Module 5 – SLIDING CONTACT BEARINGS

Lecture 3 – HYDRODYNAMIC LUBRICATION OF JOURNAL BEARINGS THEORY AND PRACTICE

Contents

3.1. LUBRICANT SUPPLY

Lubricant present at the bearing surface gets depleted due to side leakage and to main the hydrodynamic lubrication continuous supply of lubricant must be ensured. The principal methods of supply of lubricant are:

1. Oil Ring lubrication
2. Oil collar lubrication
3. Splash lubrication
4. Oil bath lubrication
5. Oil pump lubrication

3.1.1. Oil Ring lubrication

Fig.3.1 shows an oil ring lubricated bearing. The ring of 1.5 to 2 times the diameter of the shaft hangs loosely on journal. As it rotates with the journal, it lifts oil to the top. The bearing sleeve is slotted to accommodate the ring and bear against the journal. This method of lubrication has been found efficient in many applications.

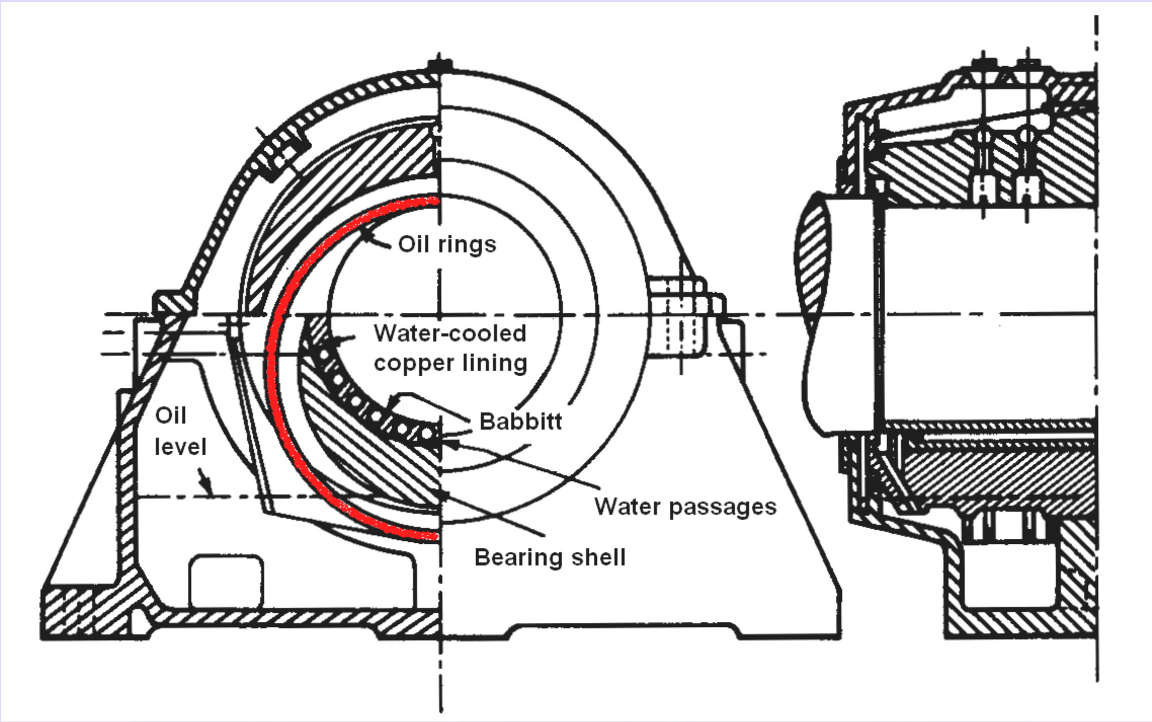

Fig.3.1 Oil ring lubricated bearing with water cooling

3.1.2 Oil collar lubrication

This case a rigid collar integral with the journal as shown in Fig.3.2 dips into the reservoir at the bottom. During rotation it carries the oil to the top and throws off into a small sump on either side of the collar. From there it flows by gravity through the oil hole and groove to the bearing surface as shown in Fig.3.3.

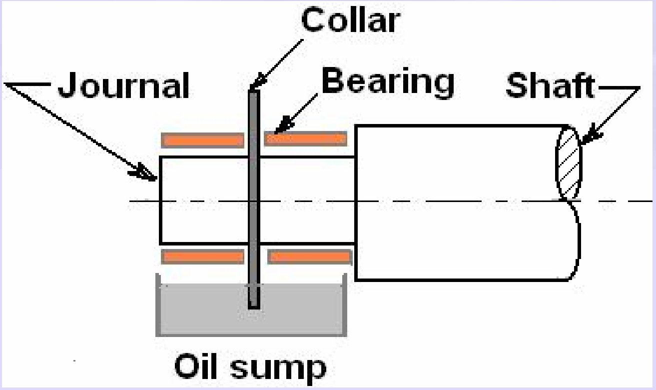

Fig. 3.2 Oil collar lubrication

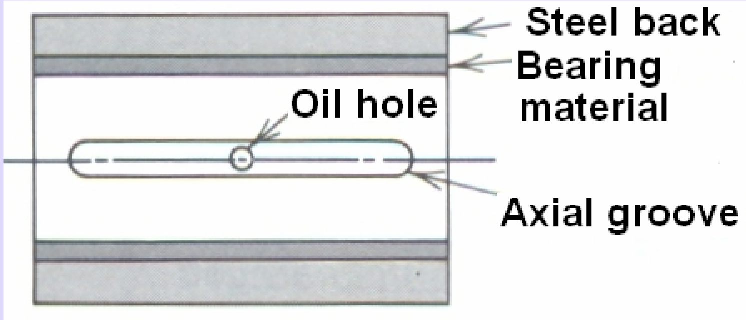

Fig. 3.3 Bearing with oil hole and axial groove

3.1. 3 Oil splash lubrication

In some machines, oil is splashed by rapidly moving parts can be channeled to small sumps maintained above the bearings. Besides this, small oil scoops on rotating parts can dip into the main oil sump and thereby carry that flow into bearings. Typical examples of this can be seen in automobile engine wrist pin lubrication wherein the crank splashes oil when it dips into the oil sump below. Another example is lubrication of the bearings of gearboxes wherein the gears splash the oil into bearings.

3.1.4 Oil pump lubrication

This is a positive means of supplying oil. Fig.3.4 shows the pressure fed lubrication system of a piston engine or Compressor. Pumped oil fills the circumferential grooves in the main bearings. Through the holes in crankshaft oil is then carried to the connecting rod bearings. Circumferential groove in them transmits the oil through riffle drilled holes to the wrist pin bearings.

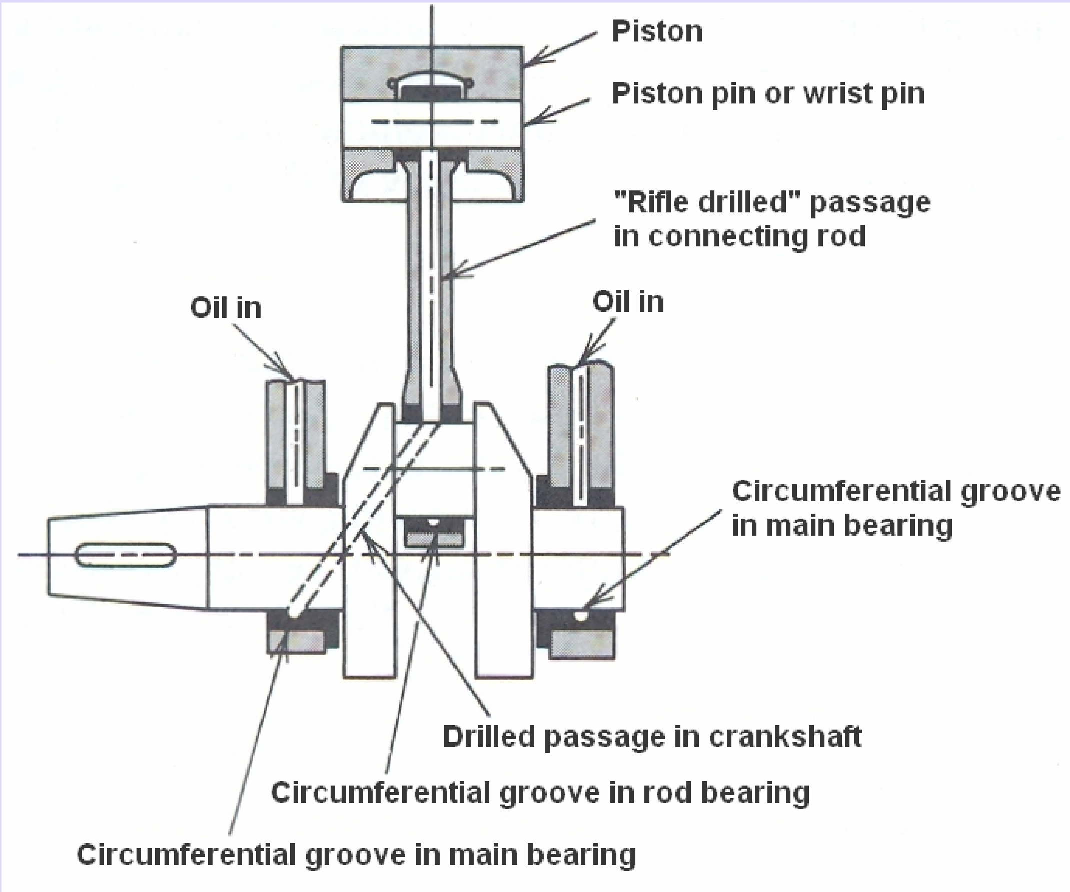

Fig. 3.4 Oil pump lubrication of an engine crank shaft

In many automobiles to reduce the cost and also weakening the crankshaft, riffle drilled holes is eliminated and the wrist pins are splash lubricated.

3.2 HEAT DISSIPATION AND EQUILIBRIUM OIL TEMPERATURE

Another important consideration in hydrodynamic lubrication is thermal aspect of design. The heat generated in the bearing should be effectively dissipated so that the equilibrium conditions are reached in a short time. Further, the average or equilibrium temperature of the oil should not exceed 93 to 123^{o}C to prevent quick deterioration of the oil.

The frictional heat generated can be found from the load (F) coefficient of friction (f), and the journal speed (n).

$v = 2\pi nd / 60000$ rad / s where n is in rpm & d in mm.

Frictional power loss: $H_g = F f v$ (3.1)

Where H_g is expressed in Nm/s or W

The oil temperature rise can be estimated from chart in Fig.2.20 devised by Raimondi and Boyd or from the heat balance equation in the case of self contained bearings as in the case of ring, collar or oil bath lubrication. Industrial applications of self contained bearings can be seen in fans, blowers, pumps, motors and so on.

$$T_{var} = \gamma C_H \left(\frac{\Delta T}{p} \right) \qquad (3.2)$$

Where γ is the density of the oil 861 kg /m^3

C_H is the specific heat of the oil, an average value of 1760 J/ kg. ^oC may be taken.

ΔT is the temperature rise ^oC and P is the film pressure in Pa.

Heat dissipated: **$H_d = C A (T_H - T_A)$ (3.3)**

Where, H_d = in W or Nm/s

C = combined the heat transfer coefficient (radiation and convection), W/m^2 .^OC

A = exposed surface area of the housing, m^2

= 20 d l

T_H = surface temperature of the housing, ^oC

T_A= temperature of surrounding air, ^oC.

The value of C depends on the material, colour, geometry and roughness of the housing, temperature difference between the housing and surrounding objects and temperature and velocity of the air.

C = 11.4 W/m^2.^{o}C for still air

C = 15.3 W/m^2.^{o}C for average design practice

C = 33.5 W/m^2.^{o}C for air moving at 2.5 m/s

An expression similar to eqn. (3.3) can be written between the temperature difference $T_o - T_H$ between the lubricant oil film and the housing.

The relationship depends on the lubrication system and the quality of lubricant circulation. Oil bath lubrication system in which a part of the journal is immersed in the lubricant provides good circulation. A ring oiled bearing in which oil rings ride on top of the journal or an integral collar on journal dip into the oil sump and provides fair circulation for many purposes. Wick feeding will result in inadequate circulation and should be limited to very light load application and is not considered here.

$$\mathbf{T_O - T_H = b\,(T_H - T_A)} \qquad \mathbf{(3.4)}$$

where T_O is the average oil film temperature and b is a constant depending on lubrication system. Since T_O and T_A are known, combining eqn. (3.3) & (3.4),

$$\mathbf{H_d = C\,A\left(\frac{1}{b+1}\right)(T_O - T_A)} \qquad \mathbf{(3.5)}$$

$$\mathbf{H_d = C\,A\,B\,(T_O - T_A)} \qquad \mathbf{(3.6)}$$

Where B = 1/ (b+1) and a rough estimate of this is given in Table 3.1.
In heat balance computation, the oil film temperature and hence the viscosity of the lubricant in a self contained bearing are unknown. The determination is based on iterative process where the heat generated and heat dissipated match giving the equilibrium temperature. This is a time involving procedure.

Table 3.1 Value of the constant B

Lubrication system	Condition	Range of B
Oil ring	Moving air	0.333 - 0.500
Oil ring	Still air	0.667 – 0.500
Oil bath	Moving air	0.667 – 0.500
Oil bath	Still air	0.714 -0.833

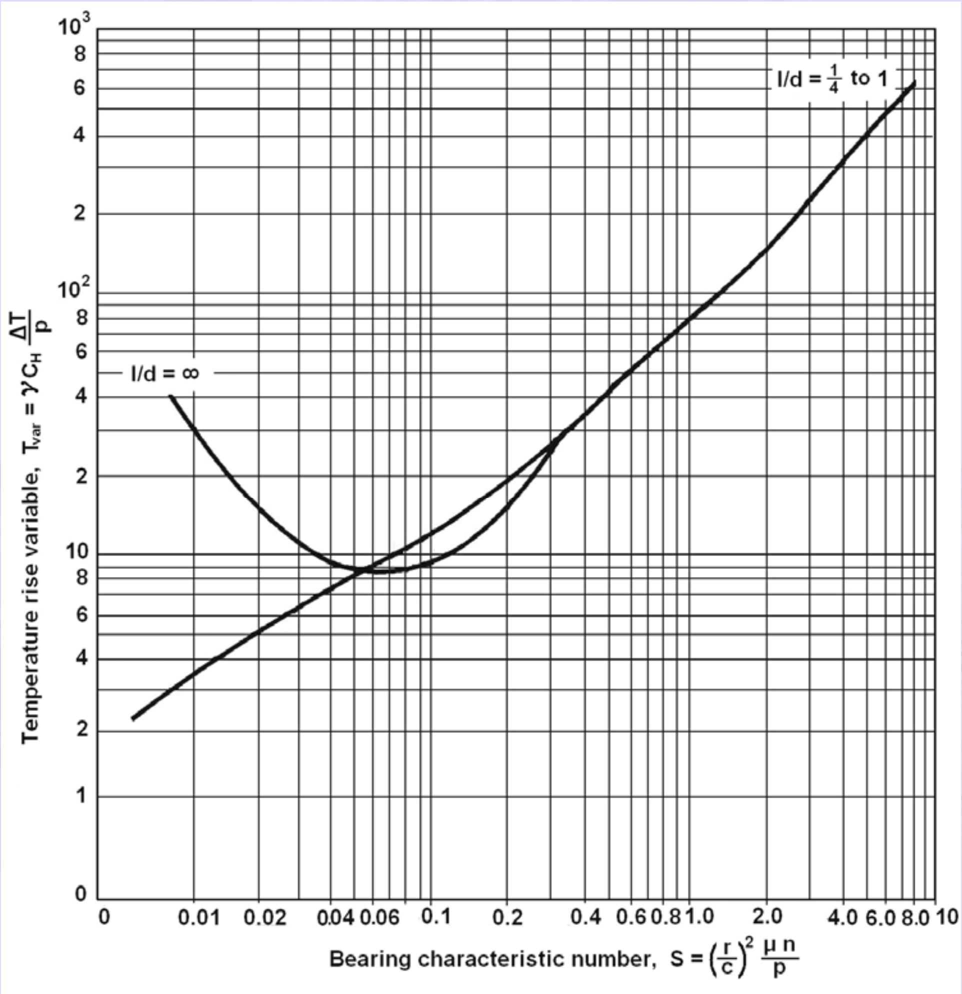

Fig.3. 5 Chart for temperature variable, $T_{var} = \gamma C_H (\Delta T/p)$

The use of this chart will be illustrated with worked out problems in arriving at equilibrium temperature.

3.3 ANALYSIS OF HYDRODYNAMIC LUBRICATED BEARING USING CHARTS – Problem 1

A journal of a stationary oil engine is 80 mm in diameter and 40 mm long. The radial clearance is 0.060mm. It supports a load of 9 kN when the shaft is rotating at 3600 rpm with SAE 40 oil supplied at atmospheric pressure and assume average operating temperature is about 65^{o}C as first trial for inlet oil temperature of 45^{o}C. Using Raimondi-Boyd charts analyze the bearing temperature under steady state operating condition.

Data: d = 80 mm; l =40 mm; c = 0.06 mm; F = 9kN;
n = 3600rpm = 60 rps; SAE 40 oil; T_o = 65^{O}C; T_i=45^{o}C.

Analysis:

1. p= F /l d = 9 x1000 /80 x 40 = 2.813 MPa

2. μ = 30 cP at 65^{o}C for SAE 40 oil from graph 2.3(a).

3. $$\mathbf{S}=\left(\frac{\mathbf{r}}{\mathbf{c}}\right)^2\left(\frac{\mu\mathbf{n}}{\mathbf{p}}\right)=\left(\frac{\mathbf{40}}{\mathbf{0.06}}\right)^2\left(\frac{\mathbf{30x10^{-3}\ x\ 60}}{\mathbf{2.813\ x\ 10^6}}\right)=\mathbf{0.284}$$

4. For S = 0.284 and l /d = ½, T_{var} = 25 from Fig.3.5 (a).

5. Rewriting the equation 2.23, $$\mathbf{T_{var}}=\gamma\,\mathbf{C_H}\left(\frac{\Delta\mathbf{T}}{\mathbf{p}}\right)\qquad\mathbf{(2.23)}$$

$$\Delta\mathbf{T}=\frac{\mathbf{T_{var}\,p}}{\gamma\,\mathbf{C_H}}=\frac{\mathbf{25\ x\ 2.813\ x\ 10^6}}{\mathbf{861\ x\ 1760}}=\mathbf{46^\circ C}$$

6. T_{av} = T_i + 0.5 ΔT = 45 + 0.5 x 46 = 68^{o}C

7. At T_{av} = 68^{o}C, μ = 26 Pa.s from Fig. 2.3(b)

8. S = 0.246, for this T_{var} = 22.5 from Fig. 3.5 (b), calculated value of $\Delta T = 41.4^{o}C$

9. Tav = Ti + 0.5 ΔT = 45+0.5x41.4 = $65.7^{o}C$. Hence equilibrium temperature will be about $66^{o}C$.

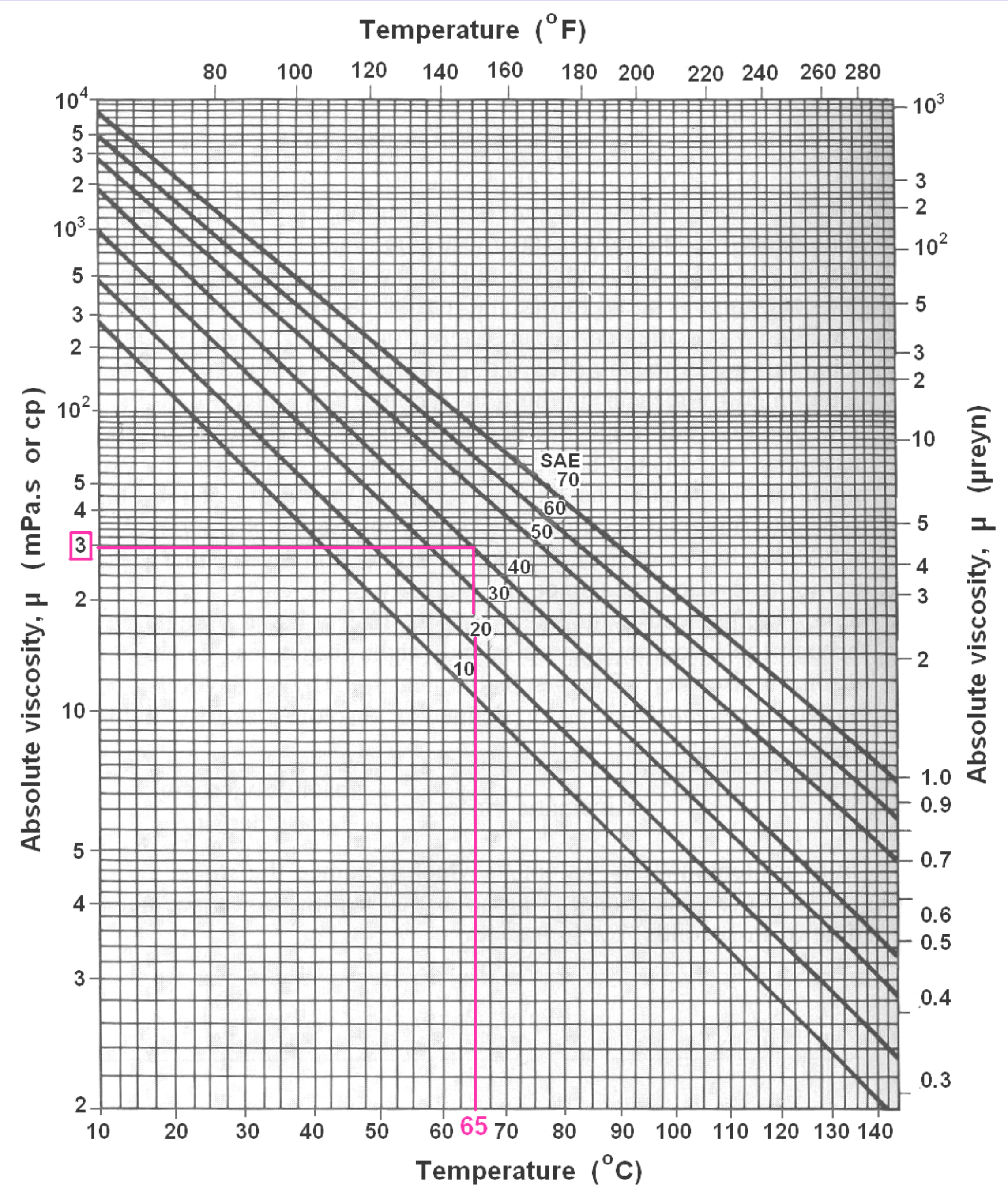

Fig. 2.3(a) Viscosity – temperature curves of SAE graded oils

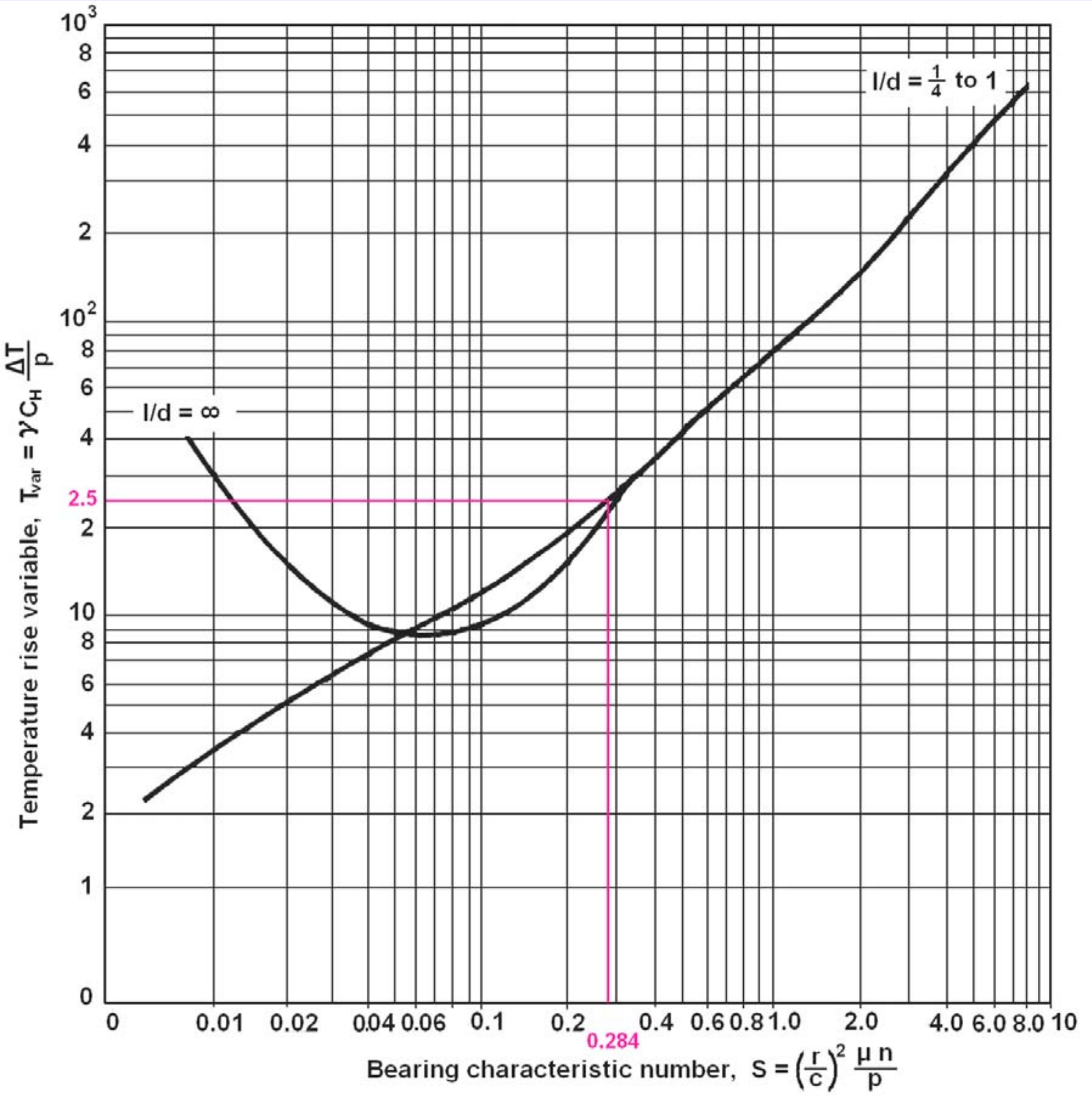

Fig. 3.5(a) Chart for temperature variable, $T_{var} = \gamma\ C_H\ (\Delta T/p)$

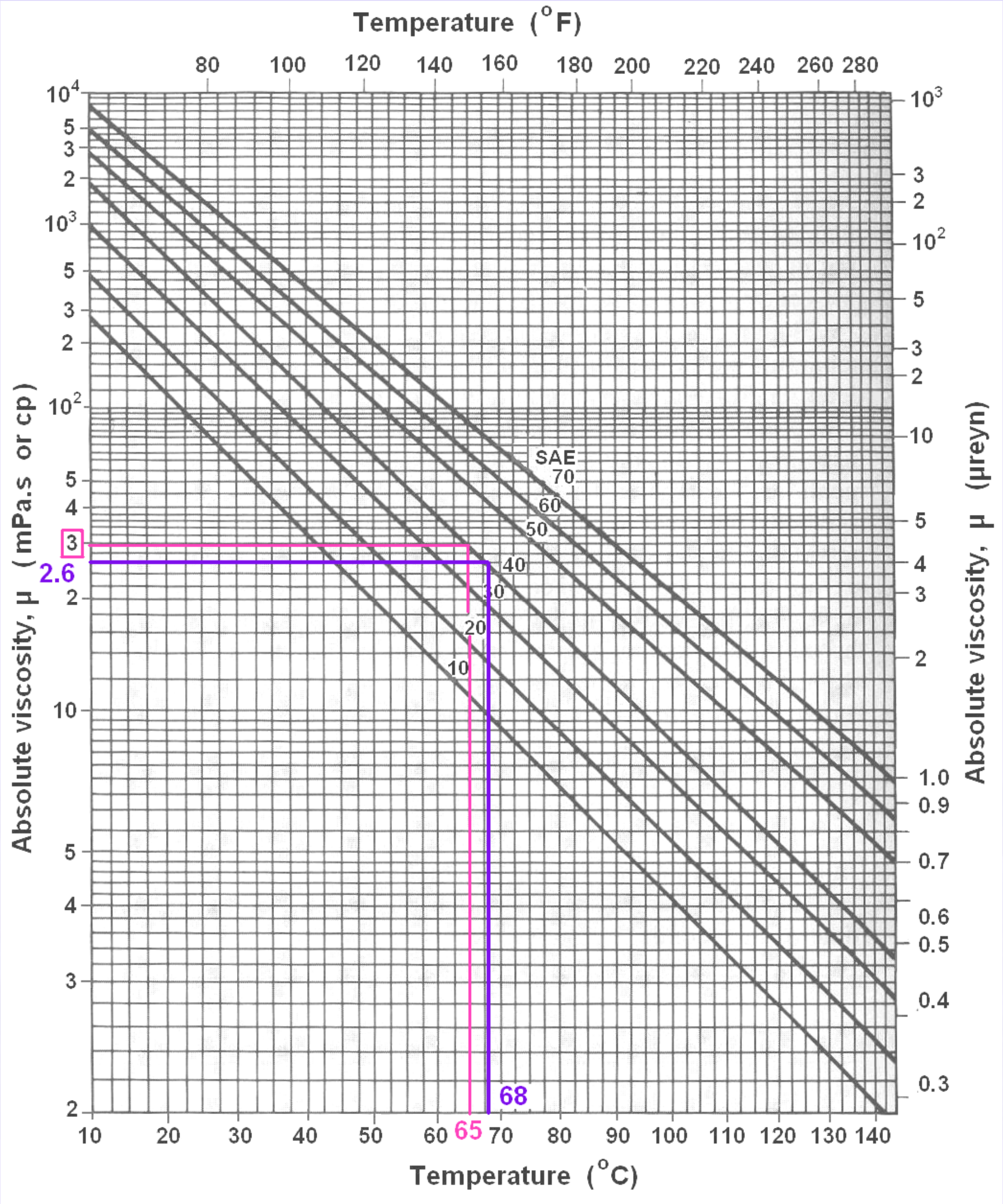

Fig. 2.3b Viscosity – temperature curves of SAE graded oils

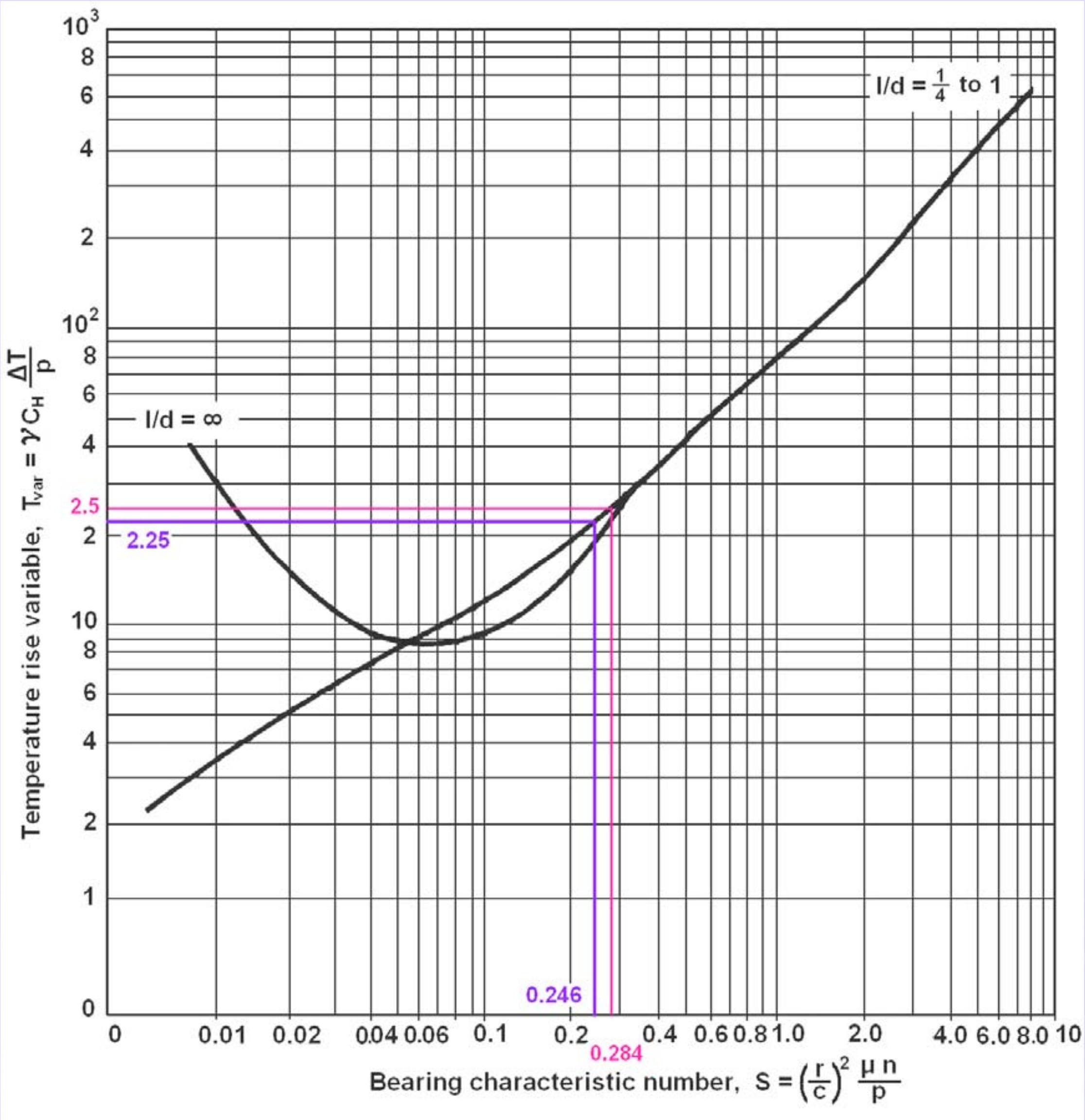

Fig. 3.5(b) Chart for temperature variable, $T_{var} = \gamma\ C_H\ (\Delta T/p)$

3.3 HEAT DISSIPATION AND EQUILIBRIUM OIL TEMPERATURE USING CHARTS – PROBLEM 2

A sleeve bearing is 40 mm in diameter. and has a length of 20 mm. The clearance ratio is 1000, load is 2.5 kN, and journal speed is 1200 rpm. The bearing is supplied with SAE 30 oil. The ambient temperature is 35^{o}C. Determine the average oil film temperature in equilibrium condition, assuming that the bearing is lubricated by an oil bath in moving air.

Data: d = 40 mm; l = 20 mm; r/c = 1000; SAE 30 Oil;

$T_A = 35^oC$; Lubrication is by oil bath in still air.

Analysis:

1. p = F /d l = $2.5 \times 10^3 / 0.04 \times 0.02 = 3.13 \times 10^6$ Pa

2. Expecting the oil average temperature to be 60^oC

μ = 26.5 cP or mPa.s for SAE 30 oil. From Fig.2.3c

3. n = 600/60 = 10 rps.

4. $$\mathbf{S = \left(\frac{r}{c}\right)^2\left(\frac{\mu n}{p}\right) = 1000^2\left(\frac{26.5x10^{-3}\,x\,10}{3.13\,x\,10^6}\right) = 0.085}$$

5. $\mathbf{\frac{r}{c}f = 3.05}$ for S = 0.085 and (l /d) = 0.5 from Fig.2.11b.

6. $$\mathbf{f = 3.05\frac{c}{r} = 3.05x10^{-3} = 0.00305}$$

7. $$\mathbf{v = \frac{\pi dn}{60000} = \frac{\pi\,x\,40\,x\,600}{60000} = 1.26\,m/s}$$

8. H_g = F f v = 2500 x0.00305 x 1.26 = 9.61 Nm /s

9. $H_d = CAB\,(T_o\text{-}T_A) = H_g$ from which

C = 33.5 W/m^2.^{o}C for moving air

B = 0.667 from Table 2.1 for oil bath in moving air.

A = 20 d l = 20 x 0.04 x 0.02 = 0.016 m^2

10. $T_o = T_A + H_g / CAB$

= 35 + 9.61 / (33.5x0.016 x0.667)

= 61.9^oC

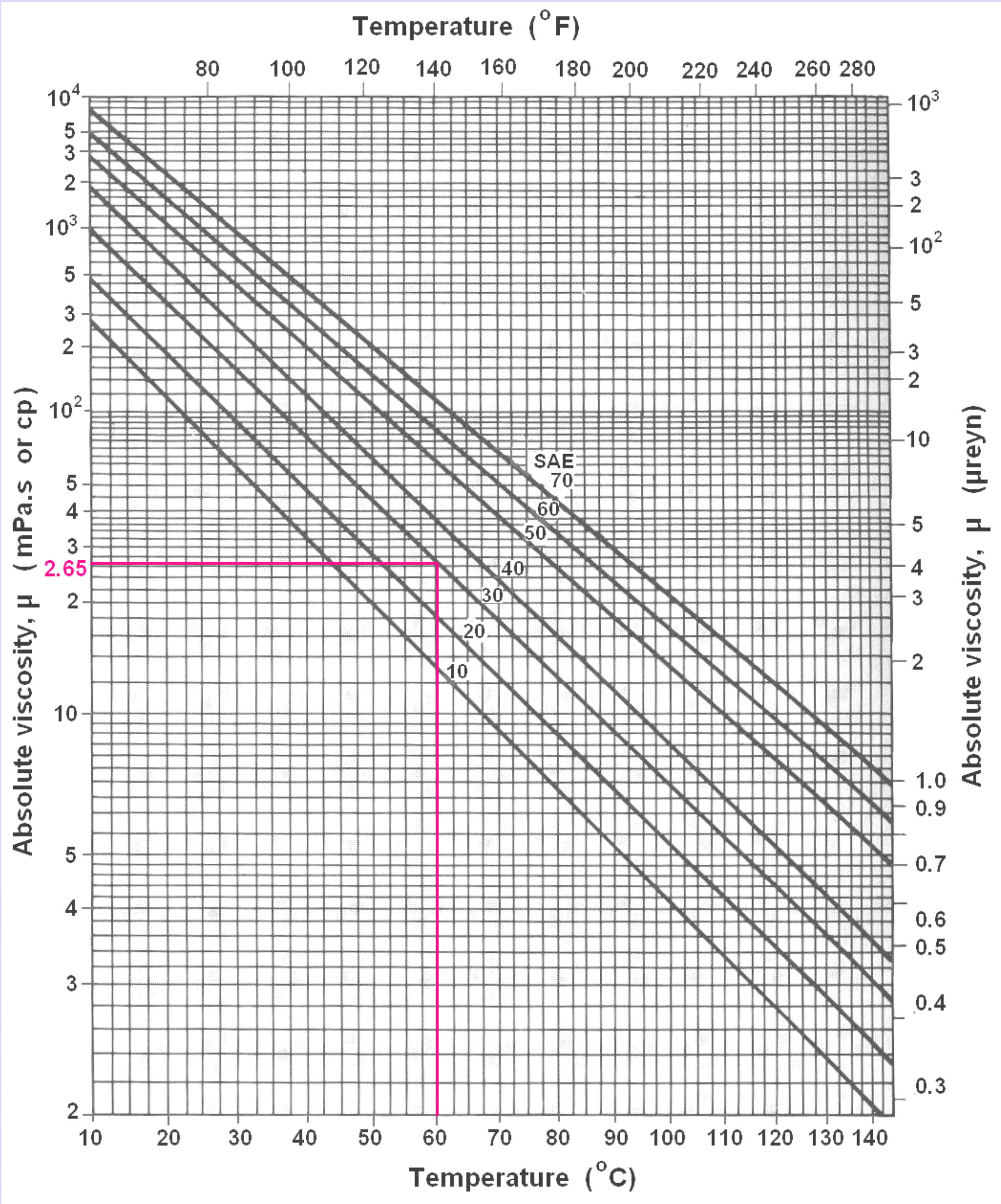

Fig.2.3c Viscosity – temperature curves of SAE graded oils

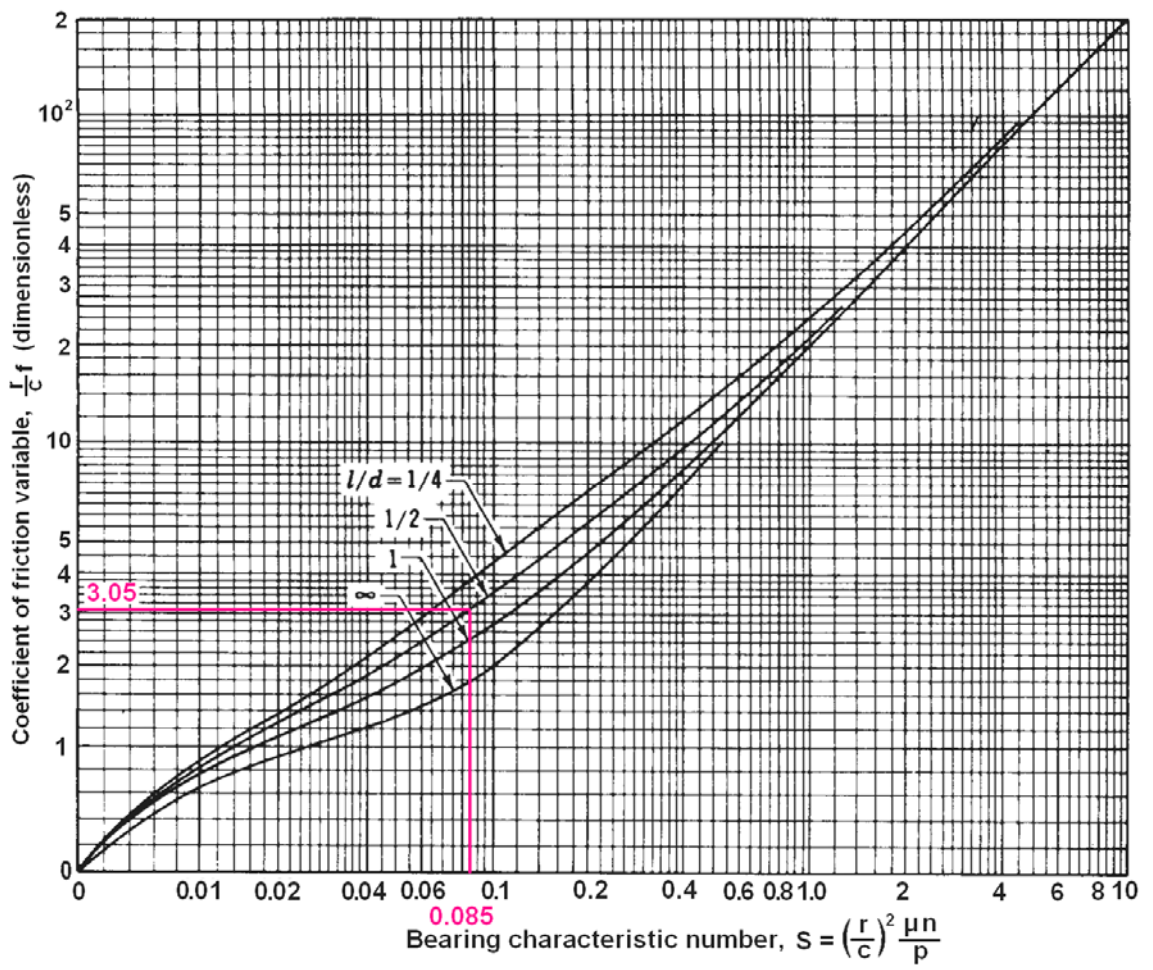

Fig. 2.11b Chart for coefficient of friction variable

Iteration 2

1. For oil temperature of 61.9°C, μ = 26.5 mPa.s for SAE 30 oil from Fig.2.3d

2. $$S = \left(\frac{r}{c}\right)^2 \left(\frac{\mu n}{p}\right) = 1000^2 \left(\frac{24.5 x 10^{-3} \; x \; 10}{3.13 \; x \; 10^6}\right) = 0.078$$

3. f = 0.00285

4. H_g = F f v = 2500 x0.00285 x 1.26 = 8.98 Nm /s

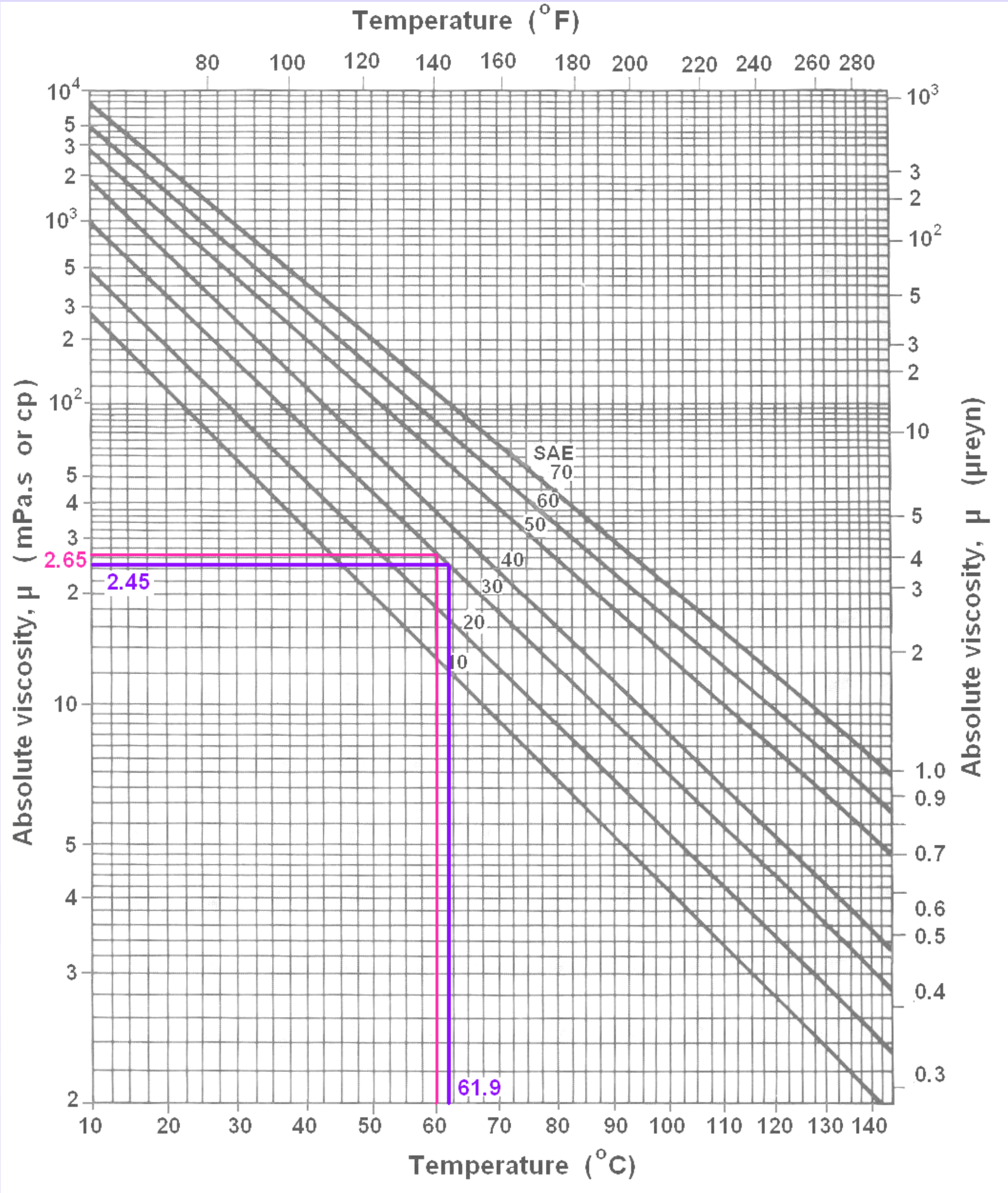

Fig.2.3d Viscosity – temperature curves of SAE graded oils

5. $T_o = T_A + H_g / CAB$

$= 35 + 8.98 / (33.5x0.016 x0.667)$

$= 60.1^oC$

6. Hence the equilibrium temperature of oil will be around 60.1°C.

End of problem 2

3.5 HYDRODYNAMIC LUBRICATED JOURNAL BEARING DESIGN

The design procedure of hydrodynamic bearing is very elaborated one with theory and practice being judiciously blended together. The following guidelines aid in design:

3.5.1 Unit loading

The load per unit journal projected area is denoted by **p**. In many applications like engine bearings, momentary peak loads result in bearing pressures of the order of ten times the steady state values. The hydrodynamic bearings can take up such peak loads without any problem. The recommended values of steady unit load for various applications are given in Table 3.1. This helps in selecting suitable diameter for any particular the application.

3.5.2 Bearing l / d ratios

Ratios - 0.25 to 0.75 are now commonly used in modern machinery whereas in older machinery closer to unity was used. Longer bearings have less end leakage and reduced oil flow requirements and high oil temperature. Short bearings are less prone to edge loading from shaft deflection and misalignments, need higher flow rate and run cooler. The shaft size is found from fatigue strength and rigidity considerations. Bearing length is found from permissible unit loads.

Table 3.2 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

(b) Rapidly fluctuating loads p = Fmax / d l

Applications	Unit loads MPa	Applications	Unit loads MPa
Diesel Engines		Automotive gasoline engines	
Main bearings	6 – 12	Main bearings	4 - 5
Connecting rod bearings	8 – 15	Connecting rod bearings	10 – 15

3.5.3. Acceptable values of h_o:

The minimum acceptable oil film thickness, h_o, depends on surface finish. Trumpler suggests the relationship

$$h_o \geq 0.005 + 0.00004\, d \quad \text{(units in mm)} \quad (3.7)$$

This equation applies only to bearings that have finely ground journal with surface roughness not exceeding 5μm, that have good standards of geometric accuracy circumferential out of roundness, axial taper, and "waviness" both circumferential and axial; and that have good standards of oil cleanliness.
A factor of safety of 2 is suggested for steady loads that can be assessed with good accuracy.

3.5.4 Clearance ratios c/r

For journals 25 to 150 mm in diameter and for precision bearings (c / r) ratio of the order 0.001 is recommended.

For less precise bearings of general machinery bearings (c / r) ratio up to about 0.002 is used.

For rough-service machinery (c / r) ratio of 0.004 is used.

In any specific design the clearance ratio has a range of values, depending on the tolerances assigned to the journal and bearing diameter.

Table 3.3 Clearance ratio: $\psi = c/r$ in 10^{-3}

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

Table 3.4.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

3.5.5 Important factors to be taken into account for designing a hydrodynamic bearing

1. The minimum oil film thickness to ensure thick film lubrication is given as

$h_o \geq 0.005 + 0.00004\, d$

2. Friction should be as low as possible to reduce the power loss ensuring adequate oil film thickness. Operation in the optimum zone in Raimondi chart ensures good design.

3. Ensure adequate supply of clean and cool oil at the bearing inlet.

4. Ensure that the oil temperature never exceeds 93^{o}C for long life of the oil.

5. ***Grooves*** are to be provided for distribution of oil admitted to the bearing over its full length. If so, they should be kept away from highly loaded areas.

6. Choose a bearing material with enough strength at operating temperatures, adequate conformability and embeddability, and sufficient corrosion resistance.

7. Shaft misalignment and deflection should not be excessive.

8. Check the bearing loads and elapsed times during start-up and shutdown. Bearing pressures should be below 2MPa during these periods.

9. To arrive at a good design, right combinations of clearance and oil viscosity for given operating condition should be chosen. This will ensure running of the bearing with minimum friction and wear, and lowest possible temperature by dissipating the heat.
