Journal bearings/Sliding bearings

Operating conditions:

Advantages:
- Vibration damping, impact damping, noise damping
- not sensitive for vibrations, low operating noise level
- dust tight (if lubricated by grease)
- simple design; separable into two parts, easy to handle
- small total required space, adjustable to different design demands and conditions
- for big shaft diameters cheaper than rolling bearings
- with good lubrication very high rotational speed possible
- with good and appropriate lubrication nearly free of wear

Disadvantages:
- without sufficient lubrication very quickly failure of bearing
- in certain cases expensive and complicated lubrication system required
- high consumption of lubricant
- run-in time necessary
- surface quality of shaft surface very important

Friction conditions in sliding bearings:

(a) Hydrodynamic (surface separated)  
(b) Mixed film (intermittent local contact)  
(c) Boundary (continuous and extensive local contact)

Three basic types of lubrication. The surfaces are highly magnified.

(Slide 01): Friction conditions between a moving and non-moving surface
Case a): **hydrodynamic lubrication** *(Flüssigkeitsreibung)*

- complete separation of both surfaces by a lubricant film, the bearing pressure is entirely supported by the fluid pressure in the film generated by the relative motion of both surfaces
- film thickness: 0.008 mm - 0.020 mm
- typical friction coefficients: $\mu = 0.002 - 0.010$
  (Optimum condition: desired situation)

Case b): **mixed-film lubrication** *(Mischreibung)*

- surface roughness peaks get in contact and their is partial hydrodynamic support under certain proper conditions the surface wear can be mild
- coefficients of friction: $\mu = 0.004 - 0.10$

Case c): **boundary lubrication** *(Trockenreibung)*

- surfaces are in contact continuously by and extensively, the lubricant is continuously “smeared” over the surfaces and provides a continuously renewed absorbed surface film that reduces friction and wear
- typical coefficients of friction: $\mu = 0.05 - 0.20$

Slide 02): **Situation of a sliding bearing:**

- a fixed bearing enclosed the rotating shaft
- rotating journal (shaft) and sleeve are separated by a lubricant
- shaft and sleeve has a eccentricity
- the gap continuously narrow
- the lubricant is drawn in to this gap in the direction to the minimum thickness
- the lubricant is sheared, there is a velocity gradient in the film

Pressure and viscous forces acting on an element of lubricant. For simplicity, only $X$ components are shown.
-shear stresses in the lubricant are
Proportional to the velocity gradient

\[ \tau \sim \frac{dv}{dy} = \mu \cdot \frac{v}{h} \]

\( h \): thickness of lubricant film
\( v \): rotational speed of shaft
\( \mu \): proportional factor → dynamic viscosity
(acc. to DIN 1342 and 51550)

Measured by falling sphere viscometer (Kugelfallviskosimeter), a ball falls inside an oil filled tube:

\[ \tau = \mu \cdot \frac{v}{h} \]
\[ \eta = \mu = \left[ \frac{Ns}{m^2} \right] = 1 \text{ Pa} \cdot \text{s} \]

\[ \frac{N}{mm^2} = \left[ \frac{Ns}{mm^2} \right] \left[ \frac{m}{S.m} \right] \]

A derivate unit is often used

\[ \eta' = \mu' = \mu \cdot \frac{1}{\rho} \]
kinematic viscosity \[ \left[ \frac{m^2}{S} \right] \]
(Slide 03): Viscosity and density of oils and lubricants depending on temperature

- SAE 70: oil type (Society of Automobile Engineers)

Old dimensions for viscosity:

1 stokes = 1 St = 1 cm²/s → Kinematic viscosity

1 Centistokes = 1 cSt = 1 mm²/s = 1 Z
- **High viscosity**: Higher forces can be taken at low speeds with high loads but high friction values

- **Low viscosity**: Smaller loads can be taken but higher speeds are possible with less internal friction

Density and specific heat of oil is also temperature-depended: (acc. to VDI-2204)

\[
\rho_t = \rho_{t_0} \cdot \left(1 - 65 \cdot 10^{-5} (t - t_0)\right)
\]

\[\rho_t: \quad \text{Density at temperature } t\]
\[\rho_{t_0}: \quad \text{Density at temperature } t_0 = 15^\circ C \ ; \ t = \text{actual temperature}\]

\[C_p = 3856 - 2.345 \cdot \rho_o + 4.605 \cdot t \quad (\rho_o > 896 \text{ kg/m}^3)\]
\[C_p = 2910 - 1.290 \cdot \rho_o + 4.605 \cdot t \quad (\rho_o \leq 896 \text{ kg/m}^3)\]

(Slide 04): Geometry and pressure distribution for a journal bearing
Geometrical definitions of bearings

- length of bearing: \( l \) or \( b \)
- sleeve diameter of bearing: \( d_1, d_1 \approx d_2 \approx d \)
- shaft diameter: \( d_2 \) (d)
- clearance \( S = d_1 - d_2 \)
- related clearance: \( \psi = \frac{S}{d} = \frac{(d_1 - d_2)}{d} \approx \frac{(d_1 - d_2)}{d_2} \)
- minimum film thickness of lubricant film:
  \[
  h_0 = h_{\text{min}} = \frac{S}{2} \cdot \delta = \frac{d}{2} \cdot \psi \cdot \delta
  \]
  \( \delta \) = related thickness of lubricant film
  \[
  \delta = \frac{h_0}{S/2} = \frac{h_0}{\psi \cdot \frac{d}{2}}
  \]
- eccentricity of bearing:
  \[
  e = \frac{S}{2} - h_0 = \frac{d}{2} \cdot \psi \cdot \varepsilon
  \]
  \( \varepsilon \) = related eccentricity
  \[
  \varepsilon = \frac{e}{S/2} = 1 - \delta
  \]
- angular speed of bearing
  \[
  \omega = \frac{2\pi n \left( \text{rad} \right)}{60 \left( \text{sec} \right)} \quad n: \text{rpm} \equiv \text{revolutions per minute}
  \]
- sliding speed of bearing
  \[
  n = \pi \cdot d \cdot n' \quad n': \text{operating rpm of bearing} \quad [1/\text{s}]
  \]
- specific pressure in the bearing
  \[
  P_L = \bar{P} = \frac{F}{b \cdot d} \quad F: \text{bearing force} ; \quad b: \text{width of bearing} ; \quad d: \text{diameter of bearing}
  \]
### Design data for journal bearings

#### Journal bearing design practices

<table>
<thead>
<tr>
<th>Machinery</th>
<th>Bearing</th>
<th>Maximum pressure, $P$</th>
<th>Diameter clearance ratio, $\psi = \frac{e}{d}$</th>
<th>Viscosity, $\eta_1$</th>
<th>Viscosity, $\eta$</th>
<th>Bearing modulus (minimum), $S' = \frac{\eta_1}{P}$</th>
<th>$S'' = \frac{\eta}{P}$</th>
<th>SI Units, $x10^{-9}$</th>
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<td>Automobile and aircraft engines</td>
<td>Main</td>
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<td>20</td>
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<td>Marine steam engines</td>
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<td>0.00 1.2 20</td>
<td>20</td>
<td>20</td>
<td>24.8</td>
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<td>Main</td>
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<td>20</td>
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<td>0.00 1.5 20</td>
<td>20</td>
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<td>24.8</td>
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<td>Reciprocating pumps and</td>
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<td>0.00 1.0 20</td>
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<td>Railway cars</td>
<td>Axle</td>
<td>0.35 0.45 0.45 2.60</td>
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<td>50</td>
<td>120.9</td>
<td>241.8</td>
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<td>1.17</td>
<td>10.00</td>
<td>50</td>
<td>120.9</td>
<td>241.8</td>
<td></td>
</tr>
<tr>
<td>pumps</td>
<td>Rotor</td>
<td>0.60 0.85 0.85 5.90</td>
<td>9.00</td>
<td>0.00 1.0 25</td>
<td>25</td>
<td>25</td>
<td>200</td>
<td>483.5</td>
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<td>Gyroscope</td>
<td>Light, fixed</td>
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<td>0.00 2.5 30</td>
<td>30</td>
<td>30</td>
<td>250</td>
<td>1330</td>
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<td>Transmission shafting</td>
<td>Self-aligning</td>
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<td>9.00</td>
<td>0.00 2.5 30</td>
<td>30</td>
<td>30</td>
<td>250</td>
<td>1330</td>
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<tr>
<td>Heavy</td>
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<td>9.00</td>
<td>0.00 2.5 30</td>
<td>30</td>
<td>30</td>
<td>250</td>
<td>1330</td>
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<td>Cotton mill</td>
<td>Spindle</td>
<td>0.0007 0.001 0.0009 0.005</td>
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<td>0.00 2.5 30</td>
<td>30</td>
<td>30</td>
<td>250</td>
<td>1330</td>
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<td>40</td>
<td>96.7</td>
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<td>Punching and shearing machine</td>
<td>Main</td>
<td>2.82 4.0 4.0 27.80</td>
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<td>0.00 1.0 100</td>
<td>100</td>
<td>100</td>
<td>2417.5</td>
<td></td>
</tr>
<tr>
<td>Crankpin</td>
<td>5.62 8.0 8.0 55.60</td>
<td>100.00</td>
<td>0.00 1.0 100</td>
<td>100</td>
<td>100</td>
<td>2417.5</td>
<td></td>
<td></td>
</tr>
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<td>Rolling mills</td>
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<td>0.00 1.1 50</td>
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<td>50</td>
<td>24.2</td>
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</tbody>
</table>

Key: $\eta_1$, $\eta$: absolute viscosity, Pa s (cP); $\nu$: speed, rpm; $n$: speed, rps; $P$: pressure, N / m² or MPa (psi); MPa = megapascal = $10^6$ N / m²; Pa = Pascal = 1 N / m²; 1 psi = 6894.757 Pa; 1 ksi = 6.89475 MPa; USCSU = US Customary System units.
**Design data for journal bearings:**

- tabulated data for sliding bearings for different conditions in engineering
- determining conditions for bearings are:

  - Max. design pressure $\bar{P} = \bar{P}_L$ in MPa
  - diameter clearance ratio (related clearance)

\[
\psi = \frac{C}{d}
\]

- bearing ratio:

\[
\frac{l}{d} \quad \text{or} \quad \frac{b}{d} = \beta
\]

**Some simple bearings:**

<table>
<thead>
<tr>
<th>Bearing material</th>
<th>Lubrication by hand</th>
<th>Sliding speed u (m/s)</th>
<th>N/mm²</th>
<th>dropped oil</th>
<th>$u_{P_L}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cast iron</td>
<td>grease lubrication</td>
<td>1</td>
<td>0.4</td>
<td>3</td>
<td>0.8</td>
</tr>
<tr>
<td>Bronze, Cu – alloy</td>
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<td>2</td>
<td>0.6</td>
<td>2</td>
<td>1.2</td>
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<tr>
<td>Al – alloy</td>
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<td>2</td>
<td>0.3</td>
<td>2</td>
<td>0.4</td>
</tr>
<tr>
<td>Pb – Sn – alloy</td>
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<td>3</td>
<td>0.1</td>
<td>3</td>
<td>0.3</td>
</tr>
<tr>
<td>Sinter material</td>
<td>(oil filled)</td>
<td>1</td>
<td>1</td>
<td>3</td>
<td>1.8</td>
</tr>
</tbody>
</table>
(Slide 06): Data about bearing materials

### Bearing Alloy Material Applications

<table>
<thead>
<tr>
<th>Material</th>
<th>Nominal composition, % by weight</th>
<th>Applications and remarks</th>
</tr>
</thead>
<tbody>
<tr>
<td>Aluminum, low tin</td>
<td>Al 92 Sn 8</td>
<td>Tin added to improve compatibility; too much tin lowers strength. Has thermal expansion problems in steel housings. Requires hard journals. Good at high temperatures. Used in diesel engines and compressors.</td>
</tr>
<tr>
<td>Aluminum, high tin</td>
<td>Al 80 Sn 20</td>
<td>Produced by special working and annealing process so tin content does not greatly reduce strength. Used in automotive engines (crankshafts) and in aircraft equipment.</td>
</tr>
<tr>
<td>Babbitt, tin-based</td>
<td>Sn 84 Cu 8 Sb 8</td>
<td>Fatigue strength decreases as thickness increases. Low load capacity, thus usually bonded to one (bimetal) or two (trimetal) backing materials. Good in dirty applications, motors.</td>
</tr>
<tr>
<td>Babbitt, lead-based</td>
<td>Pb 75 Sn 10 Sb 15</td>
<td>Antimony (Sb) greater than 15% can cause brittleness. Cheaper than tin-based babbitt. Used in crankshaft bearings, transmission bushings, and electric equipment.</td>
</tr>
<tr>
<td>Lead bronze</td>
<td>Cu 70 Pb 25 Sn 5</td>
<td>Good for high-load high-speed applications; can be used with soft journals. Used as bushings in pumps, many home appliances, railroad cars.</td>
</tr>
<tr>
<td>Phosphor bronze</td>
<td>Cu 80 Sn 10 Pb 10</td>
<td>General-duty popular bushing; tin added to improve strength. Has high hardness; should be used with harder journals (300 BHN). Good impact resistance; used in lathes, pumps, home appliances.</td>
</tr>
<tr>
<td>Copper lead (cast)</td>
<td>Cu 75 Pb 25</td>
<td>Lead in pockets in copper matrix. Lead improves bearing surface but has corrosion problems. Frequently used as lining material on steel-backed bearings. Used in heavy-duty applications.</td>
</tr>
<tr>
<td>Copper lead (sintered)</td>
<td>Cu 75 Pb 25</td>
<td>Frequently used with a babbitt overlay in a trimetal bearing. Widely used in heavy-duty (high-temperature high-load) applications.</td>
</tr>
<tr>
<td>Silver (oven-plated)</td>
<td></td>
<td>Frequently used with lead indium overlay.</td>
</tr>
</tbody>
</table>

(Babbitt metal: Lagerweissmetall: Zinnlager)
Demands on bearing materials

- fatigue strength in operation
- good sliding properties
- seizure resistance (*Notlaufeigenschaft*)
- corrosion resistance
- embedability: hard partials are embedded in the soft outer layer
- compatibility: the bearing sleeve fits itself to rotating journal during run in time

### Performance Ratings from 5 (High) to 1 (Low) for Bearing Alloy Materials

<table>
<thead>
<tr>
<th>Material</th>
<th>Nominal composition, % by weight</th>
<th>Fatigue strength</th>
<th>Corrosion resistance</th>
<th>Seizure resistance</th>
<th>Embedability</th>
<th>Compatibility</th>
<th>Thermal conductivity</th>
</tr>
</thead>
<tbody>
<tr>
<td>Aluminum, low tin</td>
<td>Al 92 Sn 8</td>
<td>2</td>
<td>4</td>
<td>2</td>
<td>1</td>
<td>1</td>
<td>4</td>
</tr>
<tr>
<td>Aluminum, high tin</td>
<td>Al 80 Sn 20</td>
<td>2</td>
<td>4</td>
<td>2</td>
<td>2</td>
<td>2</td>
<td>4</td>
</tr>
<tr>
<td>Babbitt, tin-based</td>
<td>Sn 84 Cu 8 Sb 8</td>
<td>1</td>
<td>5</td>
<td>4</td>
<td>5</td>
<td>4</td>
<td>3</td>
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<tr>
<td>Babbitt, lead-based</td>
<td>Pb 75 Sn 10 Sb 15</td>
<td>1</td>
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<td>Phosphor bronze</td>
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<tr>
<td>Copper lead (cast)</td>
<td>Cu 75 Pb 25</td>
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<td>3</td>
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<tr>
<td>Copper lead (sintered)</td>
<td>Cu 60 Pb 40</td>
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<td>3</td>
<td>3</td>
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<td>Silver (over-plated)</td>
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<td>4</td>
<td>1</td>
<td>1</td>
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<td>5</td>
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(Slide 08): Bearing geometries:

Characteristics of Lubrication Regimes

<table>
<thead>
<tr>
<th>Lubrication regime</th>
<th>Contact of bearing surfaces</th>
<th>Range of film thickness, in</th>
<th>Coefficient of friction</th>
<th>Degree of wear</th>
<th>Comments</th>
</tr>
</thead>
</table>
| Thick film         | Only during startup or stopping      | $10^{-3}$ to $10^{-4}$     | 0.01–0.005              | None           | 1. Light-loading high-speed regime  
2. Friction coefficient proportional to $\mu N/[W/(LD)]$ |
| Thin film          | Intermittent; dependent on surface roughness | $10^{-4}$ to $0.5 \times 10^{-4}$ | 0.005–0.05 | Mild         | 1. High operating temperatures |
| Boundary           | Surface to surface                  | $0.5 \times 10^{-4}$ to molecular thicknesses | 0.05–0.15      | Large        | 1. Heavy-loading (unit load $> 3000$ psi) low-speed ($< 60$ fpm) operating regime  
2. Heat generation and friction not dependent on lubricant viscosity |

Journal bearing geometries. (a) Full bearing; (b) partial bearing; (c) elliptical, or lemon, bearing; (d) offset bearing; (e) rocking journal bearing; (f) pressure dam bearing; (g) three-lobe bearing; (h) four-lobe bearing; (i) multileaf bearing; (j) floating-ring bearing; (k) tilting- or pivoted-pad bearing; (l) foil bearing.

a) full bearing 360° bearing  
b) partial bearing 180° bearing
Sliding bearings / Oil film bearings:

Specific load for a sliding bearing: (mean)

\[ P = \frac{F}{D \cdot b} \]

- **F**: Bearing force (*Lagerkraft*)
- **d**: Bearing diameter (*Lagerdurchmesser*)
- **b**: Bearing width (*Lagerbreite*)
- **P**: Mean pressure

The journal rotates in the bearing with the sliding speed \( u \):

\[ u = d \cdot \pi \cdot n \]

\( n' \) : journal rps \([ 1/ s]\) \[ n' = n_M/60 \text{ , } n_M = \text{rpm} \]

Angular speed of journal (*Lagerzapfen*)

\[ \omega = 2 \cdot \pi \cdot n = \frac{2 \cdot \pi \cdot n}{60} \text{ [s}^{-1}] \]
Due to friction a certain friction power is generated in the bearing:

\[
P_f = F \cdot \mu \cdot u \left[ \frac{Nm}{s} \right] = [W]
\]

\[\mu: \text{ friction factor} \]
\[u: \text{ sliding speed} \]
\[P_f: \text{ friction power} \]

For the capacity if a sliding bearing a dimensionless factor is defined:

\[
S_0 = \frac{\bar{p} \cdot \psi^2}{2} = \frac{F}{b \cdot d} \cdot \frac{S^2}{\eta \cdot \omega \cdot d^2} = \frac{F \cdot S^2}{\eta \cdot b \cdot d^3}
\]

- \(S: \text{ diameter difference: } d_1 - d_2 \) (Lagerspiel)
- \(\bar{p}: \text{ mean pressure of bearing} \)
- \(\eta: \text{ viscosity of oil } \text{Pa} \cdot \text{s} \)
- \(d: \text{ journal diameter (bearing diameter)} \)
- \(F: \text{ bearing force} \)
- \(\omega: \text{ angular speed of journal} \)
- \(S_0: \text{ Sommerfeld-Number} \)

A lot of design data for sliding bearings are related to the Sommerfeld number \(S_0\):

The Sommerfeld number \(S_0\) can be calculated as a function of the relative eccentricity \(\varepsilon\) and the width ratio \(\beta = b/d\) of the bearing.

**Calculation of Sommerfeld number**

\[
S_0 = \left(\frac{b}{d}\right)^2 \cdot \frac{\varepsilon}{2(1-\varepsilon)^2} \cdot \sqrt{\pi^2 \left(1-\varepsilon^2\right) + 16\varepsilon^2} \cdot \frac{\alpha_1(\varepsilon - 1)}{\alpha_2 + \varepsilon}
\]

with:

\[
\varepsilon = \frac{e}{S / 2} = 1 - \delta \quad \text{with} \quad \delta = \frac{h_0}{S / 2} \quad \text{and} \quad e = \frac{S}{2} - h_0 = \frac{d}{2} \cdot \psi \cdot \varepsilon
\]

\(\alpha_1 \text{ = f(b/d)} \) and \(\alpha_2 \text{ = f(b/d)} \)
Sommerfeldzahl $S_o$ in Abhängigkeit von der relativen Exzentrizität $\varepsilon$ und von der relativen Lagerbreite $B/D$ nach DIN 31652-2

$$S_o = \left(\frac{B}{D}\right)^2 \cdot \frac{\varepsilon}{2 \cdot (1-\varepsilon^2)} \cdot \sqrt{\pi^2 \cdot (1-\varepsilon^2) + 16 \cdot \varepsilon^2 \cdot \frac{a_1 \cdot (\varepsilon-1)}{a_2 + \varepsilon}}$$

mit:

$$a_1 = 1.1842 - 1.9456 \cdot \left(\frac{B}{D}\right)^2 + 7.1161 \cdot \left(\frac{B}{D}\right)^4 - 10.1073 \cdot \left(\frac{B}{D}\right)^3 + 5.0141 \cdot \left(\frac{B}{D}\right)^4$$

$$a_2 = -1.000026 - 0.023634 \cdot \left(\frac{B}{D}\right)^2 - 0.4215 \cdot \left(\frac{B}{D}\right)^2 - 0.03882 \cdot \left(\frac{B}{D}\right)^3 - 0.09051 \cdot \left(\frac{B}{D}\right)^4$$

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(Slide 10)

Calculation acc. to DIN31652-2 (for full bearings) (see also slide 10a,b)
The Sommerfeld number $S_o = f(\varepsilon, b/d_L)$ with pure rotation
- heavy loaded bearings → big $S_0$-numbers
- low loaded bearings → small $S_0$-numbers

Practically: $S_0 > 1$ ; only with high sliding speeds and low bearing loads $S_0 > 1 !}$
Some practical values for the minimum oil film thickness in the bearing $h_{\text{omin}}$

(Slide 11): Experimental data for $h_{\text{omin}}$[µm] acc. to DIN 31652 T3 for different shaft diameter and sliding speeds

Empirical value of smallest allowable $h_{\text{omin}}$ acc. to DIN 31652 T3

<table>
<thead>
<tr>
<th>Wellendurchmesser $d$ in mm</th>
<th>Gleitgeschwindigkeit der Welle $u$ in m/s</th>
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<tr>
<td>über 24 bis 63</td>
<td>1 3 5 7 9 11 13 16</td>
</tr>
<tr>
<td>63</td>
<td>4 5 7 9 11 13 16</td>
</tr>
<tr>
<td>160</td>
<td>6 7 9 11 13 16</td>
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<td>400</td>
<td>8 9 11 13 16</td>
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<tr>
<td>1000</td>
<td>10 12 14 16</td>
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The value $h_0$ for the smallest gap in the bearing is given by 2 formulas:

For $S_0 > 1$:  
\[ h_0 = \frac{S}{2} \cdot \frac{1}{2 \cdot S_0} \cdot \frac{2\beta}{1 + \beta} \]

For $0 < \beta \leq 2$ with $\beta = b/d$ (width ratio of bearing)

And for $S_0 < 1$:
\[ h_0 = \frac{S}{2} \left[ 1 - \frac{S_0}{2} \cdot \frac{1 + \beta}{2\beta} \right] \]

For $0.5 \leq \beta \leq 2$

These formulas give good approximation for $h_0$

During operation of the bearing is necessary:

\[ h_0 \geq h_{\text{omin}} \]

The rotational speed $n_{\text{min}}$, which is necessary to ensure $h_{\text{omin}}$ is given by:

\[ n_{\text{min}} = \frac{h_{\text{omin}} \cdot n}{h_0} \quad [1/\text{s}] \]

Minimum rotational speed

The value of $n$ is calculated by use of the $S_0$-number for $S_0 < 1$

\[ n(S_0 = 1) = \frac{\bar{P} \cdot \psi^2}{\eta \cdot 2\pi} \]
The transition rotational speed $n_T$ is given by:

$$n_T = \frac{h_{0T}}{h_0} \cdot n \quad \text{for} \quad h_0 > h_{0\text{min}} > h_{0T}$$

A thumb-rule equation for the relative bearing clearance $\psi$ is given by:

$$\psi = \frac{S}{d} = \frac{d_1 - d_2}{d_1} \approx 0.8\sqrt{u} \quad u: \text{sliding speed}$$

Where $\psi$ is given $1/1000$ (promille)

Practical conditions are:
- Small $\psi$-values for high speeds and loads, greater $\psi$-values for small loads and high speeds
- Some values for $\psi$ in relation to ISO-fits for shafts and bearings are given in (Slide 12a,b)

Zuordnung von Toleranzfeldern von ISO-Passungen zu relativen Lagerspielen $\psi$ in %
(nach VDI 2201)
Any sliding bearing do have friction in the oil film in the bearing gap.

Some thumb rules for a related friction factor is:

For $S_0 < 1$ (high speed bearing) \[ \frac{\mu}{\psi} = \frac{3}{S_0} \]

For $S_0 > 1$ (high load bearing) \[ \frac{\mu}{\psi} = \frac{3}{\sqrt{S_0}} \]
See also (slide 13): Vogelpohl diagram

Diagramm nach Vogelpohl
(Slide 13)
The related friction factor can also be calculated acc. to DIN 31652-2 to a formula system: (see slide 14)
The related friction factor $\mu/\psi_{eff}$ depending on the related excentricity $\varepsilon$ and the related bearing width $B/D$ acc. to DIN 31652-2

Bezogene Reibzahl $\mu/\psi_{eff}$ in Abhängigkeit von der relativen Exzentrizität $\varepsilon$ und von der relativen Lagerbreite $B/D$ nach DIN 31652-2

$$\frac{\mu}{\psi_{eff}} = 10^Y \quad \text{mit} \quad Y = C + E \cdot (\log S_0) + F \cdot (\log S_0)^2 + G \cdot (\log S_0)^3 + H \cdot (\log S_0)^4$$

where $Y = f$ (factor: based on $S_0$ and b/d - ratio)

With this data the friction power is calculated

$$P_f = \mu \cdot F \cdot u = \mu \cdot F \cdot \frac{d}{2} \cdot \omega$$
The total friction power generates heat in the bearing, this heat must transferred out of the bearing partly by convection through the bearing housing and partly by the oil flow through the bearing:

\[ P_f = P_A + P_Q \]

- \( P_A \): cooling power by convection
- \( P_Q \): cooling power by oil flow

- bearings without pressured flow of oil: \( \rightarrow \) convection
- bearings with pressured oil flow: \( \rightarrow \) oil cooling

If \( P_f = P_A \)

\[ \mu \cdot F \cdot u = \alpha^* \cdot A \cdot (v - u_0) \]

\( \alpha^* \): heat transition factor \[\begin{bmatrix} \text{Nm} \\ \text{s} \cdot \text{m}^2 \text{K} \end{bmatrix} = \begin{bmatrix} \text{W} \\ \text{m}^2 \text{K} \end{bmatrix} \] by surrounding air

\[ \alpha^* = 7 + 12 \sqrt{V_w} \]

with \( V_w \): air speed \( [\text{m/s}] \)

\[ V_w \approx 1.25 \text{m/s} \]

Max. values for \( \alpha^* \) are 20 \[\begin{bmatrix} \text{W} \\ \text{m}^2 \text{K} \end{bmatrix} \] and \( A \): surface area of bearing housing

For a cylindrical bearing housing: (DIN 31652)

\[ A = \frac{\pi}{2} \left( d_H^2 - d_B^2 \right) + \pi \cdot d_H \cdot b_H \]

- \( d_H \): diameter of housing
- \( d_B \): bearing diameter
- \( b_H \): width of housing

For a pedestal bearing (Stehlager)

\[ A \approx \pi \cdot H \cdot \left( b_H + \frac{H}{2} \right) \]

\( H \): height of pedestal
A bearing in a machine compound:

\[ A = (15 \div 20) \cdot d_B \cdot b_B \]

\[ \theta_0 : \text{ambient temperature (} \approx 20^\circ\text{C)} \]
\[ \theta : \text{max. bearing temperature} = 70^\circ\text{C} \div 100^\circ\text{C} \]

If cooling is done by oil flow, the necessary oil flow is:

\[ Q_c = \frac{P_f}{c \cdot \rho \cdot (\theta_2 - \theta_1)} \]
\[ Q_c : \text{oil flow} \left[ \frac{m^3}{s} \right] \]

\( c: \) specific heat of oil = \( f(\theta) \)
\( \rho: \) density of oil = \( f(\theta) \)
\( \theta_2: \) run out temperature of oil
\( \theta_1: \) run in temperature of oil

\( (\theta_2 - \theta_1) \) should be \( \approx 10 \div 20 \text{ K} \)

For regular lubrication conditions in a oil film bearing a certain necessary oil flow is required in the bearing:

\[ Q_L \approx \varphi \cdot h_0 \cdot b \cdot \frac{u}{2} \]

\( \varphi: \) oil flow rate factor \( \approx 1,5 \)
\( h_0: \) min. oil film thickness
\( b: \) bearing width
\( u: \) sliding speed

Acc. to DIN31652 the calculation is as follows:

\[ Q_L = Q_1 + Q_2 \]

Where:

\[ Q_1 = d^3 \cdot \varphi \cdot \omega \cdot q_1 \]
\[ q_1 = \frac{1}{4} \left[ \left( \frac{b}{d} \right) - 0.223 \cdot \left( \frac{b}{d} \right)^3 \right] \cdot \varepsilon \]
\( q_1: \) related oil flow due to internal bearing pressure

\[ Q_2 \text{ is:} \]

\[ Q_2 = \left( d^3 \cdot \psi^2 \cdot P_E / \eta \right) \cdot q_2 \quad \text{(see slide 15)} \]
The related oil flow $q_1$ due to intrinsic pressure in the gap a relation of related eccentricity $\varepsilon$ and related bearing ratio $B/D$ acc. to DIN 31652

Bezogener Schmierstoffdurchsatz $q_1$ infolge Eigendruckentwicklung im Schmierspalt in Abhängigkeit von der relativen Exzentrizität $\varepsilon$ und der relativen Lagerbreite $B/D$ nach DIN 31652-2

\[
q_1 = \frac{1}{4} \left[ \left(\frac{B}{D}\right) - 0.233 \cdot \left(\frac{B}{D}\right)^2 \right] \cdot \varepsilon
\]

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<th>$\varepsilon$</th>
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<th>0.4</th>
<th>0.333</th>
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<td>0.0406</td>
<td>0.0305</td>
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<tr>
<td>0.990</td>
<td>0.1910</td>
<td>0.1174</td>
<td>0.0959</td>
<td>0.0808</td>
<td>0.0612</td>
<td>0.0491</td>
<td>0.0410</td>
<td>0.0308</td>
</tr>
<tr>
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<td>0.1919</td>
<td>0.1180</td>
<td>0.0965</td>
<td>0.0812</td>
<td>0.0615</td>
<td>0.0494</td>
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<td>0.0310</td>
</tr>
<tr>
<td>0.999</td>
<td>0.1926</td>
<td>0.1185</td>
<td>0.0969</td>
<td>0.0816</td>
<td>0.0618</td>
<td>0.0496</td>
<td>0.0414</td>
<td>0.0311</td>
</tr>
</tbody>
</table>

Where $P_e$ is the entry oil pressure from lubrication system and for $q_2$ see (slide 16)

With the oil flow rate the bearing temperature is controlled, if the temperature is too high bearing conditions must be changed!!
The related oil rate $q_2$ depending on the series of the lubricant entries

Bezogener Schmierstoffdurchsatz $q_2$ in Abhängigkeit von der Anordnung der Schmierstoff-Zuführungselemente nach DIN 31652-2

$$q_2 = \frac{\pi}{28 \ln \left(\frac{d_1}{d_2}\right)}$$

zu 5

$$q_2 = \frac{1}{48} \left( \varphi_E - \varphi_A \right) \cdot \left( 1 + 1.5 \cdot \varepsilon^2 \right) \cdot \left( 3 \cdot \varepsilon^2 + \varepsilon \right) \cdot \left( \sin \varphi_E - \sin \varphi_A \right) + 0.75 \cdot \varepsilon^2 \cdot \left( \sin 2\varphi_E - \sin 2\varphi_A \right) - \frac{\varepsilon}{2} \cdot \left( \sin^3 \varphi_E - \sin^3 \varphi_A \right)$$

$$q_{n1} = 1.204 + 0.368 \cdot \left( \frac{d_1}{B} \right) - 1.046 \cdot \left( \frac{d_2}{B} \right)^2 + 1.942 \cdot \left( \frac{d_4}{B} \right)^3$$

$$q_{n2} = 1.188 + 1.582 \cdot \left( \frac{b_c}{B} \right) - 2.585 \cdot \left( \frac{b_a}{B} \right)^2 + 5.563 \cdot \left( \frac{b_a}{B} \right)^3$$

gültig für $0.05 \leq \left( \frac{b_c}{B} \right) \leq 0.7$

(Slide 16)