Chapter 7: Journal Bearings

Sliding Bearings

Sliding bearings require direct sliding of the load-carrying member on its support, as distinguished from rolling-element bearings, where balls or rollers are present between the surfaces.

Sliding bearings (also called plain bearings) are of two types:

- Journal or sleeve bearings, which are cylindrical and support radial loads (those perpendicular to the shaft axis)
- Thrust bearings, which are generally flat and, in the case of a rotating shaft, support loads in the direction of the shaft axis.

Comparison of sliding and rolling bearings

Advantages of sliding bearings over rolling bearings:

- rolling bearings may eventually fail from fatigue
- rolling bearings require more space in radial direction
- damping ability in rolling bearings is less than that in sliding bearings
- rolling bearings have higher noise
- rolling bearings have more severe alignment requirements
- rolling bearings are more expensive than sliding bearings

Comparison of sliding and rolling bearings

Advantages of rolling bearings over sliding bearings:

- rolling bearings have low starting and operating friction
- rolling bearings can support combined radial and thrust loads
- rolling bearings are less sensitive to interruptions of lubrication
- rolling bearings have no self-excited instabilities
- rolling bearings have good low-temperature starting
- rolling bearings typically require less space in axial direction

Types of Lubrication

- Hydrodynamic
- Hydrostatic
- Elastohydrodynamic
- Boundary
- Solid film

Hydrodynamic Lubrication

- Also called full-film or fluid lubrication
- Relatively thick film of lubricant - surface contact is prevented
- Does not depend on external pressurization of the lubricant
- Film pressure is created by moving surface pulling the lubricant into a wedge-shaped zone
- Requires sufficient velocity to create pressure to separate the surfaces

Hydrostatic Lubrication

- Lubricant is introduced under pressure to separate the surfaces with a relatively thick film
- Does not require motion
- Used when velocities are small or zero

Elastohydrodynamic Lubrication (EHL)

- Surface deformations play a key role in the development of a lubricant film
- Hard EHL occurs when a lubricant is between surfaces in rolling contact, such as with roller bearings or gears
  ➥ The lubricant region is of the order predicted by Hertzian contact mechanics, and is thus much smaller than the bodies themselves
- Soft EHL occurs when the contact region extends over a substantial portion of the mating surfaces

Boundary Lubrication

- When conditions do not provide a thick enough lubrication film for hydrodynamic lubrication
- The highest asperities of the surfaces may be separated by only a few molecules of lubricant
- Mixed condition of hydrodynamic and boundary lubrication often exists
- Viscosity of lubricant is not as important as chemical composition

Solid-Film Lubrication

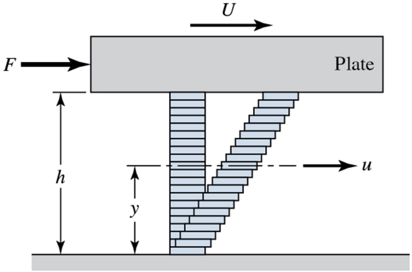
- Use of graphite or molybdenum disulfide to create a layer of solid material
- Often used at extreme temperatures

Viscosity

Shear stress in a fluid is proportional to the rate of change of velocity perpendicular to the shear stress direction:

$$\tau = \mu \frac{\mathrm{d}u}{\mathrm{d}y},$$

where μ is called the dynamic or absolute viscosity.

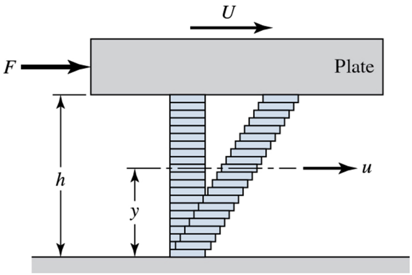


Viscosity

For most fluids, the rate of shear is constant, thus:

$$\mu \frac{\mathrm{d}u}{\mathrm{d}y} = \frac{U}{h}$$

Fluids exhibiting this behaviour are called Newtonian fluids.

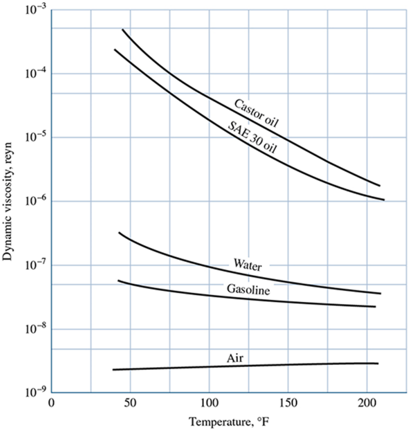


Units of Dynamic Viscosity

- Imperial units: reyn → lbf·s/in^2
- SI units: Pa·s → N·s/m^2
- CGS units: Poise → dyn·s/cm^2

Use of CGS units is discouraged, but historically it is common. Viscosity in CGS is often expressed in centipoise (cP), designated by Z. In Imperial units, the microreyn (μreyn) is often convenient.

Viscosity-Temperature Relation



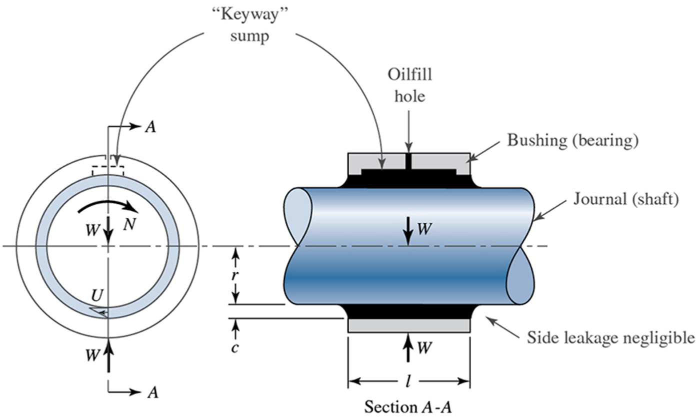
Viscosity-Temperature Relation

Approximate curve-fit:

$$\mu = \mu_0 \exp\left[\frac{b}{T + 95}\right], \qquad T \text{ in } {}^\circ\mathrm{F}$$

Oil Grade, SAE	Viscosity μ_0, reyn	Constant b, °F
10	$0.0158(10^{-6})$	1157.5
20	$0.0136(10^{-6})$	1271.6
30	$0.0141(10^{-6})$	1360.0
40	$0.0121(10^{-6})$	1474.4
50	$0.0170(10^{-6})$	1509.6
60	$0.0187(10^{-6})$	1564.0

Petroff's Lightly-Loaded Journal Bearing



Petroff's Equation

$$\tau = \mu \frac{U}{h} = \frac{2\pi r \mu N}{c}$$

$$T = (\tau A)(r) = \left(\frac{2\pi r \mu N}{c} (2\pi r l) \right) (r) = \frac{4\pi^2 r^3 l \mu N}{c}$$

$$P = \frac{W}{2rl}$$

$$\frac{T}{r} = fW \implies T = f(2rlP)(r) = 2r^2 f l P$$

$$f = 2\pi^2 \left(\frac{\mu N}{P} \right) \left(\frac{r}{c} \right)$$

Important Dimensionless Parameters

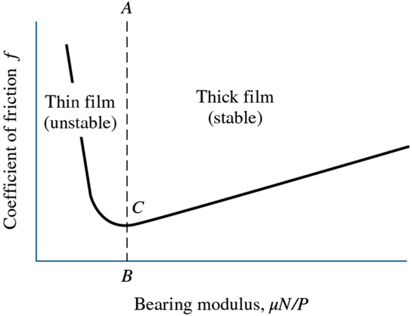
- $\frac{r}{c}$: radial clearance ratio
- $\frac{\mu N}{P}$
- $S = \left(\frac{r}{c}\right)^2 \frac{\mu N}{P}$: Sommerfeld number or bearing characteristic number

Referring to the previous slide, if we multiply the coefficient of friction with r/c, then we obtain:

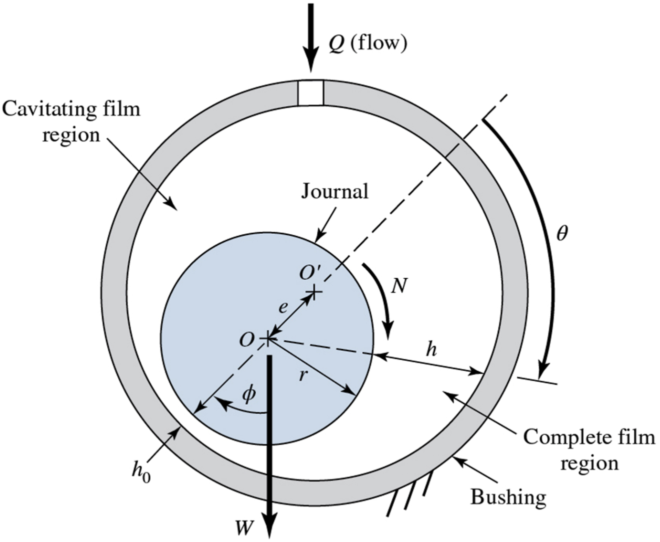
$$f\frac{r}{c} = 2\pi^2 \frac{\mu N}{P} \left(\frac{r}{c}\right)^2 = 2\pi^2 S$$

Stable vs Unstable Lubrication

- To the right of AB, changes in conditions are self-correcting and results in stable lubrication.
- To the left of AB, changes in conditions tend to get worse and results in unstable lubrication
- Point C represents the approximate transition between metal-to-metal contact and thick film separation of the parts
- Common design constraint for point B: $\dfrac{\mu N}{P} \geq 1.7 \times 10^{-6}$

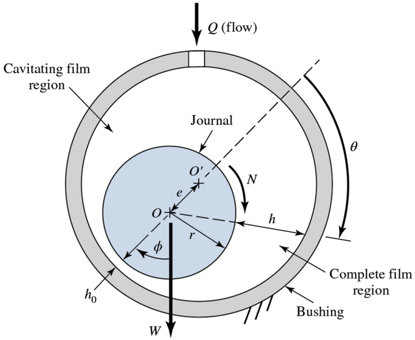


Thick film lubrication



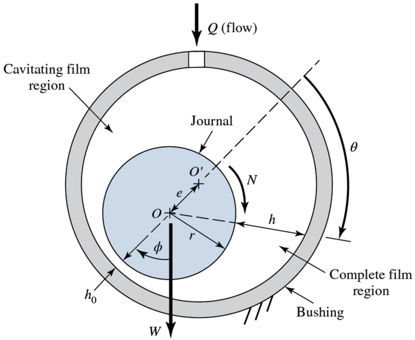
Thick film lubrication

- Centre of journal at O
- Centre of bushing at O'
- Eccentricity is e
- Radial clearance c is difference of journal and bushing radii
- Minimum film thickness h_0 occurs at line of centres
- Film thickness anywhere is $h = c + e\cos\theta$
- Eccentricity ratio: $\varepsilon = \dfrac{e}{c}$



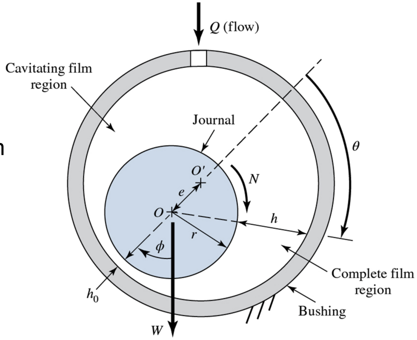
Thick film lubrication

- Steady load W applied to journal rotating CW at a steady speed N
- Attitude angle ϕ is the angle between the eccentricity and the load vector
- When a lubricant is introduced at the top, the rotation pulls the lubricant into a convergent wedge-shaped space and forces the journal centre to a steady position to the side



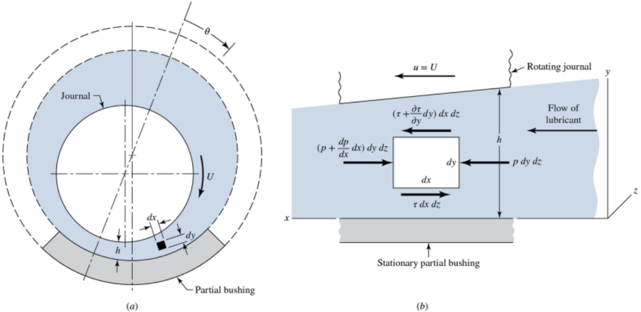
Thick film lubrication

- The minimum film thickness h_0 occurs, not along the load line, but a point displaced clockwise from the load
- The load is supported in a region of complete lubricant film, from the inlet region to a point just beyond the minimum film thickness
- As the journal and bushing surfaces diverge, significant pressure drop occurs
- The fluid film ruptures or cavitates into liquid streamers separated by a liquid-gas mixture



Reynolds[1] Plane Slider Simplification

- Reynolds realized fluid films were so thin in comparison with bearing radius that curvature could be neglected
- Replaced curved bearing with flat bearing
- Called plane slider bearing



[1]Note: It is "Reynolds", NOT "Reynold's"

Derivation of Velocity Distribution

From Fluid Mechanics:

$$\mu \frac{\partial^2 u}{\partial y^2} = \frac{\mathrm{d}p}{\mathrm{d}x}$$

$$\text{or, } \frac{\partial u}{\partial y} = \frac{1}{\mu}\frac{\mathrm{d}p}{\mathrm{d}x} y + C_1$$

$$\text{or, } u = \frac{1}{2\mu}\frac{\mathrm{d}p}{\mathrm{d}x} y^2 + C_1 y + C_2$$

Boundary conditions: At $y = 0, u = 0$ and at $y = h, u = U$

We obtain: $C_1 = \dfrac{U}{h} - \dfrac{h}{2\mu}\dfrac{\mathrm{d}p}{\mathrm{d}x}$ and $C_2 = 0$

Thus: $u = \dfrac{1}{2\mu}\dfrac{\mathrm{d}p}{\mathrm{d}x}(y^2 - hy) + \dfrac{U}{h} y$

Derivation of Reynolds Equation

Volumetric flow rate: $Q = \int_0^h u \, \mathrm{d}y = \frac{Uh}{2} - \frac{h^3}{12\mu}\frac{\mathrm{d}p}{\mathrm{d}x}$

Note that the volumetric flow rate has no variation in the x-direction:

$$\frac{\mathrm{d}Q}{\mathrm{d}x} = 0,$$
$$\implies \frac{\mathrm{d}}{\mathrm{d}x}\left(\frac{h^3}{\mu}\frac{\mathrm{d}p}{\mathrm{d}x}\right) = 6U\frac{\mathrm{d}h}{\mathrm{d}x}$$

This is the classical Reynolds equation for one-dimensional flow neglecting side leakage.

Reynolds Equation

With side leakage included, the Reynolds equation is:

$$\frac{\partial}{\partial x}\left(\frac{h^3}{\mu}\frac{\partial p}{\partial x}\right)+\frac{\partial}{\partial z}\left(\frac{h^3}{\mu}\frac{\partial p}{\partial z}\right)=6U\frac{\partial h}{\partial x}$$

For curved partial bearing with $x = r\theta$ and $U = r\omega_j$, the Reynolds equation is:

$$\frac{1}{r^2}\frac{\partial}{\partial \theta}\left(\frac{h^3}{12\mu}\frac{\partial p}{\partial \theta}\right)+\frac{\partial}{\partial z}\left(\frac{h^3}{12\mu}\frac{\partial p}{\partial z}\right)=\frac{\omega_j}{2}\frac{\partial h}{\partial \theta}$$

Important: There are no general analytical solutions for the Reynolds equation.

Design Considerations

Variables either given or under control of designer:

- The viscosity μ
- The load W
- The speed N
- The bearing dimensions

Dependent variables or performance metrics:

- The coefficient of friction f
- The temperature rise ΔT
- The volume flow rate of oil Q
- The minimum film thickness h_0

Trumpler's Design Criteria

Trumpler, a well-known bearing designer, recommended a set of design criteria:

- Minimum film thickness to prevent accumulation of ground-off surface particles

$$h_0 \geq 0.00508 + 0.00004d$$

  where d is the journal diameter in mm.

- Maximum temperature to prevent vapourization of lighter lubricant components

$$T_{\mathrm{max}} \leq 121^{\circ}\mathrm{C}$$

- Maximum starting load to limit wear at startup when there is metal-to-metal contact

$$\frac{W_{\mathrm{st}}}{lD} \leq 2068 \text{ kPa}$$

- Minimum design factor on running load: $n_d \geq 2$

Relations among the variables

Albert Raymondi and John Boyd used an interation technique to solve Reynolds equation.

Assumptions:

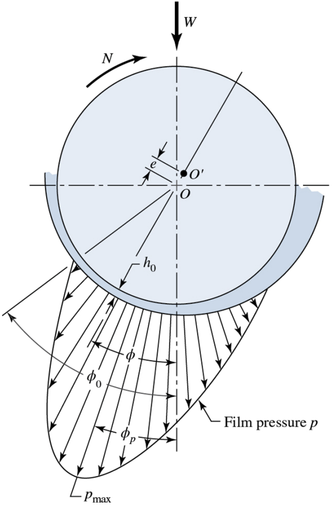
- Infinitely long bearings → no side leakage
- Full bearing
- Oil film is ruptured when film pressure becomes zero

Additionally, Raymondi and Boyd assumed constant viscosity through the loading zone

➥ Not completely true since temperature rises as work is done on the lubricant passing through the loading zone

➥ Use average temperature to find a viscosity

Pressure distribution



Polar diagram of the film pressure distribution showing notation used by Raimondi and Boyd

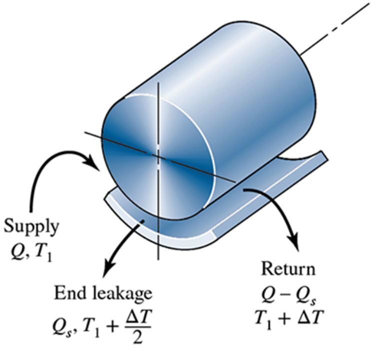
Raimondi and Boyd Charts

There are several Raimondi and Boyd charts:

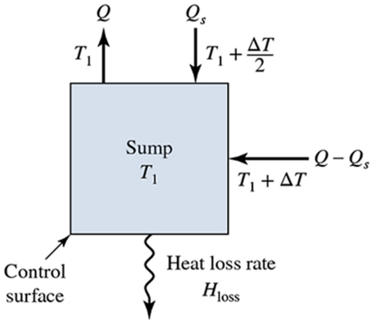
- For minimum film thickness
- For coefficient of friction
- For lubricant flow
- For film pressure

Temperature rise from energy considerations

The temperature of the lubricant rises until the rate at which work is done by the journal on the film through fluid shear is the same as the rate at which heat is transferred to the surroundings.



(a)



(b)

Temperature rise from energy considerations

Q : volumetric oil flow rate into the bearing, m^3/s

Q_s : volumetric side flow leakage rate out of the bearing into the sump, m^3/s

$Q - Q_s$: volumetric oil flow discharge from annulus to sump, m^3/s

T_1 : oil inlet temperature (equal to sump temperature, T_s in °C

ΔT : temperature rise in oil between inlet and outlet, °C

ρ : lubricant density, kg/m^3

C_p : specific heat capacity of lubricant, kJ/(kg·°C)

H : enthalpy change → effectively, the heat transfer rate, J/s or W

Temperature rise from energy considerations

$$
\begin{aligned}
H_{\text{loss}} &= \rho C_p Q_s \Delta T/2 + \rho C_p (Q - Q_s)\Delta T \\
&= \rho C_p Q \Delta T \left(1 - 0.5\frac{Q_s}{Q}\right)
\end{aligned}
$$

The thermal energy loss at steady state is equal to the rate the journal does work on the film, $H_{\text{loss}} = T(2\pi N)$. The torque $T = fWr$, the load in terms of pressure is $W = P(2rl)$. Then, multiplying the numerator and denominator by the clearance c, we have:

$$
H_{\text{loss}} = (4\pi PrlNc)\left(\frac{rf}{c}\right)
$$

Therefore, we obtain:

$$
\frac{\rho C_p \Delta T}{4\pi P} = \frac{rf/c}{(1 - 0.5Q_s/Q)\left[Q/(rcNl)\right]}
$$

Temperature rise from energy considerations

For common petroleum lubricants, $\rho = 862$ kg/m^3, $C_p = 1758$ J/(kg $\cdot^\circ$ C). So, the lhs in the previous slide becomes:

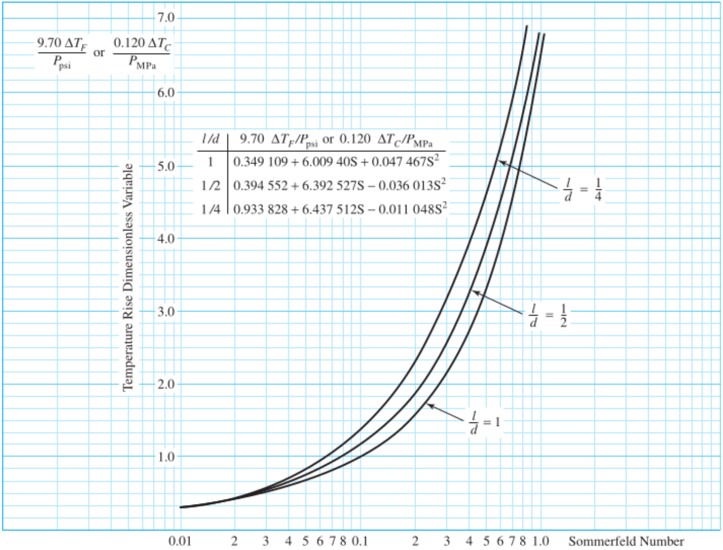
$$\frac{\rho C_p \Delta T}{4\pi P} = 0.12 \frac{\Delta T_C}{P_{\mathrm{MPa}}}$$

Thus, we have: $\dfrac{0.12\Delta T_C}{P_{\mathrm{MPa}}} = \dfrac{rf/c}{(1 - 0.5Q_s/Q)[Q/(rcNl)]}$

Here, ΔT_C is the temperature rise in $^\circ$C and P_{MPa} is the bearing pressure in MPa. Alternatively, this lhs can be written as $9.70\Delta T_F/P_{\mathrm{psi}}$, where ΔT_F is in $^\circ$F and P_{psi} is in psi.

The rhs above can be evaluated using various Raimondi-Boyd plots corresponding to different Sommerfeld numbers and l/d ratios to obtain the plot in the next slide.

Temperature rise from energy considerations



Temperature rise from energy considerations

For l/d ratios other than ones given in the plot, Raimondi and Boyd have provided the following interpolation equation:

$$
\begin{aligned}
y = \frac{1}{(l/d)^3} \Bigg[& -\frac{1}{8}\left(1-\frac{l}{d}\right)\left(1-2\frac{l}{d}\right)\left(1-4\frac{l}{d}\right) y_\infty \\
& +\frac{1}{3}\left(1-2\frac{l}{d}\right)\left(1-4\frac{l}{d}\right) y_1 \\
& -\frac{1}{4}\left(1-\frac{l}{d}\right)\left(1-4\frac{l}{d}\right) y_{1/2} \\
& +\frac{1}{24}\left(1-\frac{l}{d}\right)\left(1-2\frac{l}{d}\right) y_{1/4} \Bigg]
\end{aligned}
$$

where y is the desired variable within the interval $\infty > l/d > 1/4$, and y_∞, y_1, $y_{1/2}$, and $y_{1/4}$ are the variables corresponding to l/d ratios of ∞, 1, $1/2$, and $1/4$, respectively.

Steady-state conditions in self-contained bearings

- Previous analysis assumes lubricant carries away all enthalpy increase
- Bearings in which warm lubricant stays within bearing housing are called self-contained bearings
- Heat is dissipated from the housing to the surroundings

Heat dissipated from bearing housing

Heat given up by bearing housing:

$$H_{\text{loss}} = h_{\text{CR}} A (T_b - T_\infty)$$

where

H_{loss} : heat dissipated, J/s or W

h_{CR} : combined overall coefficient of radiation and convection heat transfer, $\text{W}/(\text{m}^2 \cdot {}^\circ\text{C})$

A : surface area of bearing housing, m^2

T_b : surface temperature of the housing, $^\circ$C

T_∞ : ambient temperature, $^\circ$C

Overall coefficient of heat transfer

The overall coefficient of radiation and convection heat transfer depends on material, surface coating, geometry, roughness, temperature difference between housing and surroundings, and air velocity. Some representative values, in units of W/(m$^2 \cdot {}^\circ$C):

$$h_{\mathrm{CR}} = \begin{cases} 11.4 & \text{for still air} \\ 15.3 & \text{for shaft-stirred air} \\ 33.5 & \text{for air moving at 24.5 m/s} \end{cases}$$

Difference between housing and ambient temperature

If one defines $\overline{T}_f$ as the average film temperature (halfway between the lubricant inlet temperature T_s and the outlet temperature $T_s + \Delta T$, i.e. $\overline{T}_f = T_s + \frac{\Delta T}{2}$), then the following proportionality has been observed between $\overline{T}_f - T_b$ and the difference between the housing surface temperature and the ambient temperature, $T_b - T_\infty$:

$$\overline{T}_f - T_b = \alpha(T_b - T_\infty)$$

where α is a constant depending on the lubrication scheme and the bearing housing geometry.

Lubrication System	Conditions	Range of α
Oil ring	Moving air	1–2
	Still air	$\frac{1}{2}$–1
Oil bath	Moving air	$\frac{1}{2}$–1
	Still air	$\frac{1}{5}$–$\frac{2}{5}$

Housing temperature

Solving for T_b from the previous slide, we obtain:

$$T_b = \frac{\overline{T}_f + \alpha T_\infty}{1 + \alpha}$$

Substituting this expression of T_b in the H_{loss} expression, we obtain:

$$H_{\mathrm{loss}} = \frac{h_{\mathrm{CR}} A}{1 + \alpha} (\overline{T}_f - T_\infty)$$

Heat generation rate

The heat generation rate H_{gen}, at steady state, is equal to the work rate from the frictional torque, $T = 4\pi^2 r^3 l \mu N / c$.

$$H_{\mathrm{gen}} = \frac{4\pi^2 r^3 l \mu N}{c}(2\pi N) = \frac{248 \mu N^2 l r^3}{c}.$$

We equate this H_{gen} to the H_{loss} obtained earlier, and solve for $\overline{T}_f$ to obtain:

$$\overline{T}_f = T_\infty + 248(1+\alpha)\frac{\mu N^2 l r^3}{h_{\mathrm{CR}} A c}$$

Challenge

Note:

- The H_{gen} calculated as above depends on μ which, in turn, depends on the temperature, and the temperature is not known!
- The H_{gen} can also be calculated as in the previous topic, i.e. directly in terms of the load or the pressure, which however involves f. Now, f can be found provided we know the Sommerfeld number, but this number can be determined only if we know μ!
- So, the temperature has to be given or guessed → **Iterative process**.