\dot{\gamma}$, i.e., $\tau = f(\dot{\gamma})$. If this function is linear so that the shear stress is proportional to the shear rate, then a Newtonian fluid exists and the proportionality coefficient is the dynamic viscosity, which also remains constant when shear rates vary (Fig. 9.22a). Pure mineral oils generally exhibit Newtonian properties up to relatively high shear rates of $10^5$–$10^6\,\mathrm{s}^{-1}$. At higher shear rates, which occur relatively often in tribological contacts such as toothed gears, ball bearing, cam-follower pairs, etc., the viscosity's constancy frequently disappears and the viscosity decreases as the shear rate increases. The lubricant begins to behave like a non-Newtonian fluid, i.e., the viscosity now depends on the shear rate.

Pseudoplastic behavior, also known as shear thinning, is characterized by a decrease of viscosity as the shear rate increases (Fig. 9.22a). Dilatant fluids manifest the opposite of pseudoplastic behavior, i.e., thickening of the lubricant as the shear rate increases (Fig. 9.22a). Dilatant fluids are normally suspensions with a high solid content.

The flow properties of greases can be compared with those of a Bingham substance. In order to generate a flow, a threshold shear stress must first be overcome (Fig. 9.22b). This means that grease behaves like a solid at first. Once the threshold shear stress $\tau_0$ is exceeded, the lubricating grease then flows, for example, with constant viscosity like a Newtonian fluid or even pseudoplastically or dilatantly.

### Consistency of Lubricating Greases

The behavior of a lubricating grease is frequently described by its consistency (plasticity). Penetration according to ASTM D-217 and ASTM D-1403 is used as a characteristic. To determine the penetration, the penetration depth of a standard cone with predetermined dimensions into the surface of a lubricating grease is measured in a penetrometer after a penetration time of 5 s at a temperature of 25 °C (in units of 0.1 mm). A difference is made between unworked and worked penetration. Unworked penetration is measured in the unused lubricating grease, whereas worked penetration in measured in already sheared grease that has been worked under standardized conditions in a standard lubricating grease mixer. The higher the worked penetration, the softer the grease. Table 9.18 shows the relationship between penetration and consistency class.

**Table 9.18** NLGI consistency classes and applications of lubricating greases (after [9.2]) (NLGI – National Lubricating Grease Institute)

| NLGI class | Penetration (0.1 mm) | Consistency | Sliding bearings | Ball bearings | Centralized lubricating systems | Gears | Water pumps | Block greases |
|---|---|---|---|---|---|---|---|---|
| 000 | 445–475 | Almost fluid | | | + | + | | |
| 00 | 400–430 | Semifluid | | | + | + | | |
| 0 | 355–385 | Extra soft | | | + | + | | |
| 1 | 310–340 | Very soft | | | + | + | | |
| 2 | 265–295 | Soft | + | + | | | | |
| 3 | 220–250 | Medium | + | + | | | | |
| 4 | 175–205 | Relatively firm | | + | | | + | |
| 5 | 130–160 | Firm | | | | | + | |
| 6 | 85–115 | Very firm | | | | | | + |

+ Primary fields of application

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