Fluid Film Bearings

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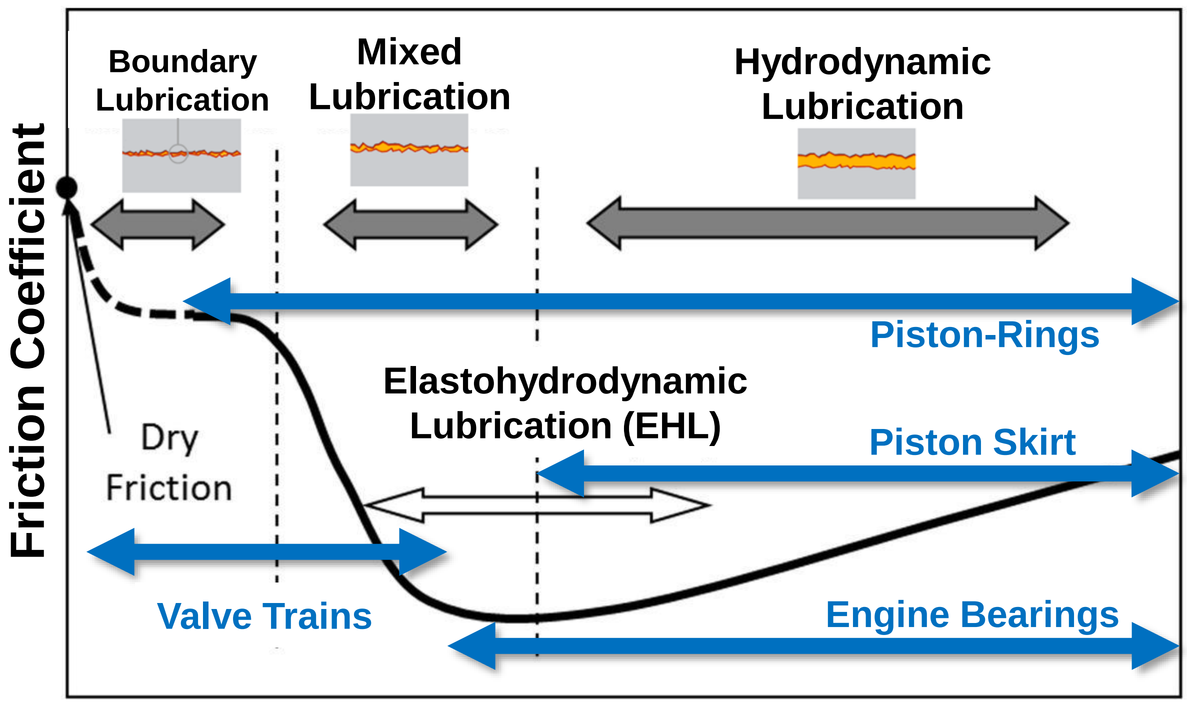
Outline

1. Introduction

1.1 Lubrication Regimes (Stribeck curves)

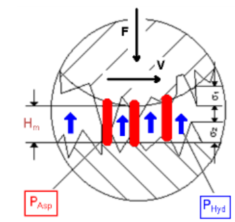
Engine Friction Losses in Passenger Cars

- Piston-rings: 35-40%
- Valve trains: 20-25%
- Bearings: 10-15%
- Cylinders: 10-15%
- Other: 5-20%



Hersey Number: $H = \frac{\eta U}{W}$ **Film Ratio**: $\Lambda = \frac{h_{min}}{Sq^*}$

Inputs:
- Geometry
- Speed
- Load
- Materials
- Lubricant
- Temperature
- Ambient



Outputs:
- Friction
- Wear
- Heat
- Noise
- Eventually failure

2. Fluid Film Lubrication

2.2 Generalized Reynolds Equation

- (Isothermal) Generalized Reynolds Equation

  - Substituting Eq. (4) in Eq. (3), one obtains the Generalized Reynolds Equation for isothermal flows:

$$\underbrace{\frac{\partial}{\partial x}\left[\frac{\rho(H_2-H_1)^3}{12\mu}\frac{\partial p}{\partial x}\right]+\frac{\partial}{\partial z}\left[\frac{\rho(H_2-H_1)^3}{12\mu}\frac{\partial p}{\partial z}\right]}_{Pressure-Flow\ (Poiseuille)}=\underbrace{\frac{\partial}{\partial x}\left[\frac{\rho(U_2+U_1)}{2}(H_2-H_1)\right]+\frac{\partial}{\partial z}\left[\frac{\rho(W_2+W_1)}{2}(H_2-H_1)\right]}_{Wedge-Flow\ (Couette)}+ \underbrace{\rho\left[\left(U_1\frac{\partial H_1}{\partial x}-U_2\frac{\partial H_2}{\partial x}\right)+\left(W_1\frac{\partial H_1}{\partial z}-W_2\frac{\partial H_2}{\partial z}\right)\right]}_{Translation\ Squeeze}+\underbrace{\rho(V_2-V_1)}_{Normal\ Squeeze}+\underbrace{(H_2-H_1)\frac{\partial\rho}{\partial t}}_{Local\ Expansion} \quad (5)$$

  - Conservative vector form:

$$\nabla\cdot(\boldsymbol{\Gamma}^{p}\nabla p_H)=\nabla\cdot(\boldsymbol{\Gamma}^{c}\vec{\boldsymbol{v}})+[S_{TS}+S_{NS}]+S_T\frac{\partial\rho}{\partial t}$$

Suitable for numerical solutions
(Tensor, vector and source terms defined accordingly)

  - Fluid shear rate and stress:

$$\begin{cases}\tau_{xy}=\mu\dfrac{\partial u}{\partial y}=\dfrac{1}{2}\dfrac{\partial p}{\partial x}[2y-(h+2H_1)]+\mu\left(\dfrac{U_2-U_1}{h}\right)\\ \tau_{zy}=\mu\dfrac{\partial w}{\partial y}=\dfrac{1}{2}\dfrac{\partial p}{\partial z}[2y-(h+2H_1)]+\mu\left(\dfrac{W_2-W_1}{h}\right)\end{cases}$$

The velocity fields of Eq. (2) were substituted on the shear rate components

Outline

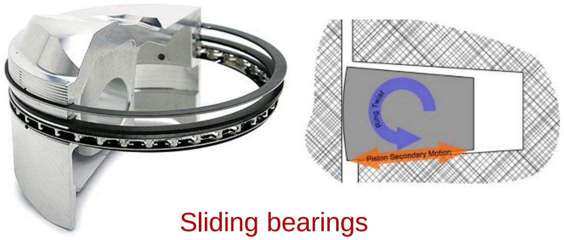
1. Bearing Types and Functions
- 1.1 Sliding and Thrust Bearings
- 1.2 Rolling Element Bearings
- 1.3 Journal Bearings

2. Lubrication of Counterformal Contacts
- 2.1 System Configuration
- 2.2 General Aspects
- 2.3 Film Thickness Calculation
- 2.4 EHD Lubrication Regimes
- 2.5 Example

3. Journal Bearing Systems
- 3.1 System Configuration
- 3.2 Lubricant Film Thickness
- 3.3 Reynolds Equation
- 3.4 Short Bearing Theory (Ocvirk Solution)
- 3.5 Bearing Design Calculation
- 3.6 Example
- 3.7 Limits of the Hydrodynamic Lubrication

1. Bearing Types & Functions

- Bearing Functions
  - (Bio)mechanical joints designed to allow power transmission and/or loading support between moving parts;
  - Fluid film bearings → Low friction and wear → Improvements in tribological performance
- Bearing Types

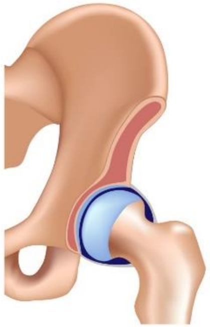


Sliding bearings
(e.g. piston-skirt and piston-rings)

Thrust bearings

Rolling element bearings

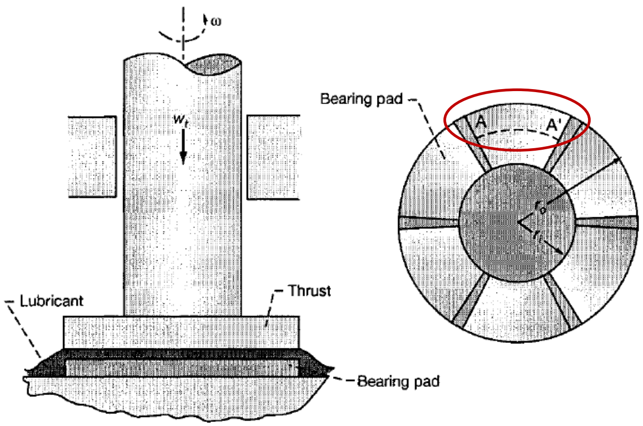
Journal bearings

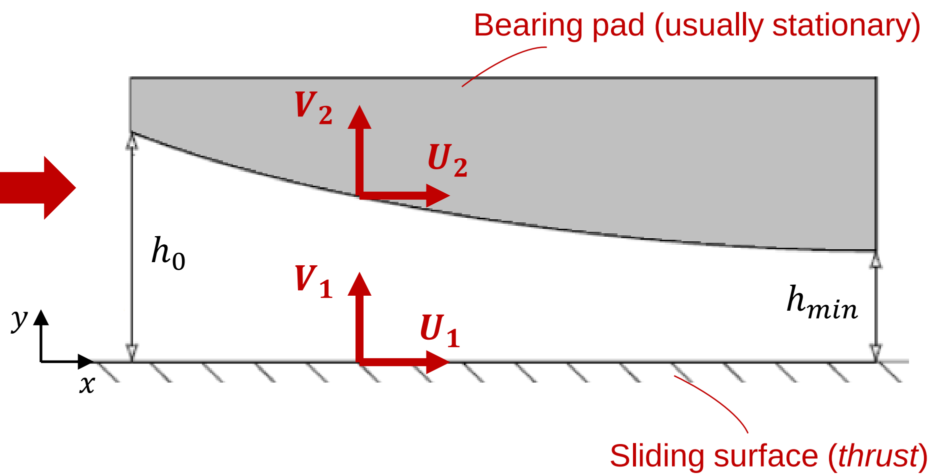


Spherical bearings

1. Bearing Types & Functions

1.1 Sliding and Thrust Bearings





- Sliding only in the x-direction: $W_1 = W_2 = 0$
- Normal velocity: $V_1 = V_{1r}$, $V_2 = V_{2r}$
- Steady-state regime: $V_{1r} = V_{2r} = 0$

$$\frac{\partial}{\partial x}\left(\frac{\rho h^3}{12\mu}\frac{\partial p}{\partial x}\right) + \frac{\partial}{\partial z}\left(\frac{\rho h^3}{12\mu}\frac{\partial p}{\partial z}\right) = \frac{\partial}{\partial x}\left[\frac{\rho h(U_2 + U_1)}{2}\right] - \rho\left(U_2\frac{\partial h}{\partial x}\right)$$

Combining...

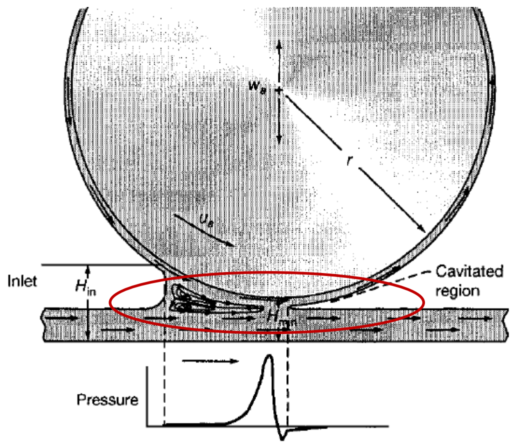
$$\frac{\partial}{\partial x}\left(\frac{\rho h^3}{12\mu}\frac{\partial p}{\partial x}\right) + \frac{\partial}{\partial z}\left(\frac{\rho h^3}{12\mu}\frac{\partial p}{\partial z}\right) = -\rho\left(\frac{U_2 - U_1}{2}\right)\frac{\partial h}{\partial x}$$

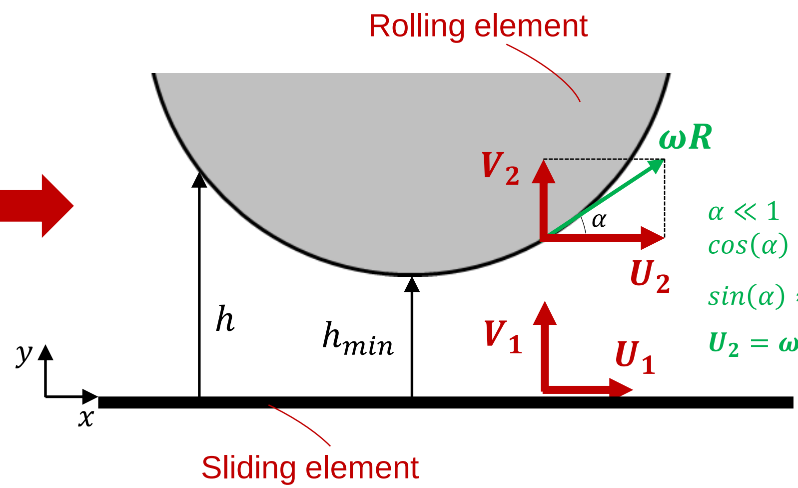
$$\tau_{xy} = -\frac{h}{2}\frac{\partial p}{\partial x} + \mu\left(\frac{U_2 - U_1}{h}\right)$$

Fluid pressure **AND** hydrodynamic friction depend on the sliding (relative) velocity

1. Bearing Types & Functions

1.2 Rolling Element Bearings





$\alpha \ll 1$
$cos(\alpha) \approx 1$
$sin(\alpha) \approx tan(\alpha) \approx \frac{\partial h}{\partial x}$
$U_2 = \omega R$

- Sliding only in the x-direction: $W_1 = W_2 = 0$
- Normal velocity: $V_1 = V_{1r}$, $V_2 = U_2 \frac{\partial h}{\partial x} + V_{2r}$ — Attention to the tangential component of the velocity
- Steady-state regime: $V_{1r} = V_{2r} = 0$

$$\frac{\partial}{\partial x}\left(\frac{\rho h^3}{12\mu}\frac{\partial p}{\partial x}\right) + \frac{\partial}{\partial z}\left(\frac{\rho h^3}{12\mu}\frac{\partial p}{\partial z}\right) = \frac{\partial}{\partial x}\left[\frac{\rho h(U_2 + U_1)}{2}\right] - \rho\left(U_2\frac{\partial h}{\partial x}\right)$$

Combining...

$$\frac{\partial}{\partial x}\left(\frac{\rho h^3}{12\mu}\frac{\partial p}{\partial x}\right) + \frac{\partial}{\partial z}\left(\frac{\rho h^3}{12\mu}\frac{\partial p}{\partial z}\right) = \rho\left(\frac{U_1 + U_2}{2}\right)\frac{\partial h}{\partial x}$$

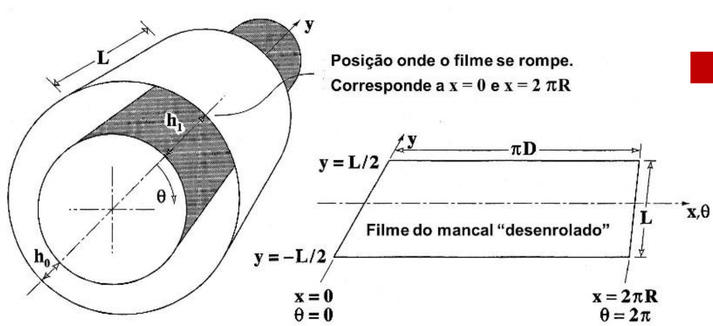
Fluid pressure depends on the average velocity (rolling/entrainment speed)

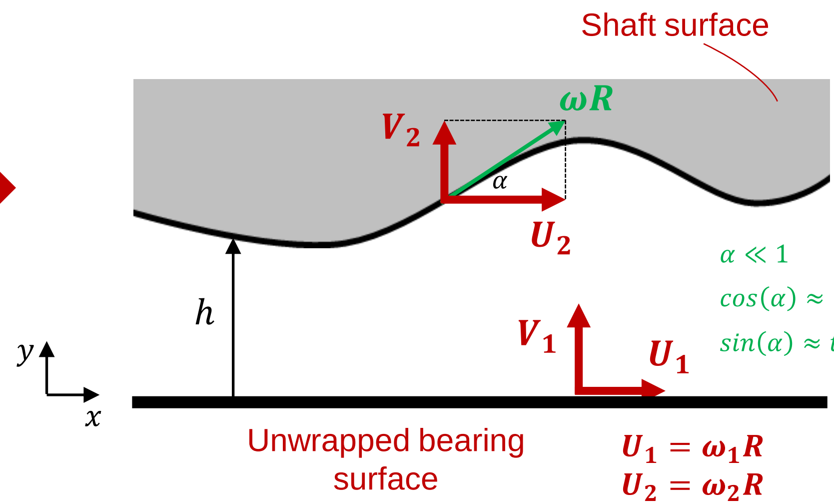
$$\tau_{xy} = -\frac{h}{2}\frac{\partial p}{\partial x} + \mu\left(\frac{U_2 - U_1}{h}\right)$$

Hydrodynamic friction depends on the sliding (relative) velocity

1. Bearing Types & Functions

1.3 Journal Bearings





$\alpha \ll 1$

$cos(\alpha) \approx 1$

$sin(\alpha) \approx tan(\alpha) \approx \frac{\partial h}{\partial x}$

$$U_1 = \omega_1 R$$
$$U_2 = \omega_2 R$$
$$x = R\theta$$

- Sliding only in the x-direction: $W_1 = W_2 = 0$
- Normal velocity: $V_1 = V_{1r}$, $V_2 = U_2 \frac{\partial h}{\partial x} + V_{2r}$ — Attention to the tangential component of the velocity
- Steady-state regime: $V_{1r} = V_{2r} = 0$

$$\frac{\partial}{\partial x}\left(\frac{\rho h^3}{12\mu}\frac{\partial p}{\partial x}\right) + \frac{\partial}{\partial z}\left(\frac{\rho h^3}{12\mu}\frac{\partial p}{\partial z}\right) = \frac{\partial}{\partial x}\left[\frac{\rho h(U_2 + U_1)}{2}\right] - \rho\left(U_2\frac{\partial h}{\partial x}\right)$$

Combining...

$$\left(\frac{1}{R}\right)\frac{\partial}{\partial \theta}\left(\frac{\rho h^3}{12\mu}\frac{\partial p}{\partial \theta}\right) + \frac{\partial}{\partial z}\left(\frac{\rho h^3}{12\mu}\frac{\partial p}{\partial z}\right) = \rho\left(\frac{\omega_1 + \omega_2}{2}\right)\frac{\partial h}{\partial \theta}$$

Fluid pressure depends on the average rotational speed

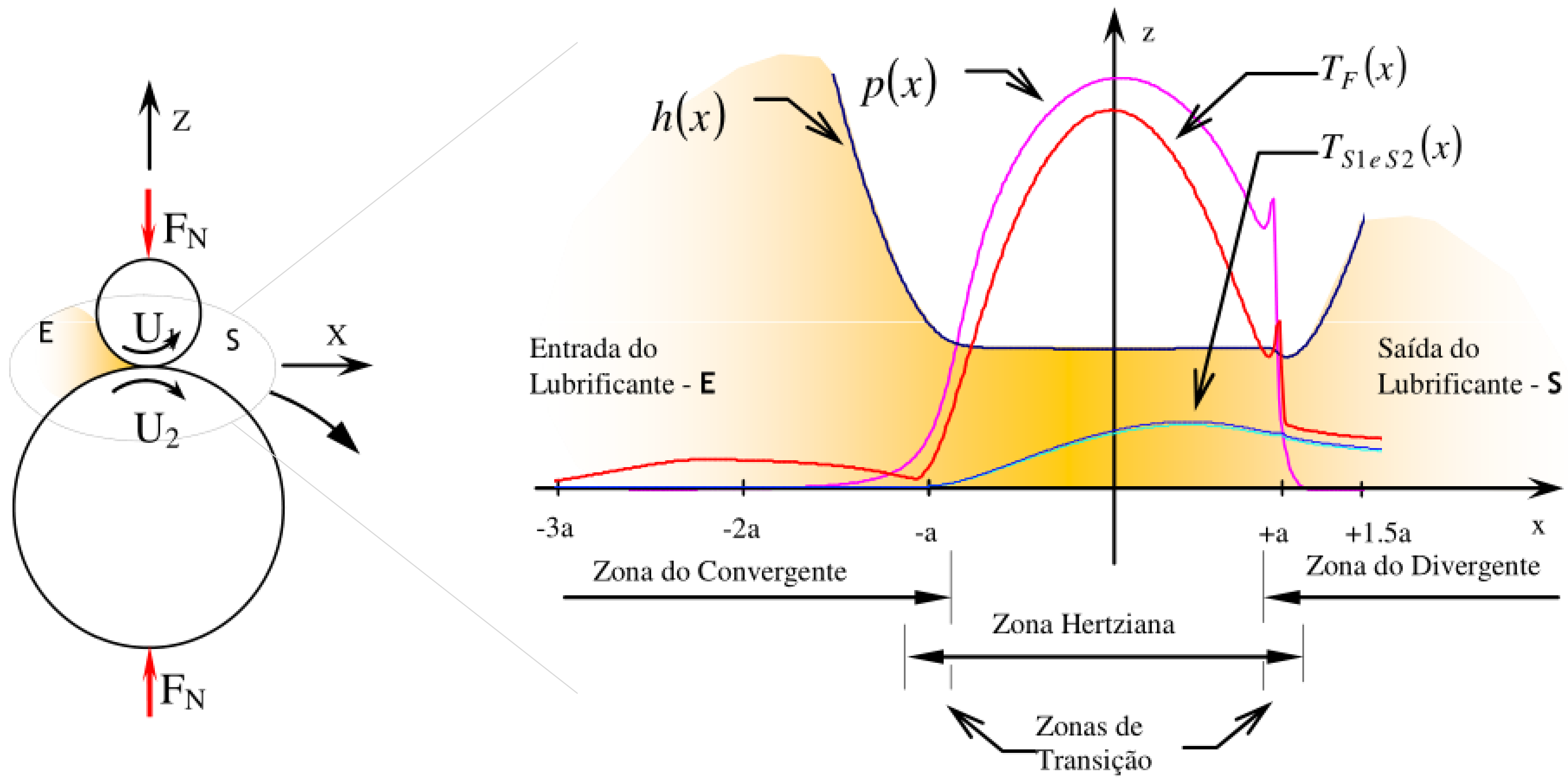
$$\tau_{xy} = -\frac{h}{2R}\frac{\partial p}{\partial \theta} + \mu R\left(\frac{\omega_2 - \omega_1}{h}\right)$$

Hydrodynamic friction depends on the sliding (relative) rotational speed

Outline

2. Lubrication of Concentrated Contacts (hard-EHL)

2.1 System Configuration and General Characteristics



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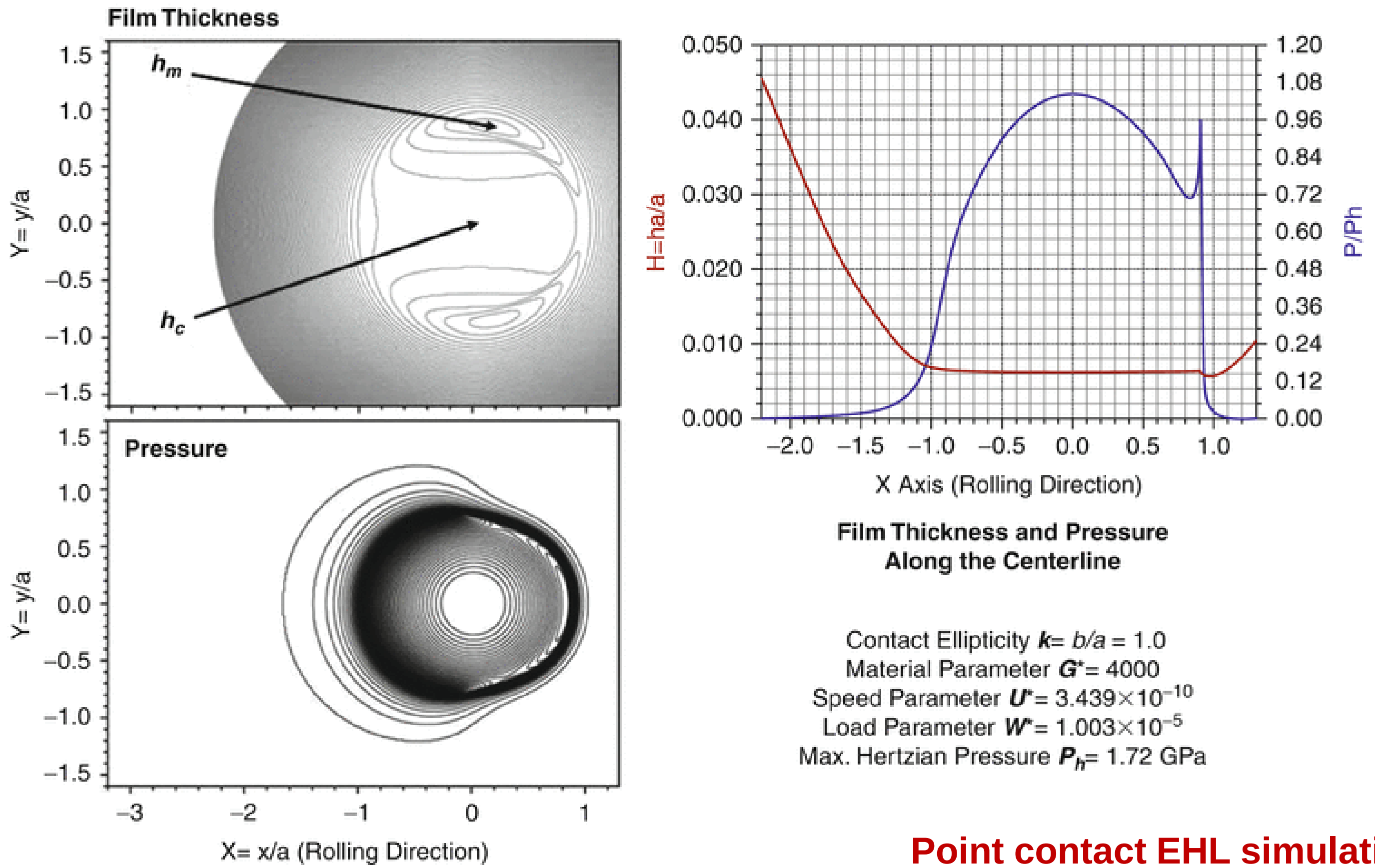
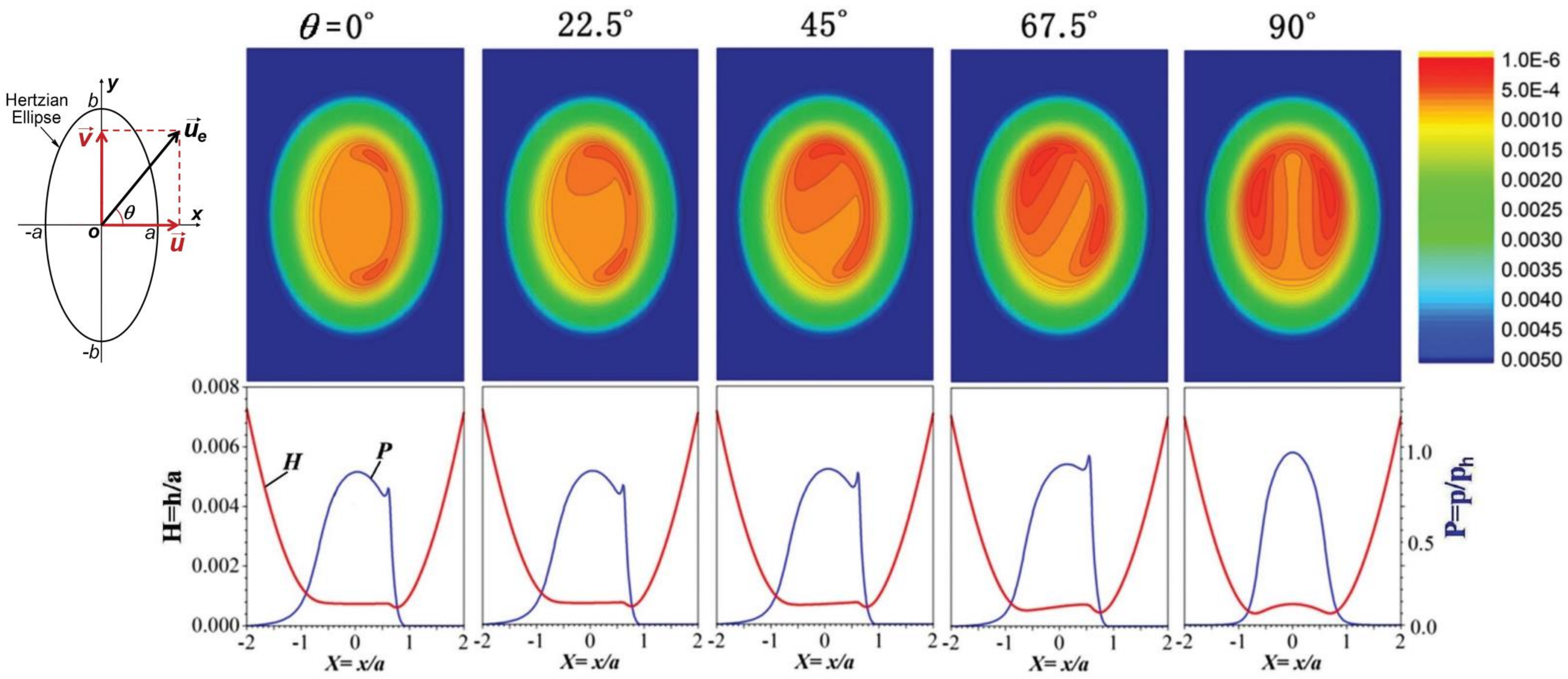


Point contact EHL simulation

2. Lubrication of Concentrated Contacts (hard-EHL)

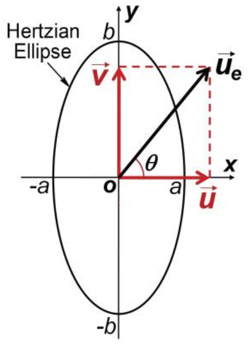
2.1 System Configuration and General Characteristics

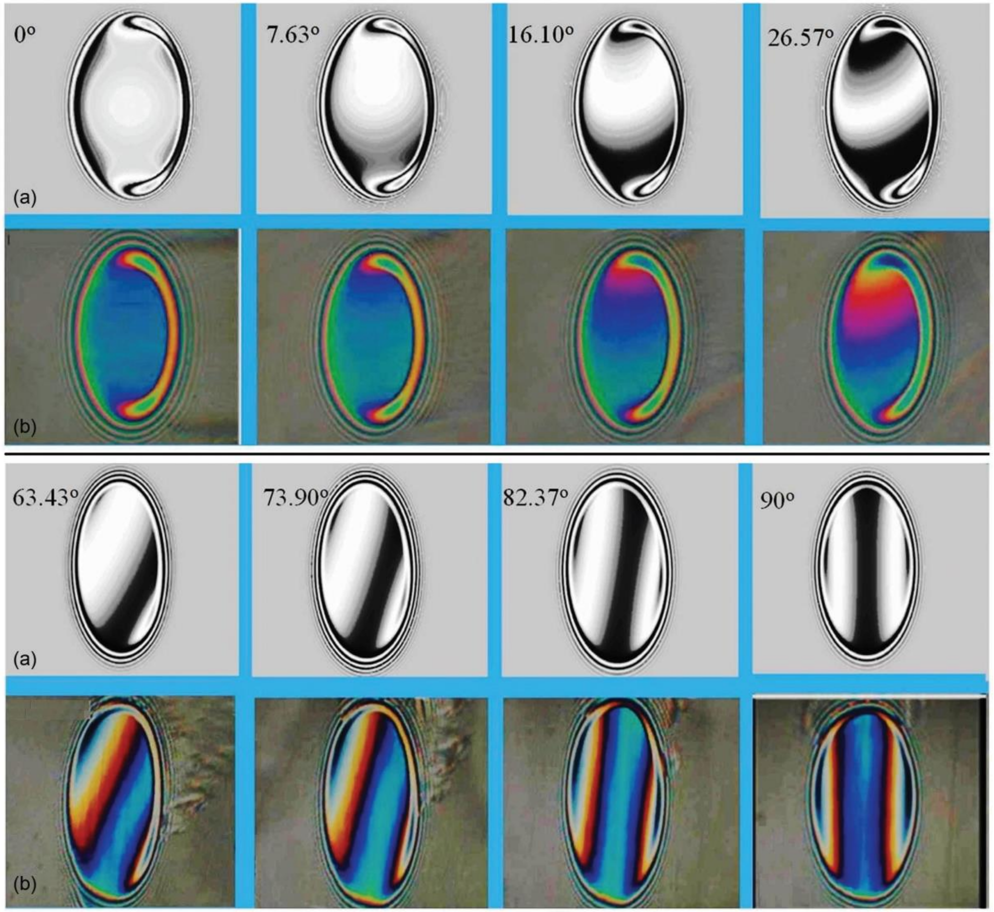


Sample of film thickness solutions with different entrainment angles. Source: [1]

2. Lubrication of Concentrated Contacts (hard-EHL)

2.1 System Configuration and General Characteristics





Simulations

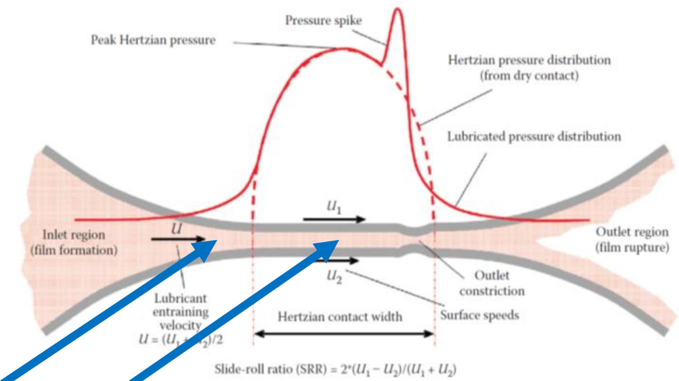
Measurements

Comparison between numerical solutions and testing results. (a) Film thickness contours from simulations and (b) measured optical interferograms. Source: [1]

2. Lubrication of Concentrated Contacts (hard-EHL)

2.2 General Characteristics

- Applications: rolling element bearings, gears, cam-tappets, etc.
- Fluid pressure magnitude: 0.5 – 5 GPa
- Lubrication mechanisms strongly influenced by:
  - Surface deformation (fluid-solid interaction problem)
  - Lubricant rheology (piezoviscosity and shear-thinning behaviour)
  - Thermal effects (high viscous dissipation and flash temperature)



- Operational parameters for engineering design:
  - Central and minimum film thickness: governed by the lubricant properties and rolling (entrainment) velocity at the inlet region
  - Friction (or traction) coefficient: governed by the lubricant rheology, local temperature rise and sliding velocity at the central contact region

2. Lubrication of Concentrated Contacts (hard-EHL)

2.3 Film Thickness Calculation

- EHD film thickness calculations based on curve-fitted formulas obtained from numerical simulation and validated with experimental data

Line Contact (Dowson & Higginson)

$$h_0 = 0.975\, R_X\, U^{0.727}\, G^{0.727}\, W^{-0.091}$$

$$h_m = 1.325\, R_X\, U^{0.70}\, G^{0.54}\, W^{-0.13}$$

$$h_0 = 0.975 \frac{\left[\alpha\, \eta_0 (U_1 + U_2)\right]^{0.727} R_X^{0.364} \left(\ell E^*\right)^{0.091}}{F_n^{0.091}}$$

$$h_m = 1.186 \frac{\left[\eta_0 (U_1 + U_2)\right]^{0.70} \alpha^{0.54} R_X^{0.43} \ell^{0.13}}{F_n^{0.13} E^{*0.03}}$$

Point Contact (Hamrock & Dowson)

$$H_0 = 1.345\, R_x U^{0.670} G^{0.530} W^{-0.067} \underbrace{\left\{1 - 0.61\, Exp\left[-0.752 \left(\frac{R_y}{R_x}\right)^{0.64}\right]\right\}}_{C_o}$$

$$H_m = 1.815\, R_x U^{0.680} G^{0.490} W^{-0.073} \underbrace{\left\{1 - Exp\left[-0.7 \left(\frac{R_y}{R_x}\right)^{0.64}\right]\right\}}_{C_M}$$

$$h_0 = 1.165\, C_0 \frac{\left[\eta_0 (U_1 + U_2)\right]^{0.67} \alpha^{0.53} R_X^{0.464}}{F_n^{0.067} E^{*0.073}}$$

$$h_m = 1.438\, C_m \frac{\left[\eta_0 (U_1 + U_2)\right]^{0.68} \alpha^{0.49} R_X^{0.466}}{F_n^{0.073} E^{*0.117}}$$

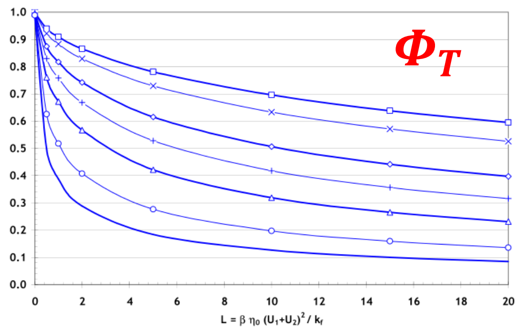
2. Lubrication of Concentrated Contacts (hard-EHL)

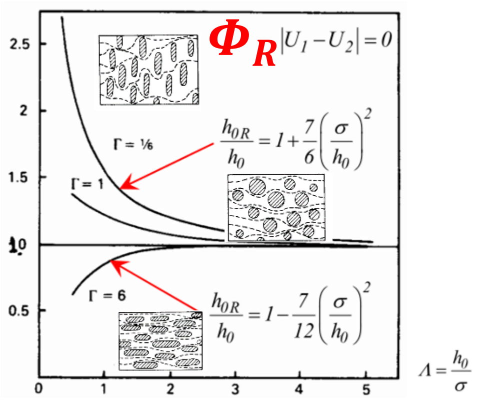
2.3 Film Thickness Calculation

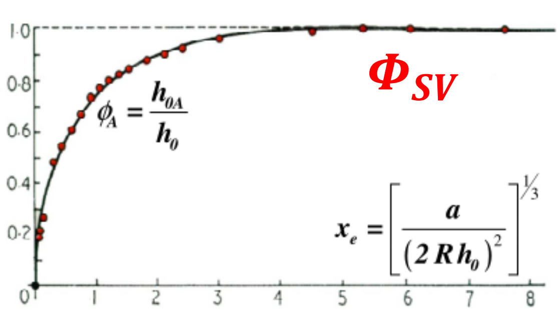
- Film thickness correction:
  - Inlet temperature (Φ_T)
  - Inlet shear-thinning (Φ_{ST})
  - Surface roughness (Φ_R)
  - Starvation (Φ_{SV})

$$h^*_{o,m} = \left(\Phi_{T,R,ST,SV}\right) h_{o,m}$$

- Correction factors determined from analytical expressions, tables or charts available in specialized literature.







2. Lubrication of Concentrated Contacts (hard-EHL)

2.4 EHL Regimes

Specific film thickness $\Lambda = \frac{h_0}{\sigma}$	Regime	Observações
$\Lambda \geq 10 \times \Lambda_1$	Hidrodinâmico *(hidrodynamic)*	Superfícies em contacto completamente separadas por um filme lubrificante muito espesso (*20 μm*).
$\Lambda \geq \Lambda_1$	Filme completo *(full film)*	Superfícies em contacto completamente separadas pelo filme lubrificante (*1 μm*).
$\Lambda_0 < \Lambda < \Lambda_1$	Filme Misto *(mixed film)*	Superfícies em contacto parcialmente separadas pelo filme lubrificante, ocorrendo em alguns pontos contacto metal / metal.
$\Lambda \leq \Lambda_0$	Filme Limite *(boundary film)*	Não existe um filme lubrificante a separar as superfícies em contacto, predominando o contacto metal / metal.

2. Lubrication of Concentrated Contacts (hard-EHL)

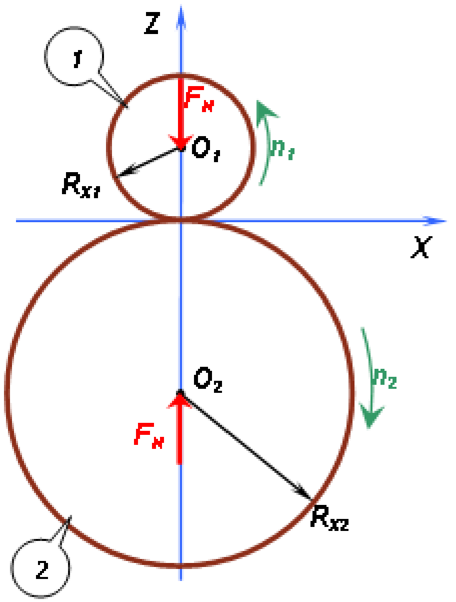
2.4 EHD Lubrication Regimes

$\Lambda = \frac{h_0}{\sigma}$	Rolamentos	Engrenagens
	$\Lambda_0 = 1.0$ e $\Lambda_1 = 3.0$	$\Lambda_0 = 0.7$ e $\Lambda_1 = 2.0$
Filme Completo	$\Lambda \geq 3.0$	$\Lambda \geq 2.0$
Filme Misto	$1.0 < \Lambda < 3.0$	$0.7 < \Lambda < 2.0$
Filme Limite	$\Lambda \leq 1.0$	$\Lambda \leq 0.7$

2. Lubrication of Concentrated Contacts (hard-EHL)

2.5 Example

Line Contact



Parameter		Unit	Disc 1	Disc 2
Radius x-dir	Rxi	m	$30\ x10^{-3}$	$45\ x10^{-3}$
Radius y-dir	Ryi	m	∞	∞
Nominal Width	ℓ	m	$15\ x10^{-3}$	$15\ x10^{-3}$
Speed	ni	rpm	1000	667
Young Modulus	Ei	GPa	210	210
Poisson Ratio	ni	/	0.3	0.3
Roughness (RMS)		um	0.30	0.30
Viscosity	v	Pas	0.016	
Piezoviscous Coef.	α	Pa-1	0.2E-7	

Contacto seco	
Po	2,020E+09
Pm	1,587E+09
a	6,303E-04
Ac	1,891E-05

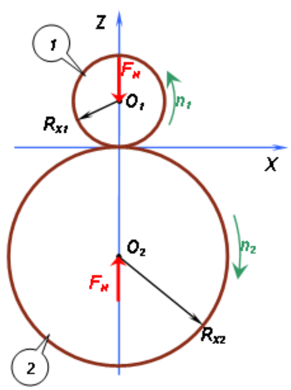
Espessura filme lub.	
U	1,190E-11
G	4,553E+03
W	4,815E-04
ho	3,669E-07
hm	2,742E-07
$\Lambda_{ISOT.}$	0,865

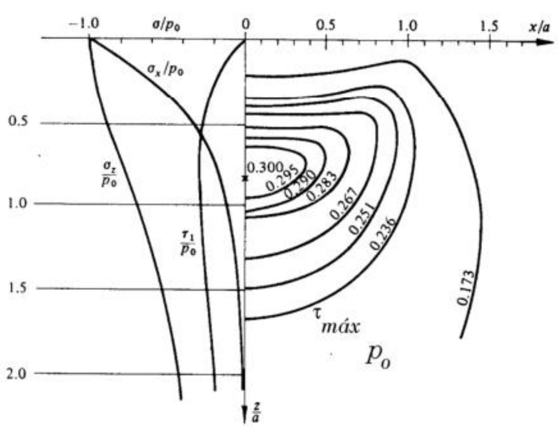
Maximum Hertz pressure (p_0)?

Specific film thickness (Λ)?

2. Lubrication of Concentrated Contacts (hard-EHL)

2.5 Example





Classificação dos Aços Estruturais através do Limite de Escoamento mínimo		
Grupo	Limite de escoamento mínimo	Exemplos, segundo ASTM
Aço carbono de média resistência	195 a 260 MPa	A36
Aço de alta resistência e baixa liga	290 a 345 MPa	A572, A242, A588, A992
Aços ligados tratados termicamente	630 a 700 MPa	A709

Discs material: Steel 1045 (ASTM A36) → Does plastic deformation occur?

Contacto seco	
Po	2,020E+09
Pm	1,587E+09
a	6,303E-04
Ac	1,891E-05

Tensões instaladas	
σ11	-2,020E+09
σ22	-1,212E+09
σ33	-2,020E+09
Tmax	6,060E+08
Zs	4,955E-04
2τo	1,010E+09
Zo	2,647E-04

Espessura filme lub.	
U	1,190E-11
G	4,553E+03
W	4,815E-04
ho	3,669E-07
hm	2,742E-07
$\Lambda_{ISOT.}$	0,865

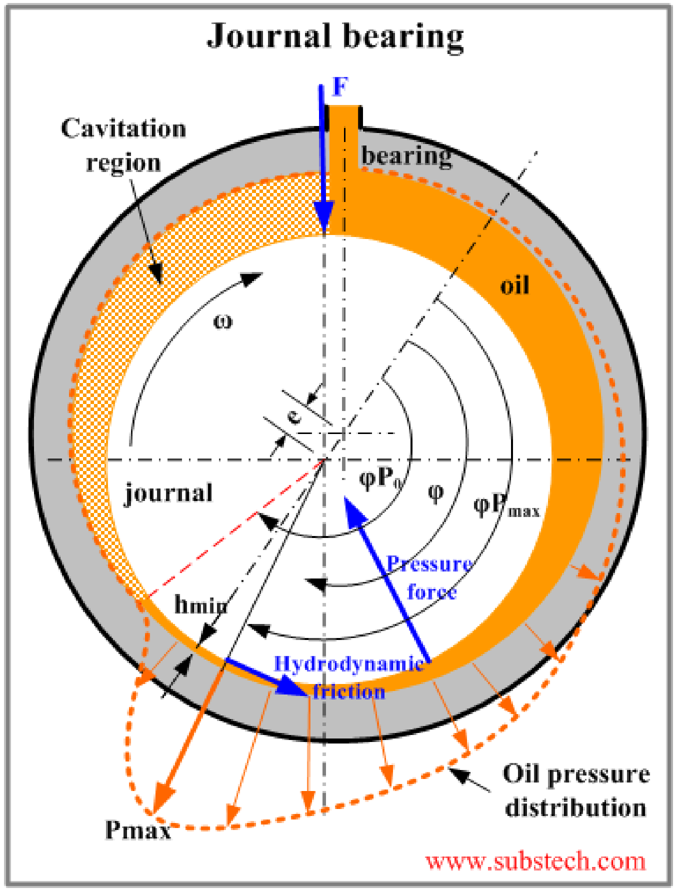
What's the lubrication regime?

How to improve the system reliability?

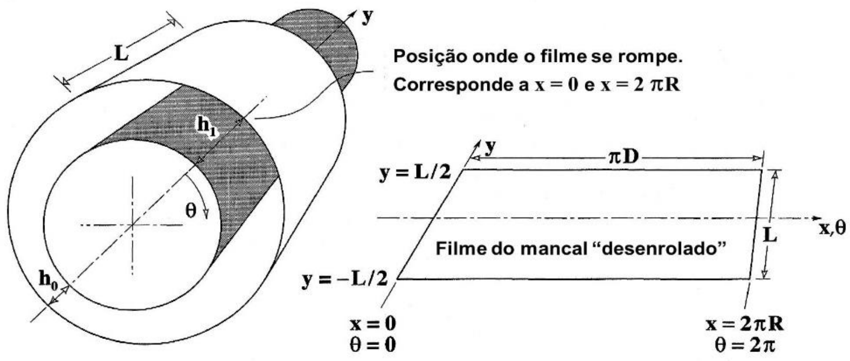
Outline

3. Journal Bearing Systems

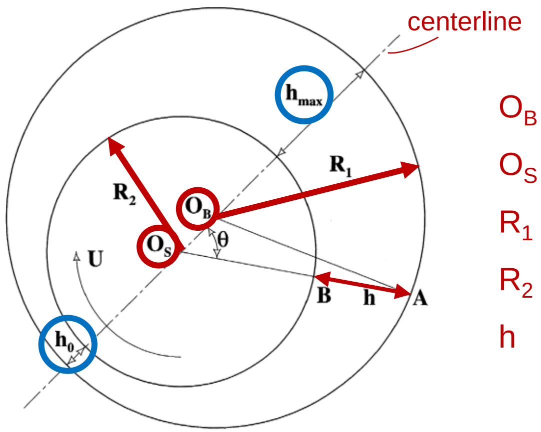
3.1 System Configuration



- Unwrapped domain (from centerline)



- Bearing geometry (plane rigid bearing)



O_B : bearing center

O_S : shaft center

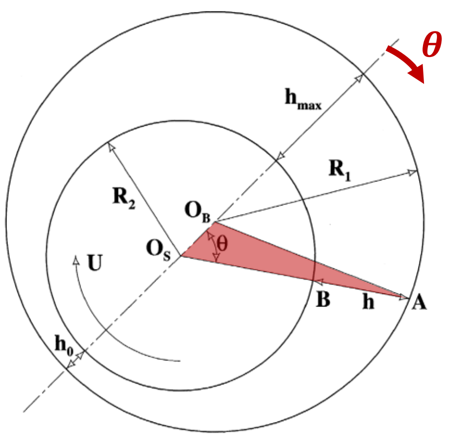
R_1 : bearing radius

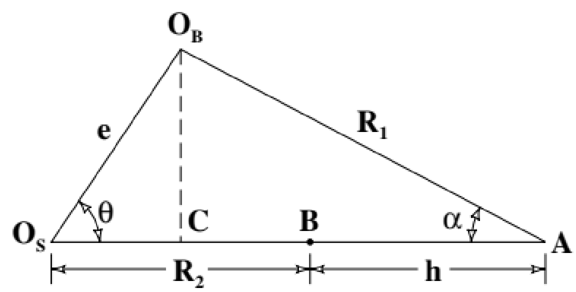
R_2 : shaft radius

h : film thickness

3. Journal Bearing Systems

3.2 Lubricant Film Thickness





$$\overline{AO_S} = \overline{CO_S} + \overline{AC} = \overline{BO_S} + \overline{AB}$$

$$\overline{AO_S} = e\cos\theta + R_1\cos\alpha = R_2 + h$$

$$\boxed{h = e\cos\theta + R_1\cos\alpha - R_2}$$

O_B : bearing center
O_S : shaft center
R_1 : bearing radius
R_2 : shaft radius
C : radial clearance
e : eccentricity
ε : eccentricity ratio
h : film thickness

- Sine rule for $\Delta O_S O_B C$: $\dfrac{e}{\sin\alpha} = \dfrac{R_1}{\sin\theta} \Rightarrow \sin\alpha = \dfrac{e}{R_1}\sin\theta$
- Trigonometric identity: $(\sin\alpha)^2 + (\cos\alpha)^2 = 1$

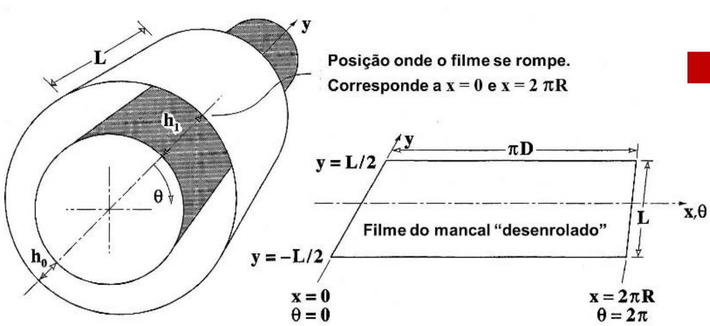
$$\cos\alpha = \sqrt{1-(\sin\alpha)^2} = \sqrt{1-\left(\frac{e}{R_1}\right)^2(\sin\theta)^2}$$

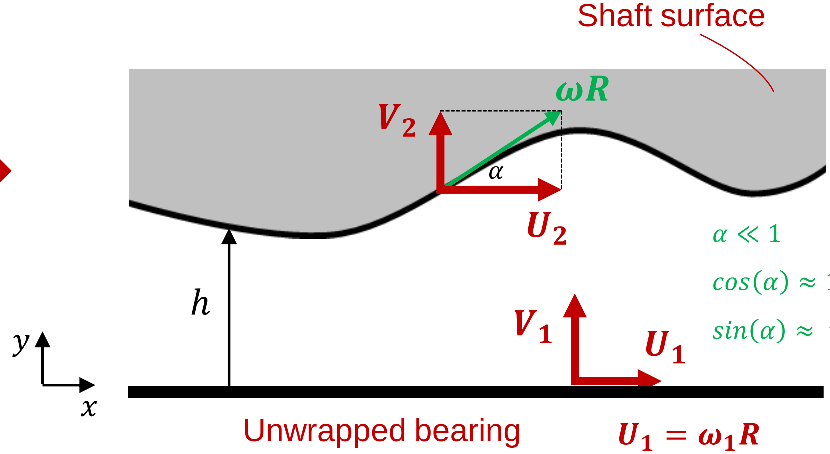
- Lubrication theory: $\left(\dfrac{e}{R_1}\right) \ll 1 \Rightarrow \cos\alpha \approx 1$
- Finally: $h = e\cos\theta + (R_1 - R_2) = e\cos\theta + C$

$$\boxed{h(\theta) = C(1 + \varepsilon\cos\theta\ )}$$ with $\varepsilon = \dfrac{e}{C}$

3. Journal Bearing Systems

3.3 Reynolds Equation





$\alpha \ll 1$

$cos(\alpha) \approx 1$

$sin(\alpha) \approx\ tan(\alpha) \approx \frac{\partial h}{\partial x}$

$$U_1 = \omega_1 R$$
$$U_2 = \omega_2 R$$
$$x = R\theta$$

- Plain bearing:

$$V_1 = V_{1r}$$
$$V_2 = U_2 \frac{\partial h}{\partial x} + V_{2r}$$
$$W_1 = W_2 = 0$$

Attention to the tangential component of the velocity

$$\left(\frac{1}{R}\right)\frac{\partial}{\partial \theta}\left(\frac{\rho h^3}{12\mu}\frac{\partial p}{\partial \theta}\right) + \frac{\partial}{\partial z}\left(\frac{\rho h^3}{12\mu}\frac{\partial p}{\partial z}\right) = \rho\left(\frac{\omega_1 + \omega_2}{2}\right)\frac{\partial h}{\partial \theta}$$

Fluid pressure depends on the average rotational speed

- Steady-state regime: $V_{1r} = V_{2r} = 0$

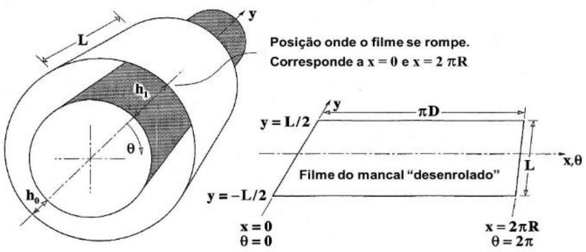
$$h(\theta) = C(1 + \varepsilon cos\theta\ )$$

How to solve the Reynolds equation for the fluid pressure?

Analytical vs. Numerical solutions

3. Journal Bearing Systems

3.4 Short Bearing Theory (Ocvirk Solution)



- Short bearing assumption: $\frac{L}{D} < \frac{1}{3} \quad \Rightarrow \quad \frac{\partial p}{\partial x} \ll \frac{\partial p}{\partial z}$

- Simplified Reynolds equation: $\left(\frac{1}{R}\right)\frac{\partial}{\partial \theta}\left(\frac{\rho h^3}{12\mu}\frac{\partial p}{\partial \theta}\right) + \frac{\partial}{\partial z}\left(\frac{\rho h^3}{12\mu}\frac{\partial p}{\partial z}\right) = \rho\left(\frac{\omega_1 + \omega_2}{2}\right)\frac{\partial h}{\partial \theta}$

$$\frac{\partial}{\partial z}\left(\frac{\rho h^3}{12\mu}\frac{\partial p}{\partial z}\right) = \rho\left(\frac{\omega_1 + \omega_2}{2}\right)\frac{\partial h}{\partial \theta}$$

Integration and boundary conditions in z

$$p(z) = \left[\frac{3\mu(\omega_1 + \omega_2)}{h^3}\right]\frac{\partial h}{\partial \theta}\left(z^2 - \frac{L^2}{4}\right)$$

$$h(\theta) = C(1 + \varepsilon cos\theta\ )$$

$$\frac{\partial h}{\partial \theta} = -C(\varepsilon sin\theta)$$

3. Journal Bearing Systems

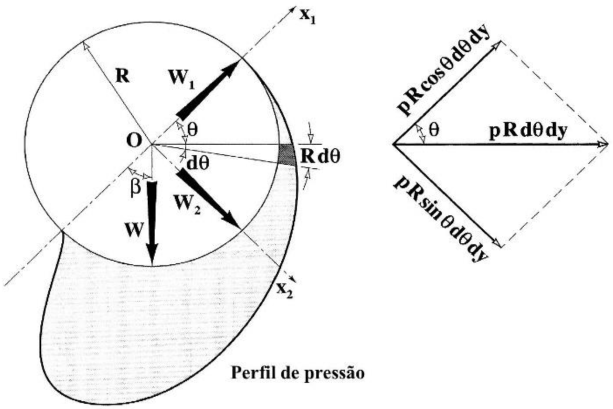
3.4 Short Bearing Theory (Ocvirk Solution)

- Load Carrying Capacity

$$W_1 = \int_0^{\pi}\int_{-L/2}^{L/2}(pR\cos\theta)dzd\theta = -\frac{\mu(\omega_1+\omega_2)RL^3\varepsilon^3}{c^2(1-\varepsilon^2)^2}$$

$$W_2 = \int_0^{\pi}\int_{-L/2}^{L/2}(pR\sin\theta)dzd\theta = -\frac{\mu(\omega_1+\omega_2)R\varepsilon\pi L^3}{4c^2(1-\varepsilon^2)^{3/2}}$$

Coordinate system defined from the centerline



Carga e pressão no mancal

- Load Magnitude

$$W = \sqrt{W_1{}^2 + W_2{}^2} \quad \Rightarrow \quad W = \left[\frac{\mu(\omega_1+\omega_2)R\varepsilon L^3}{c^2(1-\varepsilon^2)^2}\right]\frac{\pi}{4}\sqrt{\left(\frac{16}{\pi^2}-1\right)\varepsilon^2+1}$$

3. Journal Bearing Systems

3.4 Short Bearing Theory (Ocvirk Solution)

- Friction Torque

Short bearing

$$T = \int_0^{\pi} \int_{-L/2}^{L/2} (\tau R^2) dz d\theta \qquad \tau = \mu \frac{\partial u}{\partial y} \qquad u(y) = \left(\frac{y^2 - yh}{2\mu R}\right)\frac{\partial p}{\partial \theta} + (\omega_1 + \omega_2) R \frac{y}{h}$$

$$T = \int_0^{\pi} \int_{-L/2}^{L/2} \left[\frac{\mu(\omega_1 + \omega_2)R}{h}\right] dz d\theta$$

$$T = \left[\frac{2\pi\mu(\omega_1 + \omega_2)R^3 L}{c}\right] \frac{1}{\sqrt{1 - \varepsilon^2}}$$

- Concentric bearings

$e = \varepsilon = 0$

Petroff bearing (1883)

- Surfaces contact

$e = c \qquad \varepsilon = 1$

"Infinite" hydrodynamic friction (metal-to-metal contact, mixed/boundary lubrication)

3. Journal Bearing Systems

3.4 Short Bearing Theory (Ocvirk Solution)

- Coefficient of Friction

$$COF = \frac{T}{RW} \quad with \quad \begin{cases} W = \left[\frac{\mu(\omega_1+\omega_2)R\varepsilon L^3}{c^2(1-\varepsilon^2)^2}\right]\frac{\pi}{4}\sqrt{\left(\frac{16}{\pi^2}-1\right)\varepsilon^2+1} \\ T = \left[\frac{2\pi\mu(\omega_1+\omega_2)R^3L}{c}\right]\frac{1}{\sqrt{1-\varepsilon^2}} \end{cases}$$

$$COF = \frac{8Rc(1-\varepsilon^2)^{3/2}}{\varepsilon L^2\sqrt{0.621\varepsilon^2+1}}$$

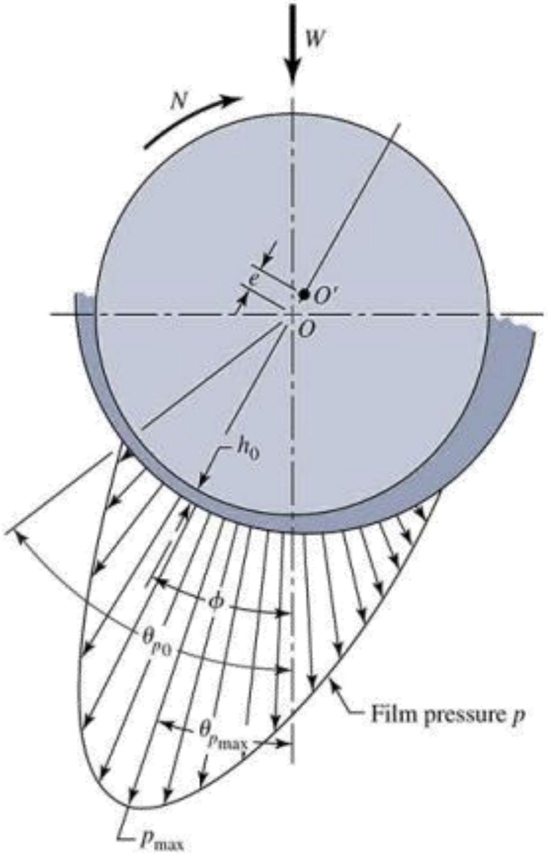
DO NOT depend on lubricant viscosity

3. Journal Bearing Systems

3.5 Bearing Design Calculation

- Input or "*controllable*" variables
  - Lubricant viscosity: μ [Pa.s]
  - Average load pressure: P [Pa]
  - Speed rotation: N [RPM]
  - Bearing dimensions: R, L, c [m]

- Design or "*dependent*" variables
  - Eccentricity factor: ε [-]
  - MOFT: h_0 [m]
  - Attack angle: φ [deg]
  - Coefficient of friction: COF [-]
  - Avg. temperature raise: ΔT [^oC]
  - Leakage flow: Q [m^3/s]



Fundamental Problem: determine satisfactory limits for the "dependent" variables by varying the "controllable" ones.

3. Journal Bearing Systems

3.5 Bearing Design Calculation

- Sommerfeld number (dimensionless)
  - Characteristic number for the design of hydrodynamic bearings
  - Defined in term of the main “controllable” variables

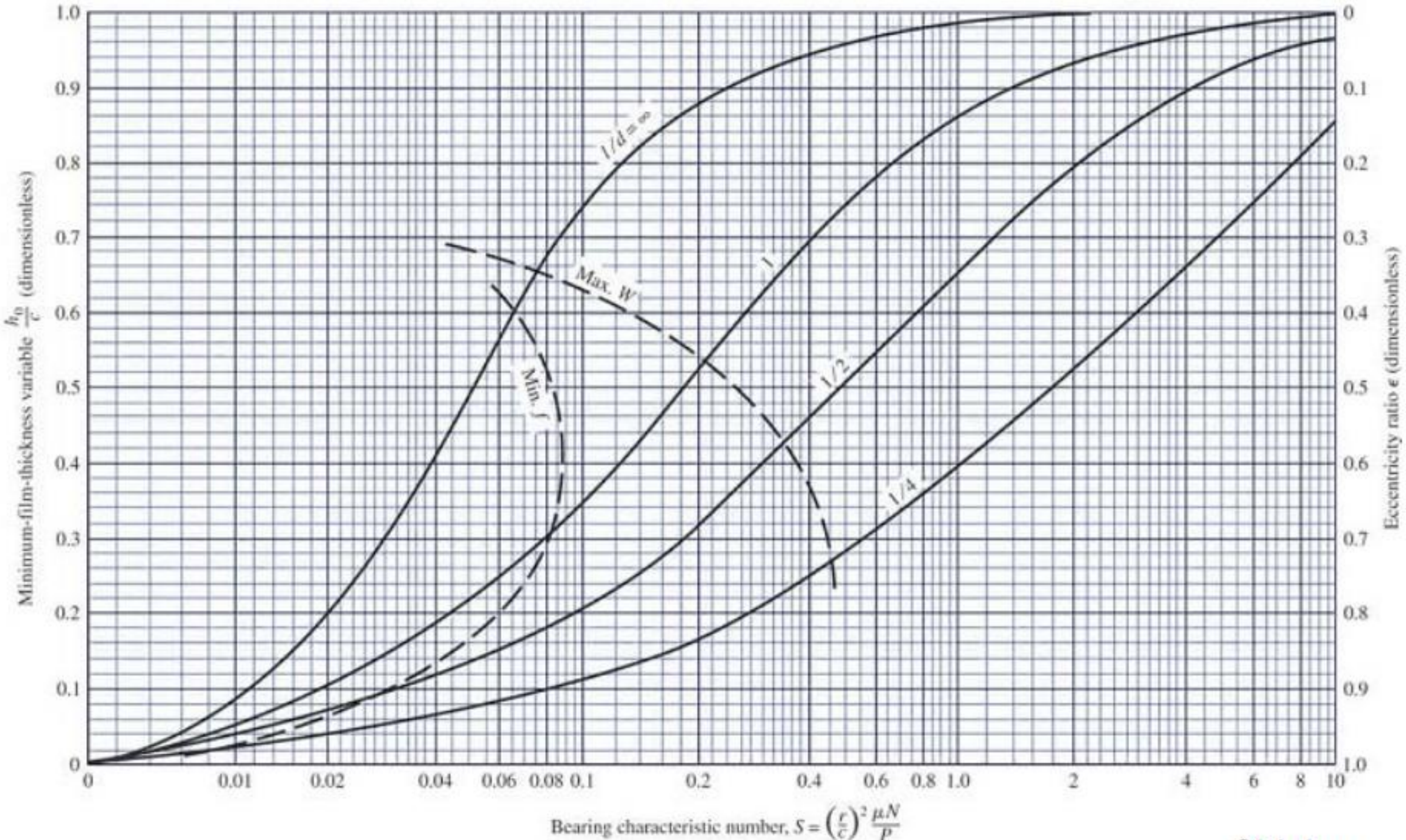
$$\Delta = \frac{W}{LU\mu}\left(\frac{C}{R}\right)^2 \qquad or \qquad S = \Delta\pi = \frac{P}{N\mu}\left(\frac{C}{R}\right)^2$$

- R : bearing radius [m]
- L : bearing width [m]
- C : radial clearance [m]
- μ : Lubricant viscosity [Pa.s]
- W : Load force [N]
- P : Average load pressure [Pa]
- N : Rotational speed [RPM]

3. Journal Bearing Systems

3.5 Bearing Design Calculation

MOFT and ε vs. Sommerfeld



Shigley

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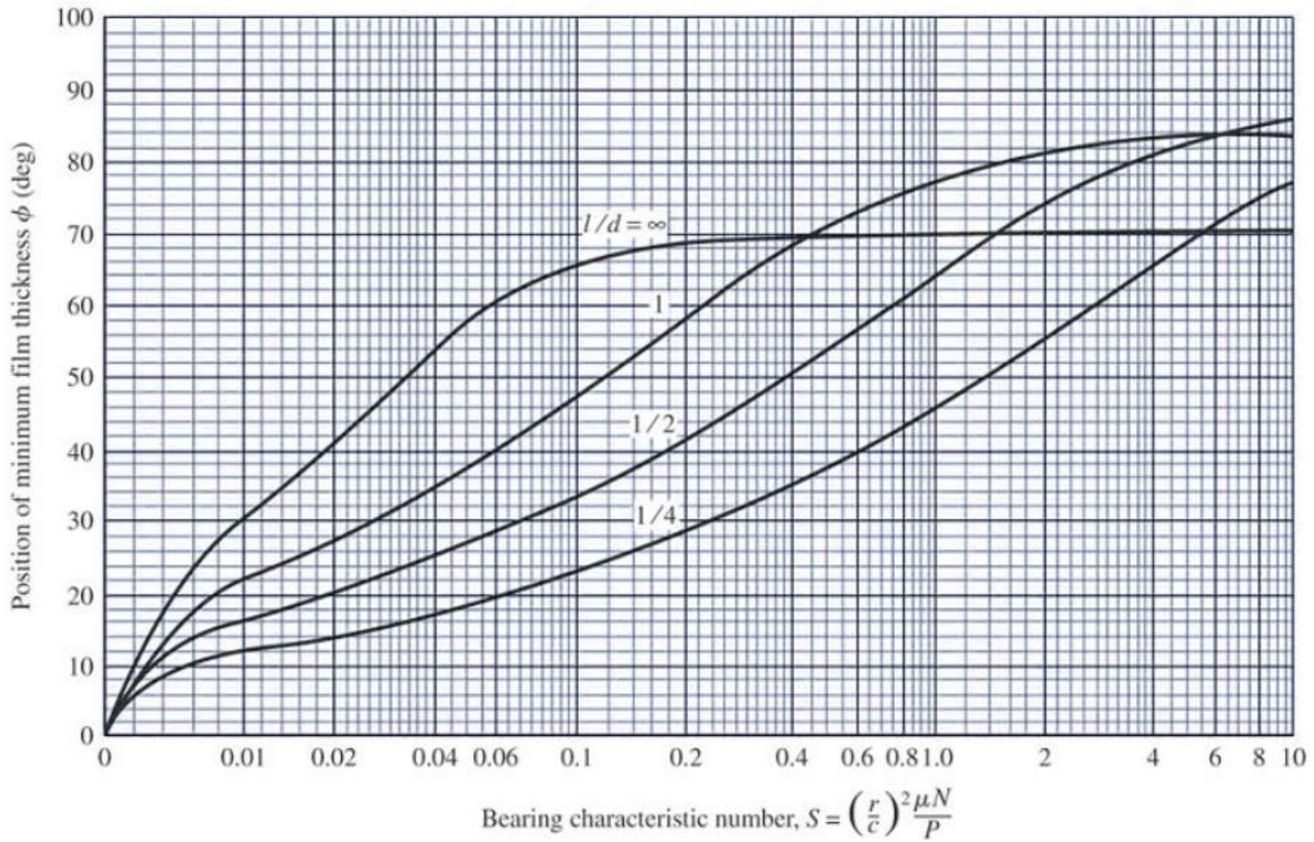
3.5 Bearing Design Calculation

Position of MOFT vs. Sommerfeld



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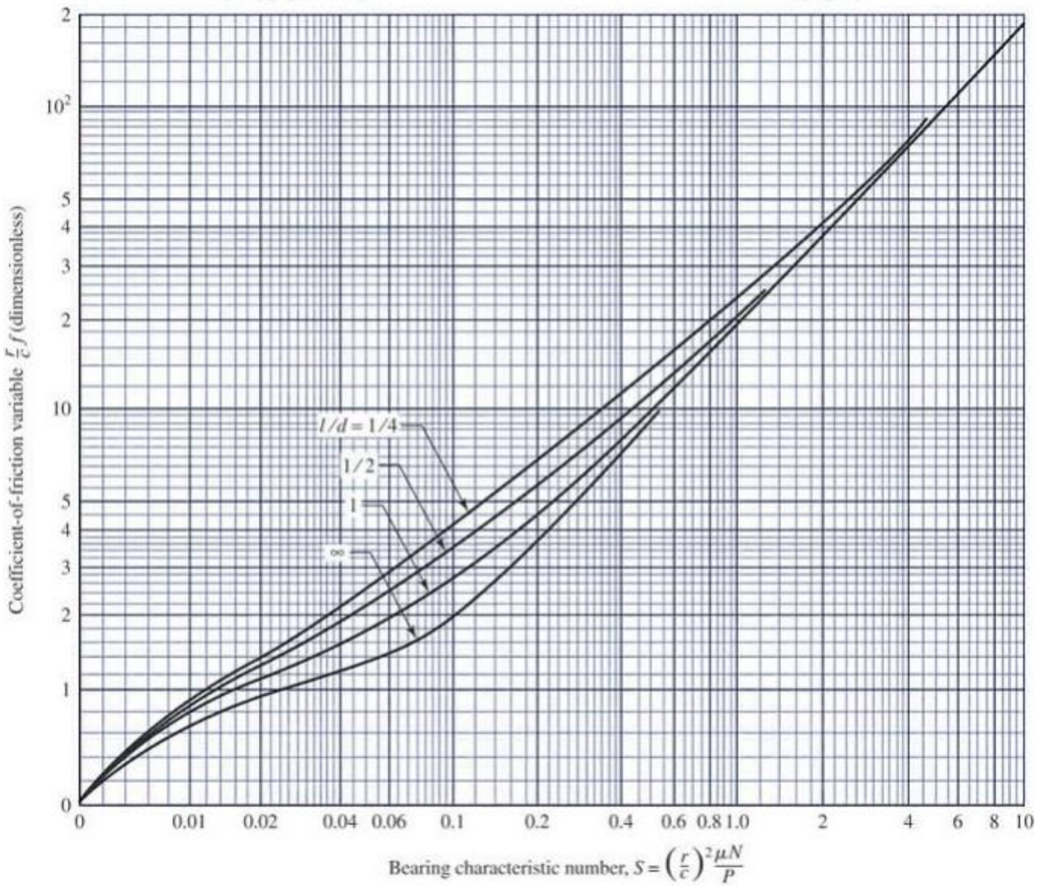
3.5 Bearing Design Calculation

COF vs. Sommerfeld



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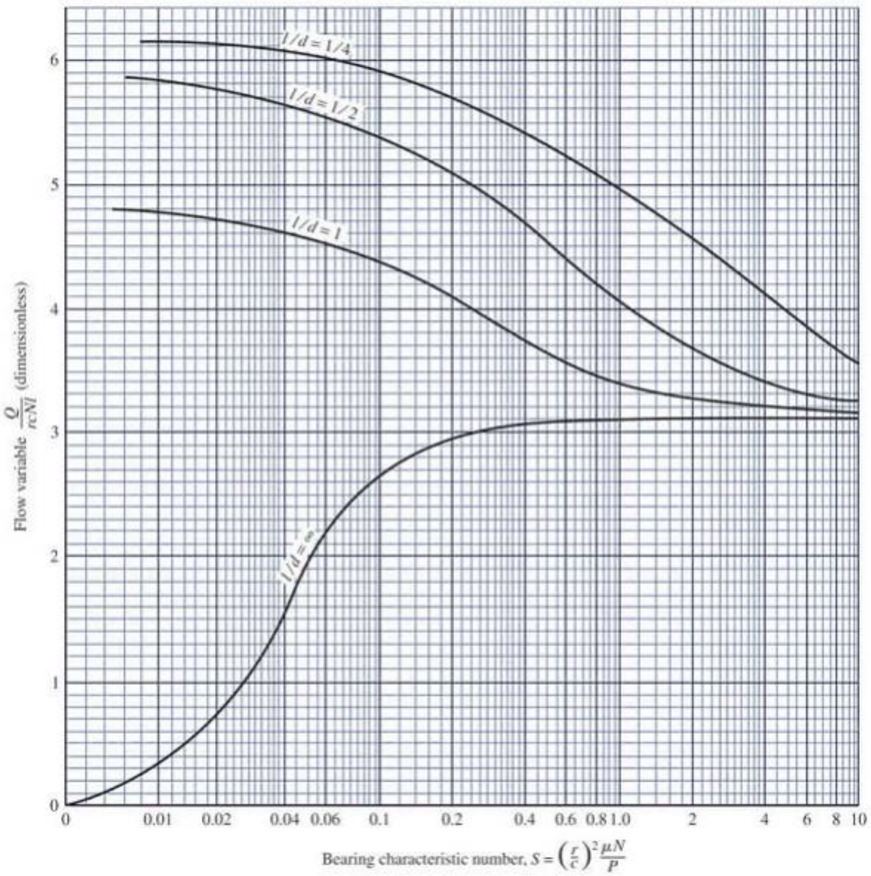
3.5 Bearing Design Calculation

Leakage flow vs. Sommerfeld



3. Journal Bearing Systems

3.6 Example

The specifications and operating conditions of a given journal bearing are summarized as follows:

N = 30 rps
W = 2200 N
R = 20 mm
L = 40 mm
T = 50°C
Lubricant: SAE60 (μ = 170 mmPa.s)

Based on the curves shown in the previous slides, determine the following operational parameters that ensure the system operate under **maximum loading conditions**.

a) Radial clearance (C)

b) Eccentricity (e)

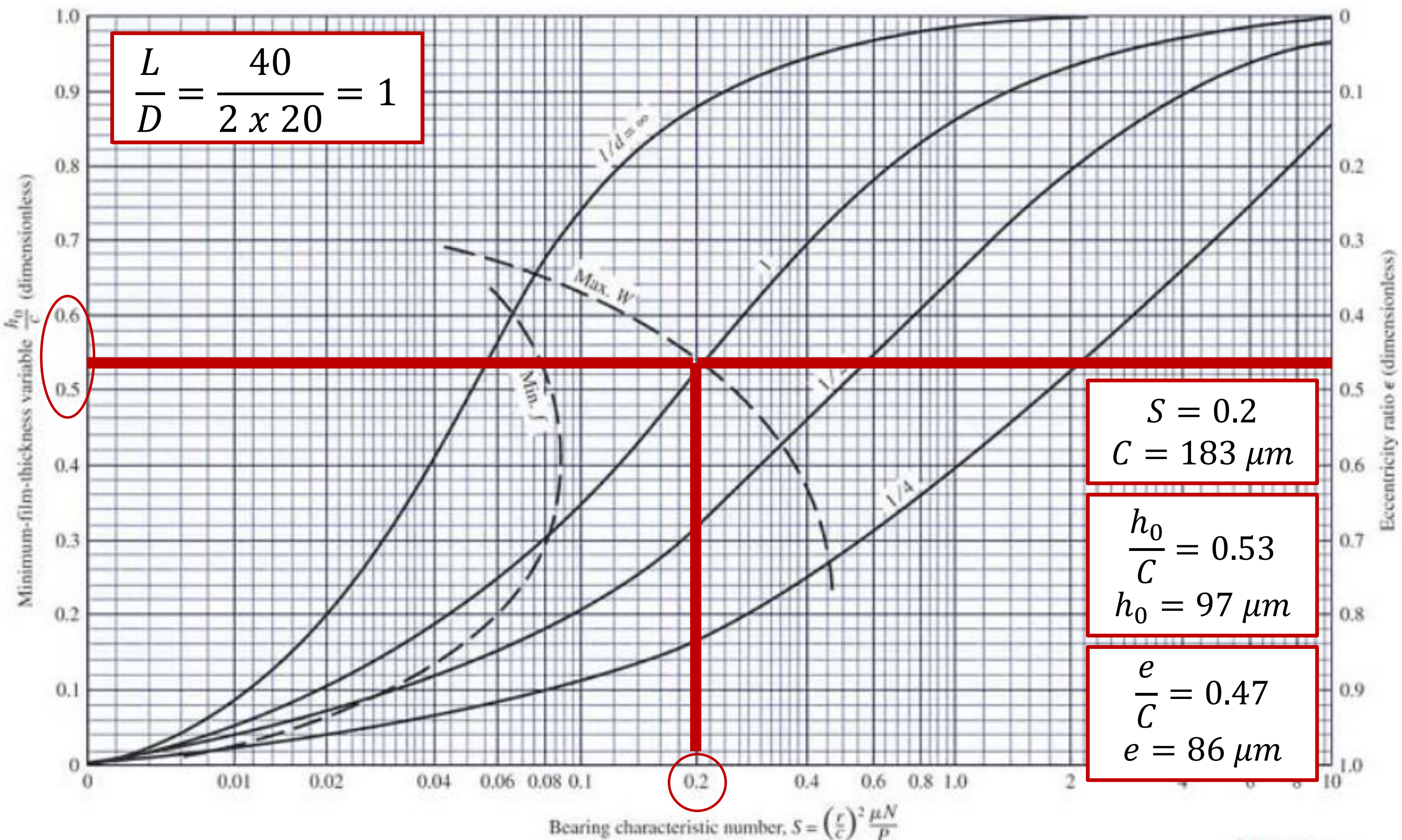
c) MOFT (h_0)

d) Position of the MOFT (φ)

3. Journal Bearing Systems

$$S = \frac{P}{N\mu}\left(\frac{C}{R}\right)^2$$

3.6 Case Study



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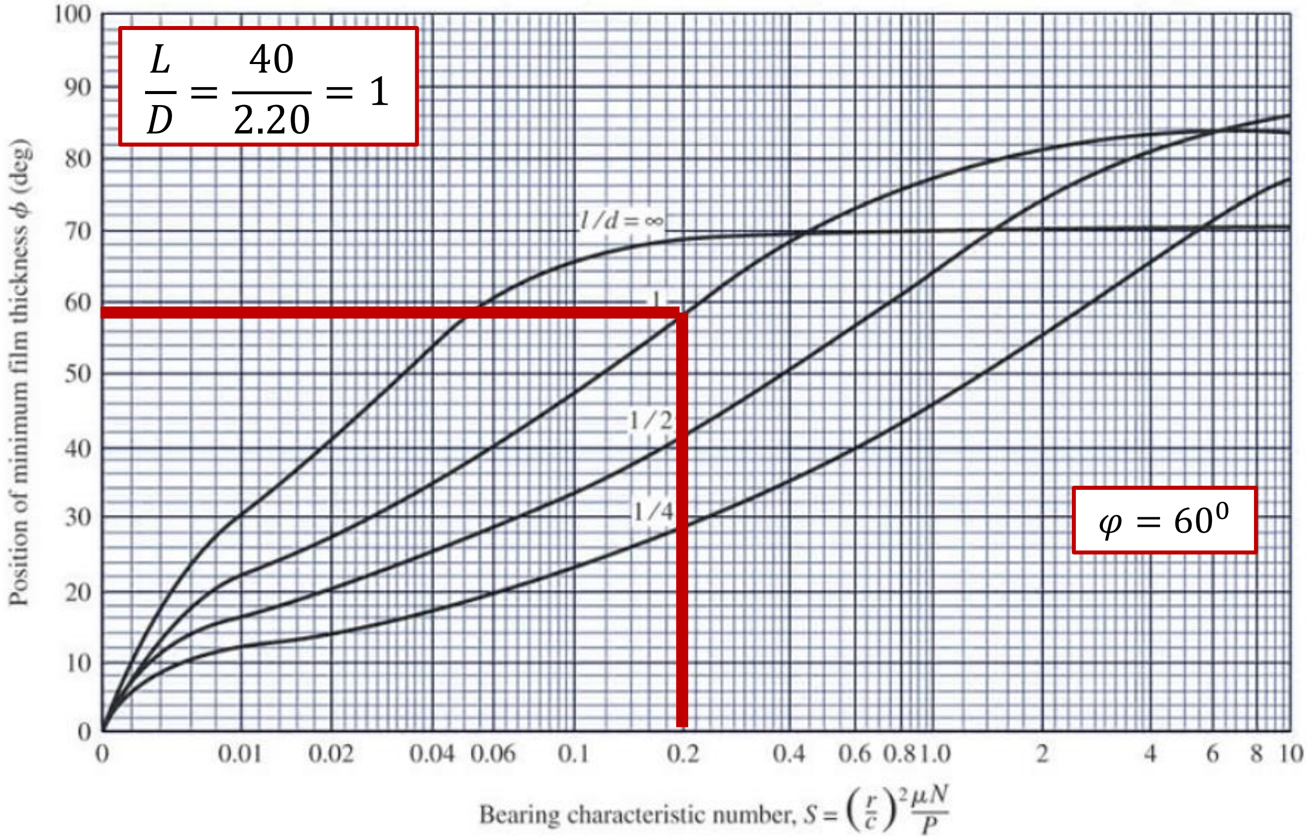
3. Journal Bearing Systems

3.6 Case Study



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References

General

1. **Hamrock B.J., Schmid S.R., Jacobson B.O., Fundamentals of Fluid Film Lubrication, 2nd edition, Marcel Dekker, 2004.**
2. **Wang J., Zhu D., Interfacial Mechanics: Theories and Methods for Contact and Lubrication, CRC Press, 2019.**
3. **Frene J., Nicolas D., Degueurce B., Berthe D., Godet M., Hydrodynamic Lubrication: Bearings and Thrust Bearings, Elsevier, 1997.**
4. Szeri A.Z., Fluid Film Lubrication, 2nd edition, Cambridge University Press, 2011.
5. Pinkus O., Sternlicht B., Theory of Hydrodynamic Lubrication, McGraw-Hill, 1961.
6. Gohar R., Elastohydrodynamics, Imperial College Press, 2002.
7. Bair S.S., High Pressure Rheology for Quantitative Elastohydrodynamics, 2nd edition, Elsevier, 2019.
8. Seabra J.H.O., Campos A., Sottomayor A, Lubrificação Elastohidrodinâmica (Apostila), 2ª edição, Faculdade de Engenharia da Universidade do Porto (FEUP), 2002.
9. Wang J., Chung Y.-W., Encyclopedia of Tribology, Springer, 2013.
10. Stachowiak G.W., Batchelor A.W., Engineering Tribology, 4th edition, Butterworth-Heinemann, 2014.

Modelling & Simulation

1. **Habchi W., Finite Element Modelling of Elastohydrodynamic Lubrication Problems, Wiley, 2018.**
2. Venner C.H., Lubrecht A.A., Multilevel Methods in Lubrication, Elsevier, 2000.

Bearings Design & Applications

1. **Adams M.L., Bearings – Basic Concepts and Design Applications, CRC Press, 2018.**
2. Harnoy A., Bearing Design in Machinery, Marcel Dekker, 2002.
3. Pirro D.M., Webster M., Daschner E., Lubrication Fundamentals, CRC Press, 2016.
4. Qiu M., Chen L., Li Y., Yan J., Bearing Tribology: Principles and Applications, Spring, 2017.
5. Totten G.E., Handbook of Lubrication and Tribology – Vol. I Application and Maintenance, 2nd edition, CRC Press, 2006.
6. Bruce R.W., Handbook of Lubrication and Tribology – Vol. II Theory and Design, 2nd edition, CRC Press, 2012.