Module 5 - SLIDING CONTACT BEARINGS

Lecture 4 – JOURNAL BEARINGS - PRACTICE

Contents

4.1 BEARING MATERIALS

Bearing materials constitute an import part of any journal bearing. Their significance is at the start of the hydro-dynamic lubrication when metal to metal contact occurs or during mixed and boundary lubrication period.

4.1.1 Desirable properties of a good bearing material

1. Conformability (low elastic modulus) and deformability (plastic flow) to relieve local high pressures caused by misalignment and shaft deflection.

2. Embeddability or indentation softness, to permit small foreign particles to become safely embedded in the material, thus protecting the journal against wear.

3. Low shear strength for easy smoothing of surface asperities.

4. Adequate compressive strength and fatigue strength for supporting the load and for enduring the cyclic loading as with engine bearings under all operating conditions.

5. Should have good thermal conductivity to dissipate the frictional heat and coefficient of thermal expansion similar to the journal and housing material.

6. It should be compatible with journal material to resist scoring, welding and seizing.

7. Should have good corrosion resistance against the lubricant and engine combustion products.

4.1.2 Composition of bearing materials

Babbits are the most commonly used bearing materials. Babbitts have excellent conformability and embeddability, but have relatively low compressive and fatigue strength, particularly above 77˚C. Babbitts can seldom be used above about 121˚C.

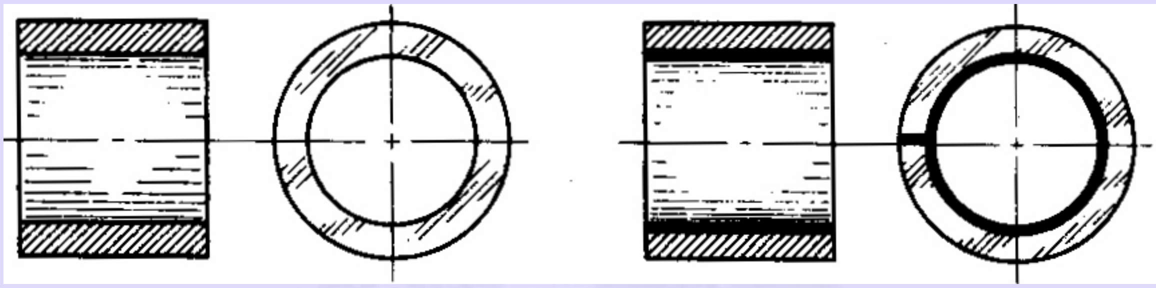
Other materials such as tin bronze, leaded bronze, copper lead alloy, aluminium bronze, aluminium alloys and cast iron are also used in many applications. Widely used bearing material compositions are given below:

a.Tin-base babbitts with 89% Sn, 8% Pb and 3% Cu,

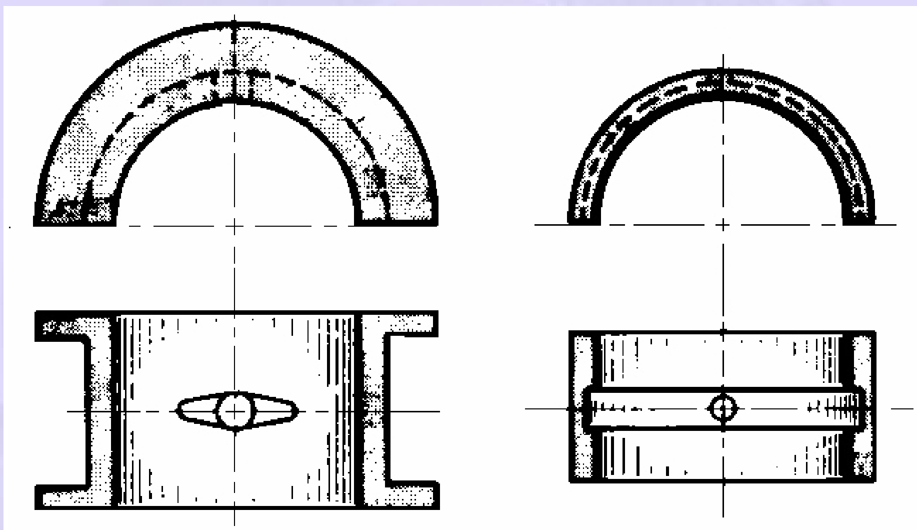
b. Lead- base babbitts with 75% Pb, 15% Sb and 10% Sn,

c. Copper alloys such as Cu- 10% to 15% Pb.

Bimetal and trimetal bearings are used in engine application to reduce the size of the bearing and obtain good compatibility and more load capacity. The bearings can be of solid bushings or lined bushings. Some times two piece with or without flanges are also used. These are shown in Fig.4.1. The inner surfaces of the bearings are grooved to facilitate the supply of lubricant to the surface of the journal. Various groove pattern used in industry are shown in Fig. 4.2



(a) Solid bushing **(b) Lined bushing**



© Fanged **(d) Straight**

Fig.4.1 Various types of bush bearings

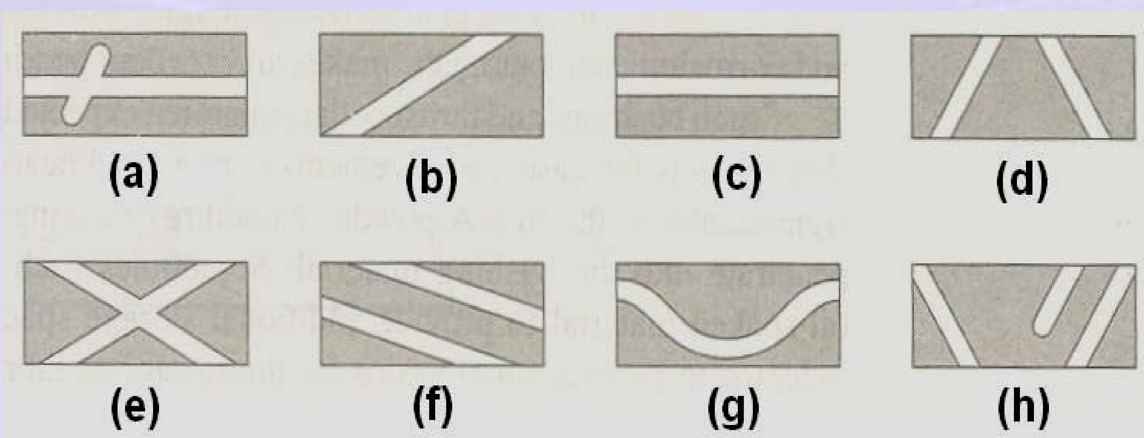


Fig 4.2 Developed views of typical groove patterns

4.1.3 BEARING MATERIALS- RECOMMENDED RADIAL CLEARANCES FOR CAST- BRONZE

Recommended radial clearances for cast bronze bearings are shown in Fig.4.3.

A – Precision spindles made of hardened ground steel, running on lapped cast bronze bearings (0.2 to 0.8 μm rms finish) with a surface velocity less than 3 m/s.

B - Precision spindles made of hardened ground steel, running on lapped cast bronze bearings (0.2 to 0.4 μm rms finish) with a surface velocity more than 3 m/s.

C- Electric motors, generators, and similar types of machinery using ground journals in broached or reamed cast-bronze bearings (0.4 to 0.8 μm rms finish)

D – General machinery which continuously rotates or reciprocates and uses turned or cold rolled steel journals in bored and reamed cast-bronze bearings (0.8 to 1.6 μm rms finish)

E- Rough service machinery having turned or cold rolled steel journals in bored and reamed cast-bronze bearings (0.8 to 1.6 μm rms finish)

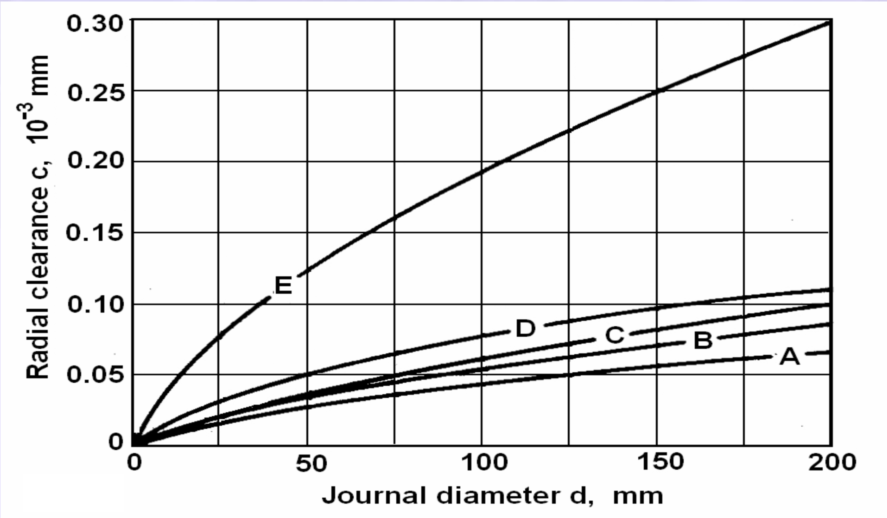


Fig.4.3 Recommended radial clearance for cast bronze bearings

4.2 HYDRODYNAMIC LUBRICATED BEARING DESIGN – Problem 1

A journal bearing of a centrifugal pump running at 1740 rpm has to support a steady load of 8kN. The journal diameter from trial calculation is found to be 120 mm. Design suitable journal bearing for the pump to operate under hydrodynamic condition.

Data:

n = 1740 rpm = 29 rps; F = 8 kN = 8000 N; r = 0.5d= 60mm

Solution:

1. From Table 4.1a, for centrifugal pumps, recommended unit load is 0.6 to 1.2MPa

2. Recommended l/d ratio for centrifugal pumps is 0.75 to 2.
A value of l/d = 0.75 is chosen. L = 0.75 d = 0.75x120 = 80mm

3. p = F/ l d = 8000 / 80 x 120 = 0.833 MPa which is within the range for centrifugal pump 0.6 to 1.2 MPa

4. v = πdn = π x0.12 x 29 = 10.93 m/s

5. Choosing cast bronze material for the bearing, the recommended clearance is coming under C curve of Fig.4. 3a.

C- Electric motors, generators, and similar types of machinery using ground journals in broached or reamed cast-bronze bearings (0.4 to 0.8 μm rms finish)
From Fig. 4.3a, the recommended clearance for 120 mm diameter journal is 0.07 mm.

6. $h_o \geq$ 0.005 + 0.00004 d =0.005 +0.00004x120= 0.0098mm

Table 4.1 (a) Unit loads for journal bearings

(a)Relatively steady loads $p = F_{max} / d\ l$

Applications	Unit loads MPa	Applications	Unit loads MPa
Electric motors	0.8 – 1.5	Air compressors Main bearing	1.0 - 2.0
Steam turbines	1.0 – 2.0	Air compressors Crank pin bearing	2.0 – 4.0
Gear reducers	0.8 – 1.5	Centrifugal pumps	**0.6 – 1.2**

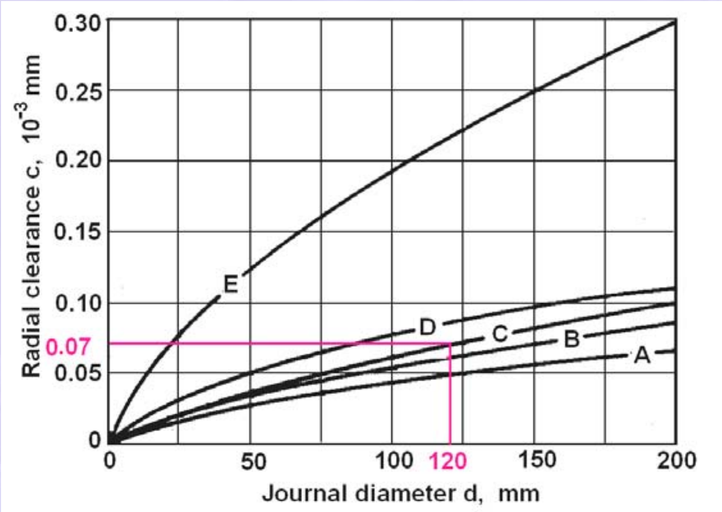


Fig. 4.3a Recommended radial clearance for cast bronze bearings

7. The peak to valley height of roughness R_1 = 1.5 μm for fine ground journal and R_2 = 2.5 μm lapped bearing assumed.

8. $h_o > 0.5 (R_1 + R_2) = 0.5 (1.5+2.5) = 2$ μm

9. Hence , h_o = 0.012 is aimed at which is at least 6 times the average peak to valley roughness of journal and bearing and safe working regime for hydro-dynamic lubrication.

10. The recommended viscosity of oil for the centrifugal pump application is 30 – 80 cP. Hence from the chart SAE 30 oil is chosen.

11. Assuming the bearing to operate between 50 to 60^{O}C and average oil temperature of 55^{O}C, μ = 34 cP from Fig. 2.3e

12. Clearance ratio of ψ for p < 8 MPa and v > 3 m /s. (c/r) =2x10 $^{-3}$ assumed. Or r/c = 500.

13. $$\mathbf{S} = \left(\frac{r}{c}\right)^2 \left(\frac{\mu n}{p}\right) = (500)^2 \left(\frac{34\text{x}10^{-3}\ \text{x}29}{0.833\text{x}\,10^{6}}\right) = 0.296$$

Table 4.2a Clearance ratio: ψ = c/r in 10 $^{-3}$

Working pressure p MPa	Peripheral speed m/s		
	Low < 2	Medium – 2 to 3	High >3
Low to medium p< 8 MPa	0.7-1.2	1.24 – 2.0	2 - 3
High p>8 MPa	0.3 – 0.6	0.8 – 1.4	1.5 – 2.5

Table 4.3a Surface roughness values R_1 and R_2 in μm (peak to valley height of shaft and bearing surface roughness)

Type of machining	Roughness values	Type of machining	Roughness values
Rough turning finish	16 - 40	Fine turning, reaming, grinding, broaching finish	**2.5 – 6.0**
Medium turning finish	6 - 16	Very fine grinding, lapping, honing	1 – 2.5

14. S = 0.296 and l/d = 0.75, $T_{var} = \gamma\ C_H\,(\Delta T/p)$ =26.5 from Fig.2.20c.

15. $\Delta T = 26.5\ p/\ \gamma\ C_H = 26.5 \times 0.833 \times 10^6 / 861\text{x}1760 = 14.6^{o}C$

16. $T_{av} = T_i + 0.5\ \Delta T = 50 + 0.5 \times 14.6 = 57.3^oC$

17. For $T_{av} = 57.3^oC$, $\mu = 31.5cP$ from Fig. 2.1e

18. Recalculated S = 0.274

19. For S = 0.274 and l/d = 0.75, $T_{var} = 24$ from Fig. 2.20d

20. $\Delta T = 24\ p/\ \gamma\ C_H = 24 \times 0.833 \times 10^6 / 861x1760 = 13.2^oC$

21. $T_{av} = T_i + 0.5\ \Delta T = 50 + 0.5 \times 13.2 = 56.6^oC$

22. For $T_{av} = 56.6^oC$, $\mu = 32cP$, S =0 283, $T_{var} = 24$, $\Delta T = 13.8^oC$

22. For $T_{av} = 56.6^oC$, $\mu = 32cP$, S =0.28, $T_{var} = 24$, $\Delta T = 13.8^oC$

23. $T_{av} = T_i + 0.5\ \Delta T = 50 + 0.5 \times 13.8 = 56.9^oC$

25. For $T_{av} = 56.9^oC$, $\mu = 32.5cP$, S =0. 283, $h_o/c = 0.492$; T_{var}=25;
Q / r c n l = 4.45; $Q/Q_{max} = 0.605$; (r/c) f = 6.6;
$P/_{pmax} = 0.42$; $\Phi = 54.8^o$; $\theta po = 78^o$; $\theta_{pmax} = 17.8^o$;

26. $h_o = 0.492 \times c = 0.492 \times 0.12 = 0.059$ mm

27. $f = 6.6(c/r) = 6.6x\ 2.0 \times 10^{-3} = 0.0132$

28. $\Delta T = 25\ p/\ \gamma\ C_H = 24 \times 0.833 \times 106 / 861x1760 = 13.74^oC$

29. $T_{av} = T_i + 0.5\ \Delta T = 50 + 0.5x\ 13.74 = 56.87^oC = 56.9^oC$

30. Q = 4.45 x rcnl = 4.48 x .06 x0.00012x29x0.08
$= 7.43 \times 10^{-5}\ m^3/s = 73.4\ cm^3/s$

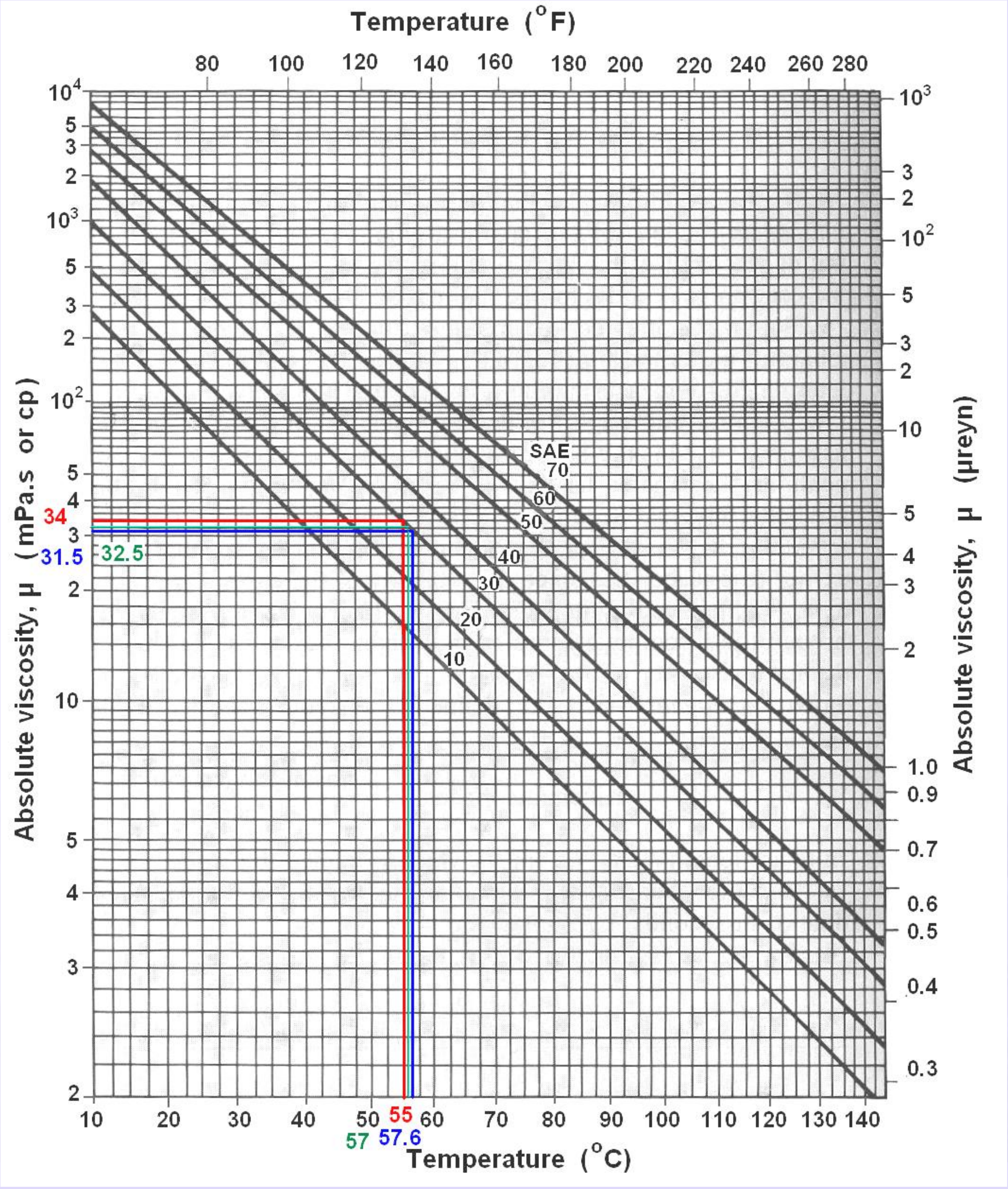


Fig.2.3e Viscosity – temperature curves of SAE graded oils

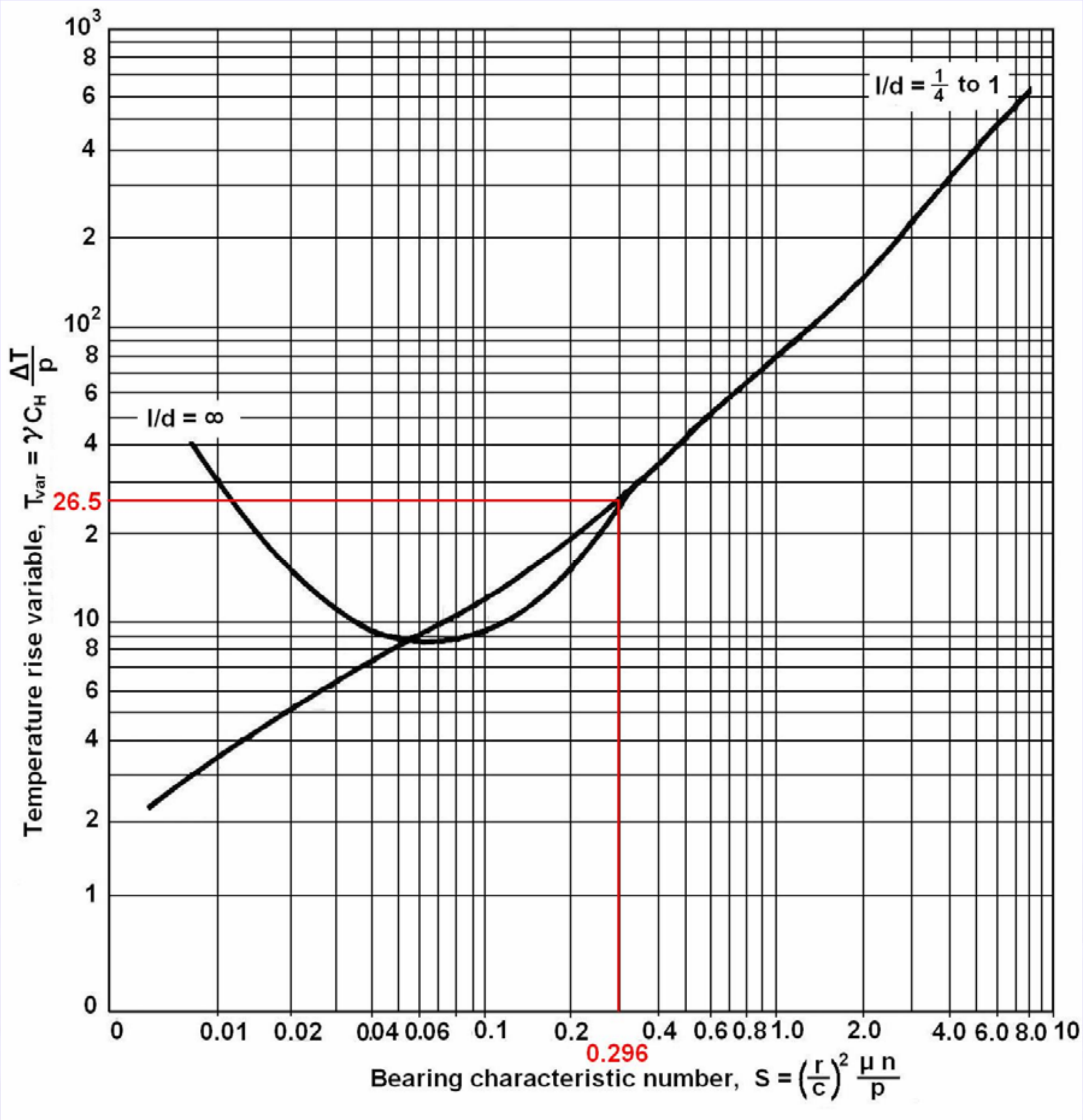


Fig. 2.20c Chart for temperature variable, $T_{var} = \gamma\ C_H\ (\Delta T/p)$

31. $Q_s = 0.605 \times 73.4 = 45\ cm^3/s$

32. $p_{max} = p/0.42 = 0.833/0.42 = 1.98$ MPa

Bearing diameter: 120 H7 - 120.00 / 120.035

Journal diameter-120 f8 -119.964 / 119.910

Fit = 120 H7/f8

33. Frictional power loss: f.Fv = 0.0132x8000x10.93=1154 W

Final details of the designed bearing are given in tabular form in Table 4.4

Table 4. 4 Final details of the designed bearing

d=120mm	l = 80mm	l/d = 0.75	SAE 30 oil	C= 120μm
h_o =59 μm	p=0.833MPa	p_{max}=1.98MPa	T_{av}=56.9^oC	T_i = 50^oC
φ = 54.8^o	**θ_{pmax} = 17.8^o**	**θ_{po} =78^o**	Q =73.4cc/s	Q_s=45 cc/s
Bearing material	Cast Bronze Reamed and honed	f = 0.0132 Fit 120 H7/ f8	Journal Hardened & ground	T_H =63.8^oC μ = 32.5 cP

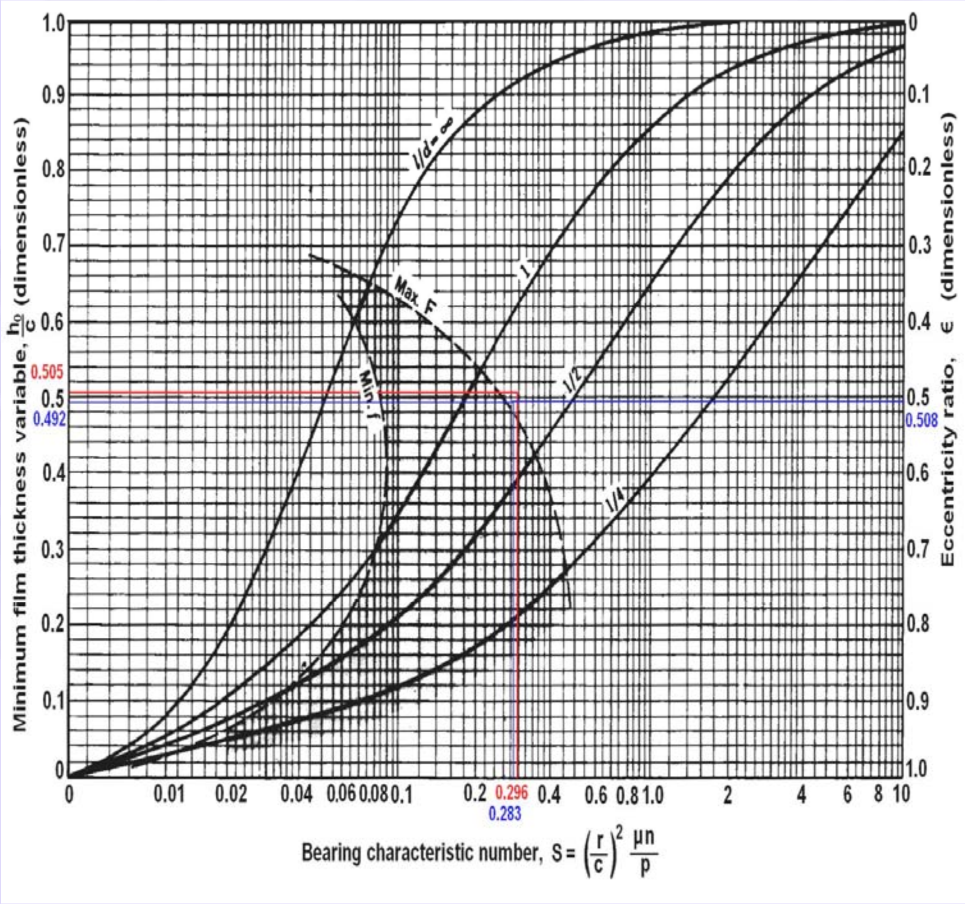


Fig.2.8b Chart for minimum film thickness variable and eccentricity ratio. The left shaded zone defines the optimum h_o for minimum friction; the right boundary is the optimum h_o for maximum load

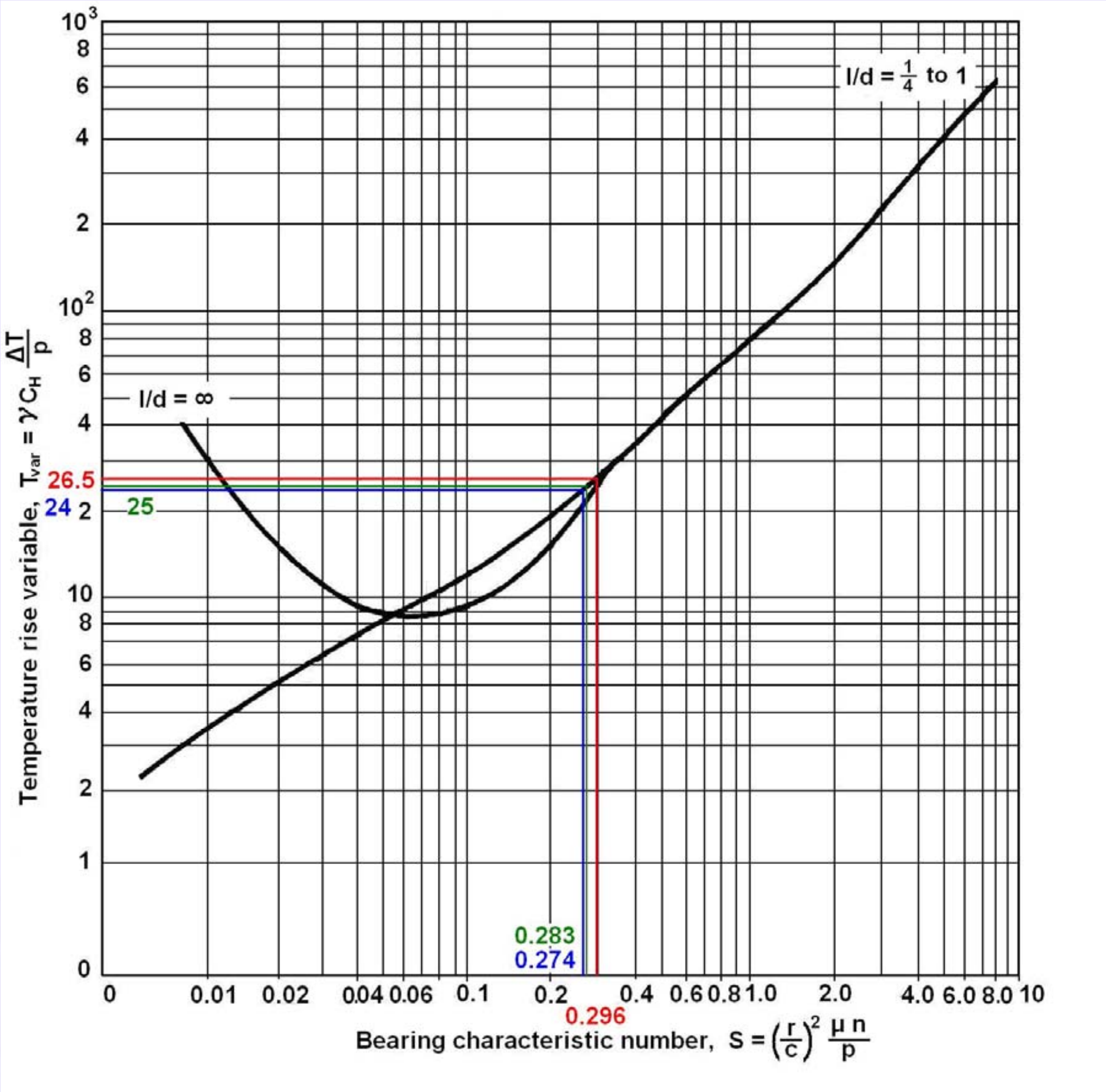


Fig. 2.20d Chart for temperature variable, $T_{var} = \gamma\ C_H\ (\Delta T/p)$

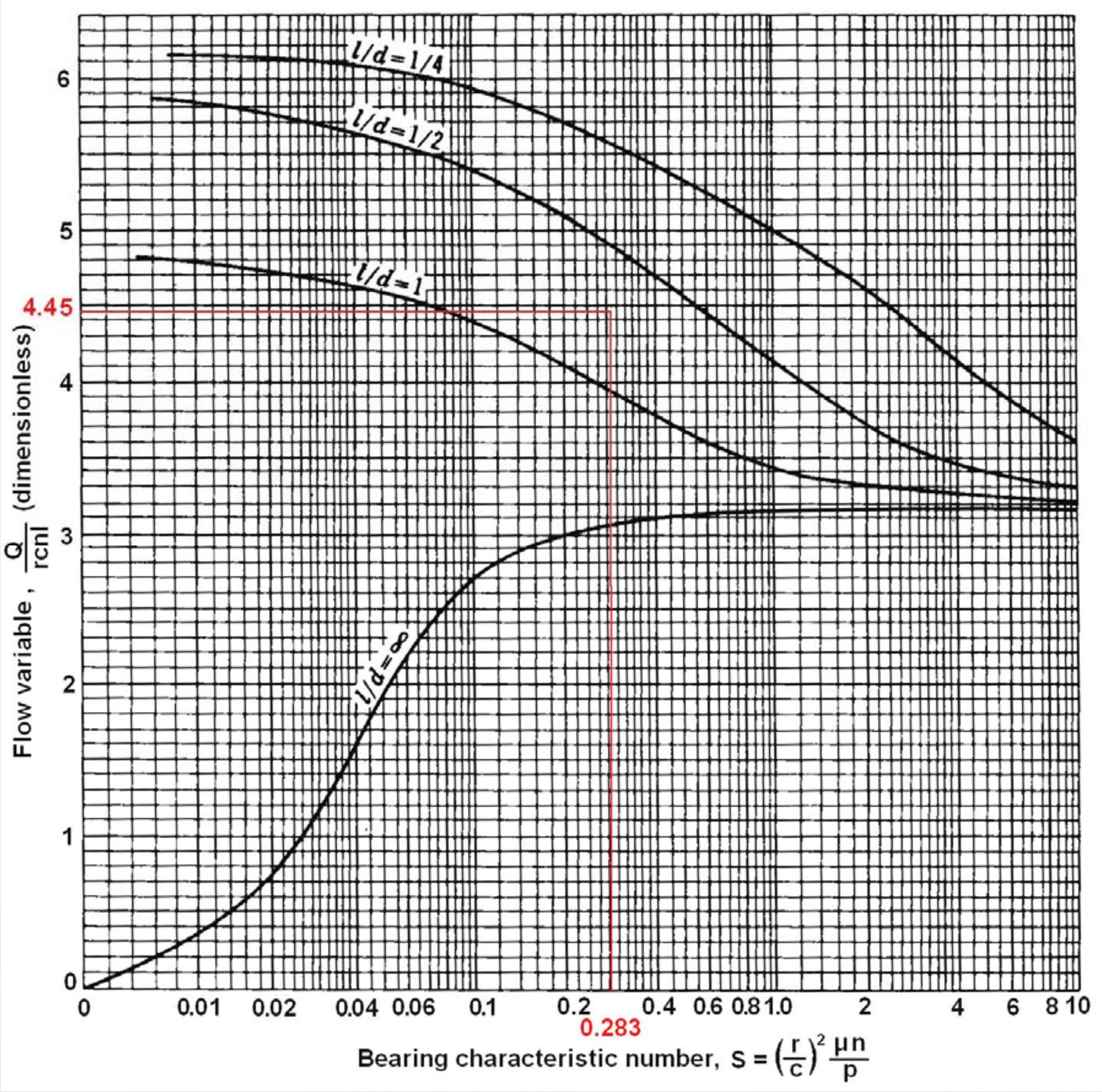


Fig. 2.12b Chart for flow variable.

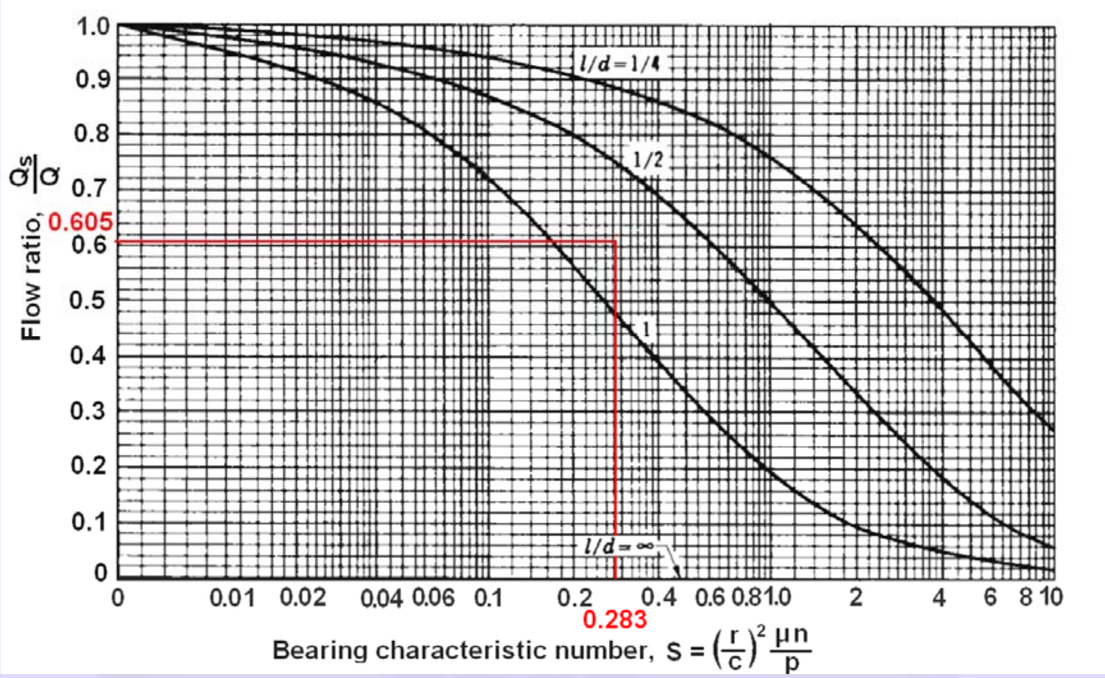


Fig.2.13b Chart for determining the ratio of side flow to total flow

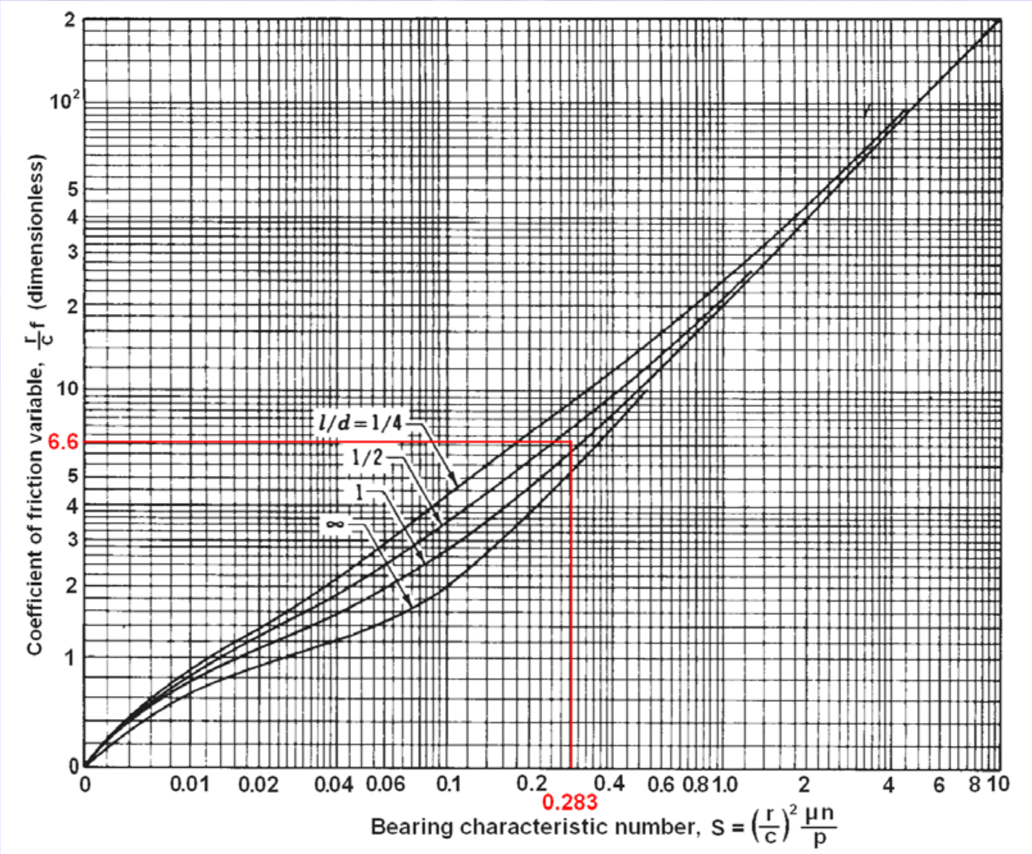


Fig. 2.11b Chart for coefficient of friction variable

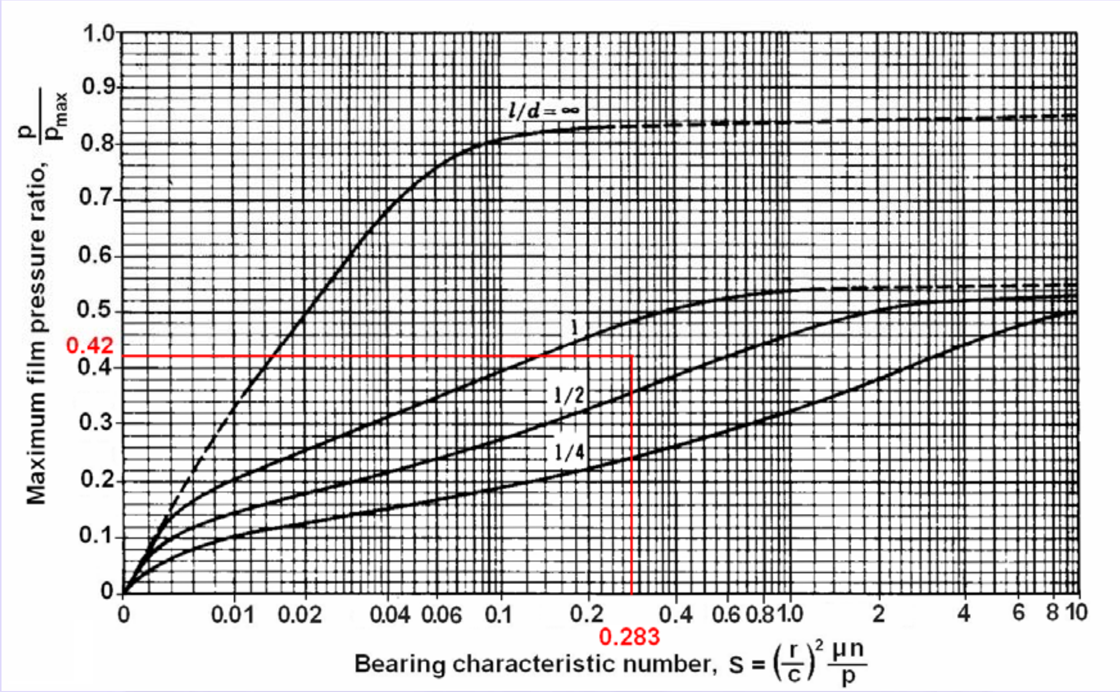


Fig. 2.14a Chart for determining the maximum film pressure

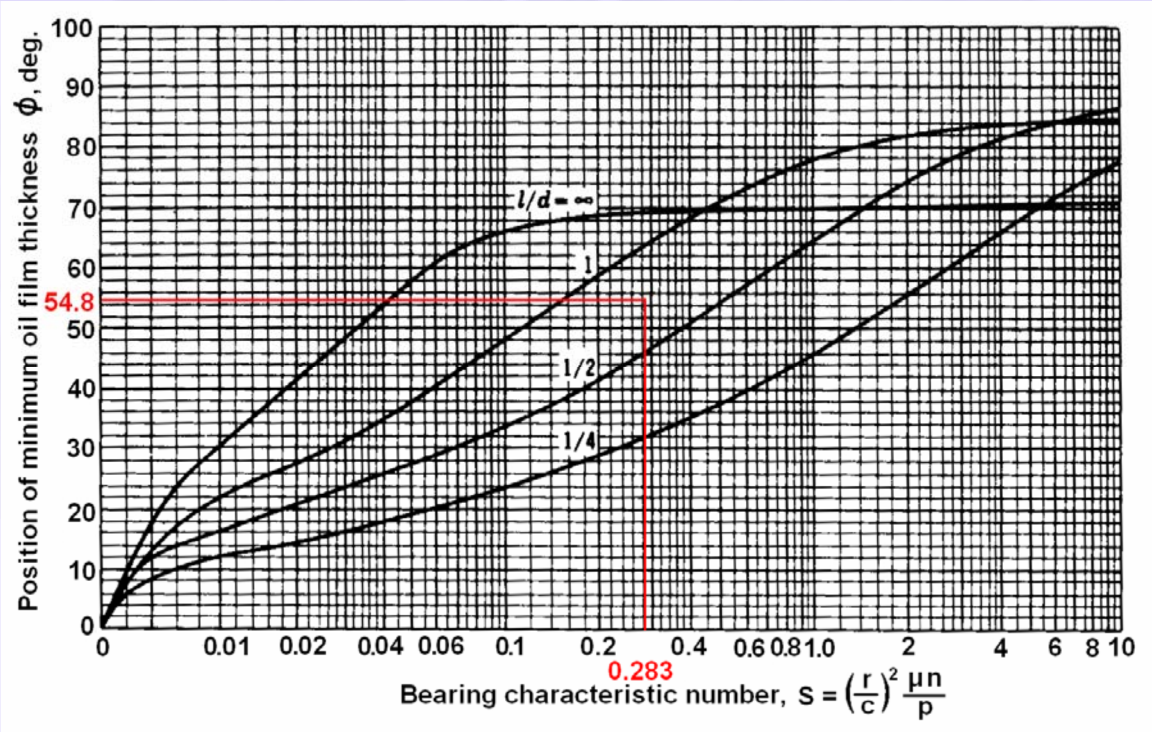


Fig.2.9b Chart for determining the position of minimum film thickness h_o

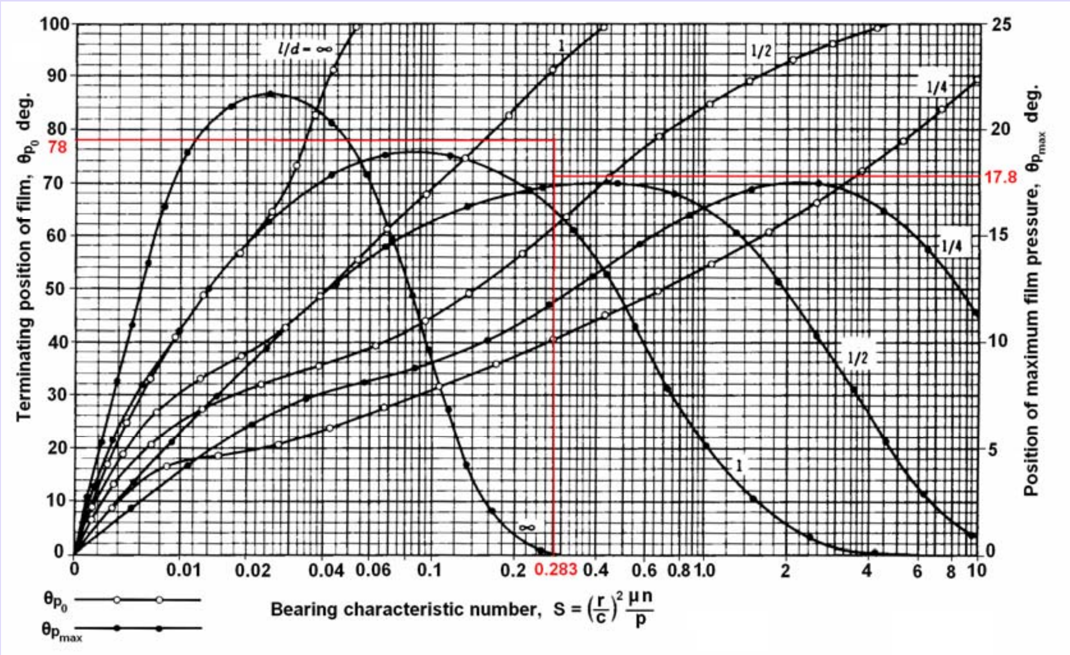


Fig. 2.15b Chart for finding the terminating position of oil film and position of maximum film pressure

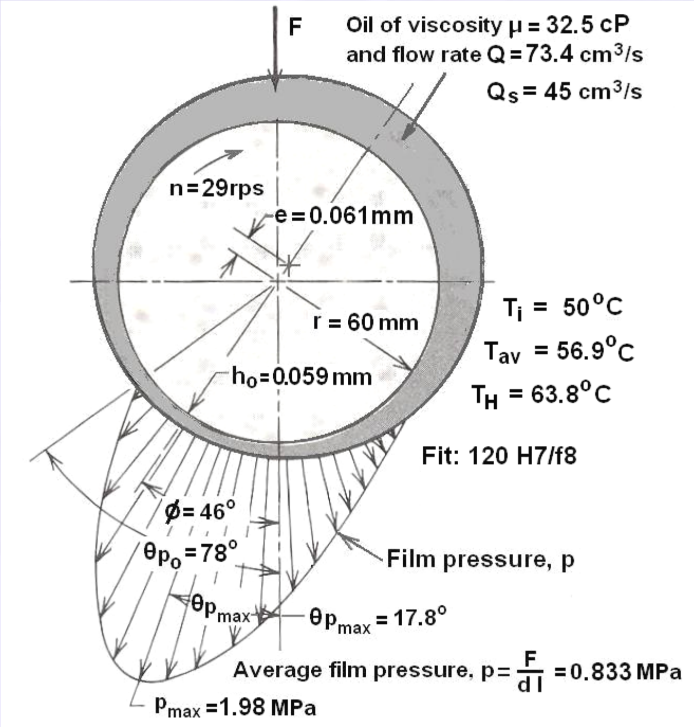


Fig 4.4 Journal position under stable hydrodynamic lubrication condition problem1

----- End of problem 1---

4.3 BOUNDARY AND MIXED-FILM LUBRICATION

There are many bearings in several machineries which run at relatively low speeds and high loads. Under these unfavorable conditions, hydrodynamic pressure developed is inadequate to support the load and they operate under either mixed-film or boundary lubricated conditions as depicted in the Stribeck curve shown in Fig. 4.5. Bearings operating in this regime have extensive metal-to-metal contact and partial hydrodynamic lubrication.

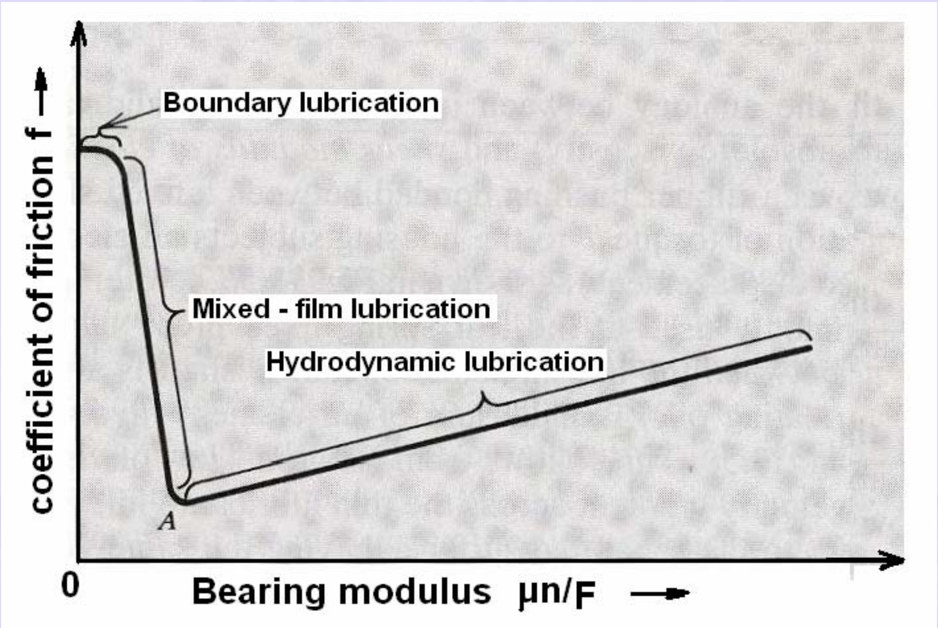


Fig. 4.5 Stribeck curve for bearing friction

The typical hydrodynamic, mixed and boundary lubricated surfaces are depicted in Fig. 4.6(a), (b) and (c).

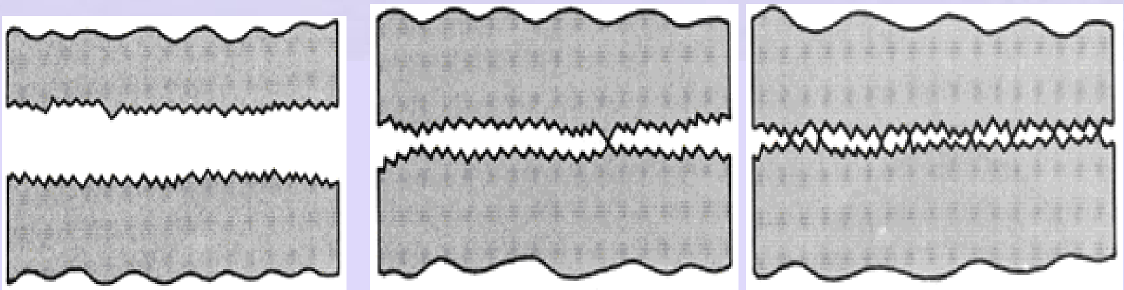


Fig. 4.6(a) Hydrodynamic lubrication **Fig. 4.6(b) Mixed film lubrication** **Fig. 4.6(c) Boundary lubrication**

Hence, in boundary lubricated regime to keep the adhesive wear low, oils with some amount of blend with solid lubricants like MoS_2, Teflon and graphite are quite often used. Since wear is proportional to the frictional work done or pv value, the design is based on this factor.

Further to prevent cold flow of the bearing material, p_{max} should be less than the permissible value for the material and the maximum sliding velocity is also limited to permissible value for the material, as it increases the dynamic load. Hence for a good design,

$$(p\ v) \le (p\ v)_{max} \qquad (4.\ 2)$$

permissible value of , $p \le p_{max}$ (4.3)

and $v \le v_{max}$ (4.4)

The choice of journal and bearing material pairing play vital role in design apart from the lubricant in reducing adhesive wear, seizure, scoring etc. The permissible value of the pv, p and v for different materials are given Table 4.2.

Another important criterion which should not be forgotten in bearing design is thermal aspect.

$$\mathbf{pv = \frac{k\,(T_B - T_A)}{f_m}} \qquad \mathbf{(4.5)}$$

Where p is the unit load Pa (N / m^2)

v is the surface velocity of journal relative to bearing m/s

T_A is the ambient temperature of the air ^oC

T_B is the bearing temperature ^oC

k is the constant that depends upon the ability of the bearing to dissipate the heat. A best estimate of the k value is from the previous design application and working performance. A rough estimate done by considering maximum pv value and minimum friction in Fig. 3.6 and maximum pv value from Table 4.5.

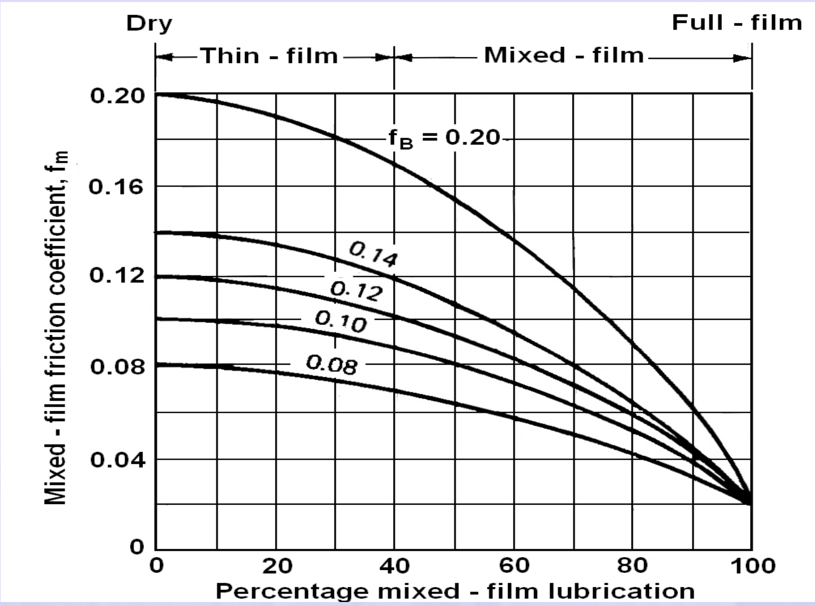


Fig. 4.6 Coefficient of friction under various percentage of mixed - film lubrication

Table 4.5(a) Bearing material properties

Material	Maximum pressure p_{max} MPa	Maximum Temperature T_{Bmax} ^oC	Maximum Speed V_{max} m/s	Maximum pv value MPa.m/s
Cast Bronze	31	165	7.5	1.75
Sintered bronze	31	65	7.5	1.75
Sintered Fe	55	65	4	1.75
Pb-bronze	24	150	7.6	2.1
Sintered Fe-Cu	28	65	1.1	1.2

Table 4.5(b) Bearing material properties

Material	Maximum pressure p_{max} MPa	Maximum Temperature T_{Bmax} ^oC	Maximum Speed V_{max} m/s	Maximum pv value MPa.m/s
Cast iron	4	150	1.5	0. 5
Hardenable Fe-Cu	55	--	0.2	2.6
Bronze-iron	17	--	4.1	1.2
Lead- iron	7	--	4.1	1.8
Aluminium	14	--	6.1	1.8

Table 4.5(c). Bearing material properties

Material	Maximum pressure p_{max} MPa	Maximum Temperature T_{Bmax} ^oC	Maximum Speed V_{max} m/s	Maximum pv value MPa.m/s
Phenolics	41	93	13	0.53
Nylon	14	93	3	0.11
TFE	3.5	260	0.25	0.035
Filled TFE	17	260	5.1	0.35
TFE fabric	414	260	0.76	0.88

Table 4.5(d) Bearing material properties

Material	Maximum pressure p_{max} MPa	Maximum Temperature T_{Bmax} ^oC	Maximum Speed V_{max} m/s	Maximum pv value MPa.m/s
Polycarbonate	7	104	5.1	011
Acetal	14	93	3	0.11
Carbon graphite	4	400	13	0.53
Rubber	0.35	66	20	-------
Wood	14	71	10	0.42

In boundary lubricated bearing considerable sliding wear takes place and it decides the life of the bearing. The sliding wear 'w' (in mm) is given by

$$\mathbf{w = K \times p \times v \times t} \qquad (4.6)$$

Where K – specific wear, mm / (MPa). (m/s).h

K depends on the type of load and lubrication.

p – load per unit area MPa

v – sliding velocity = π d n / 60, m/s

t - sliding time in hours

Table 4.6 Properties of Oiles 500 bearing under continuous oil lubrication

p_{max} MPa	25
v_{max} m/s	0.3
$(pv)_{max}$ MPa.ms^{-1}	1.636
T_{max} ^oC	90
f	0.03
K (specific wear) mm/MPa.ms^{-1}.h	6 – 30 x 10^{-6}

Lower values of K refer to oil lubricated bearings with ground journal and steady load. Higher values refer to Oscillatory loads.

4.4. BOUNDARY AND MIXED-FILM LUBRICATED BEARINGS- PROBLEM 1

A bush bearing has to operate under boundary lubricated condition with a radial load of 150 N and speed of 4 rps. Its wear should be less than 0.03 mm in 5000 h of operation. Maximum operating temperature is 85^{o}C. Factor of safety desired is 2. Choose suitable oiles bearing for the application. Assume an air temperature of 30^{O}C.Take k = 15.3 W/m^2. ^{o}C

Data: $F = 150$ N ; $n = 4$ rps ; $w = 0.03$ mm;

$t = 5000$h; $T_{max} = 85^oC$; f.s. = 2; $T_A = 30^oC$; $k = 15.3\ W/m^2.\ ^oC$

Solution:

1. For Oiles 500 bearing $p_{max} = 25$ MPa;

$v_{max} = 0.3$ m/s;$(pv)_{max} = 1.636\ MPa.ms^{-1}$ from Table 8.

2. We will take (id) d = 18 mm, od D= 28 mm and

l = 25 mm available standard bearing as a first trial from Olies catalog from net.

3. $p = F/dl = 150/ 18 \times 25 = 0.333$ MPa < 25 MPa OK

4. $v = \pi\ d\ n = \pi \times 18 \times 4 \times 10^{-3} = 0.226$ m/s < 0.3 m/s OK

5. $pv = 0.333 \times 0.226 = 0.075\ MPa.ms^{-1} < 1.636$, $(pv)_{max}$ OK.

6. Check for thermal aspects:

Assuming a wall thickness of 7.5 mm for the housing, the surface area A is given by

$A = \pi\ D_H\ L + 2\pi\ (\ D_H^2 - d^2)/4\ ] \times 10^{-6}\ m^2$

$= [\pi\ (\ 28 + 15)\ 25\ + 0.5\ \pi\ (43^2 - 18^2) \times\ 10^{-6}$

$= 5.77 \times 10^{-3}\ m^2$

$F\ f\ v = k\ A\ (T_B - T_A)$

$150 \times 0.03 \times 0.226 = 15.3 \times 5.77 \times 10^{-3} \times (T_B - 30)$

$T_B = 30 + 11.5 = 41.5\ ^oC < T_{max}\ (85^oC)$ OK

7. Check for wear:

$w = K \times p \times v \times t$

$K = 30 \times 10^{-6}$ worst case is assumed from Table 8.

$w = 30 \times 10^{-6} \times 0.333 \times 0.226 \times 5000$

$= 0.011$ mm < 0.03 mm

hence from wear consideration also the selection of bearing is satisfied. The factor of safety is more than 2 here. This indicates that the chosen bearing Oiles id 18 x od 28 x length 25 mm is adequate for the operation with a factor of safety.

4. 4 THRUST BEARINGS

When shaft **axial loads** are **great** (as with vertical shafts of substantial weight, and propeller shafts subjected to substantial thrust loads),hydrodynamic thrust bearings can be provided which is shown in the following figure.

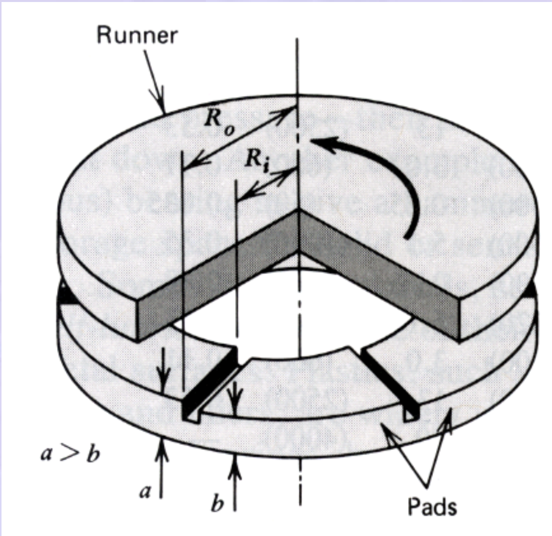


Fig 4.7 Thrust Bearing

a. Oil supplied to the inside diameter of the rotating *collar* or *runner* flows outward by centrifugal force through the bearing interface.

b. As the oil is dragged circumferentially through the bearing, it experiences a wedging action, which is due to the tapered pads on the stationary member.

c. This is directly analogous to the wedging action produced by the eccentricity of a journal bearing.

d. As in figure, the fixed pads may have a fixed taper angle, or the pads may be pivoted and allowed to assume their own optimum tilt angle, or they may be partially constrained and permitted a small variation in tilt angle.

e. If the pads have a fixed taper, it is obvious that a load can be supported hydrodynamically for only one direction of rotation.
