

Module 5 – SLIDING CONTACT BEARINGS

Lecture 1- SLIDING CONTACT BEARINGS: INTRODUCTION

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1.1 SLIDING CONTACT BEARINGS - INTRODUCTION

Bearings are machine elements which are used to support a rotating member viz., a shaft. They transmit the load from a rotating member to a stationary member known as frame or housing.

They permit relative motion of two members in one or two directions with minimum friction, and also prevent the motion in the direction of the applied load.

The bearings are classified broadly into two categories based on the type of contact they have between the rotating and the stationary member

- a. Sliding contact
- b. Rolling contact

The sliding contact bearings having surface contact and are coming under lower kinematic pair.

1.2 SLIDING CONTACT BEARINGS - ADVANTAGES AND DISADVANTAGES

These bearings have certain advantages over the rolling contact bearings. They are:

1. The design of the bearing and housing is simple.
2. They occupy less radial space and are more compact.
3. They cost less.
4. The design of shaft is simple.
5. They operate more silently.
6. They have good shock load capacity.
7. They are ideally suited for medium and high speed operation.

The disadvantages are:

1. The frictional power loss is more.
2. They required good attention to lubrication.
3. They are normally designed to carry radial load or axial load only.

1.3 SLIDING CONTACT BEARINGS - CLASSIFICATION

Sliding contact bearings are classified in three ways.

1. Based on type of load carried
2. Based on type of lubrication
3. Based on lubrication mechanism

1. 3.1 Bearing classification based on type of load carried

- a. Radial bearings
- b. Thrust bearings or axial bearings
- c. Radial – thrust bearings

1.3.1(a) Radial bearings

These bearings carry only radial loads.

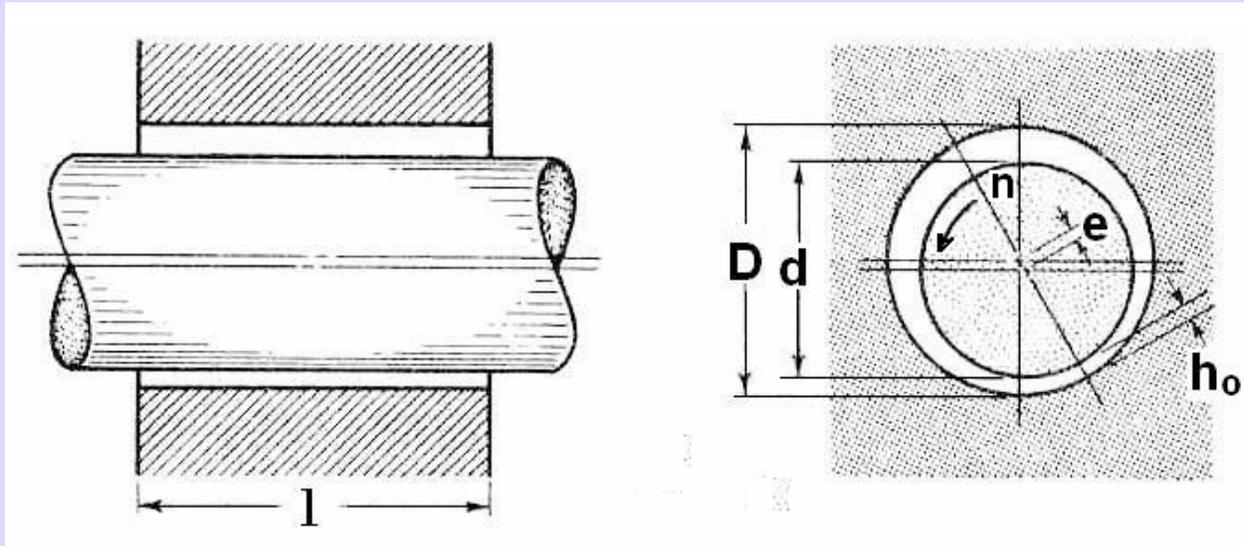


Fig.1.1 Radial Bearing

1.3.1(b) Thrust or axial bearings

These bearings carry only axial loads

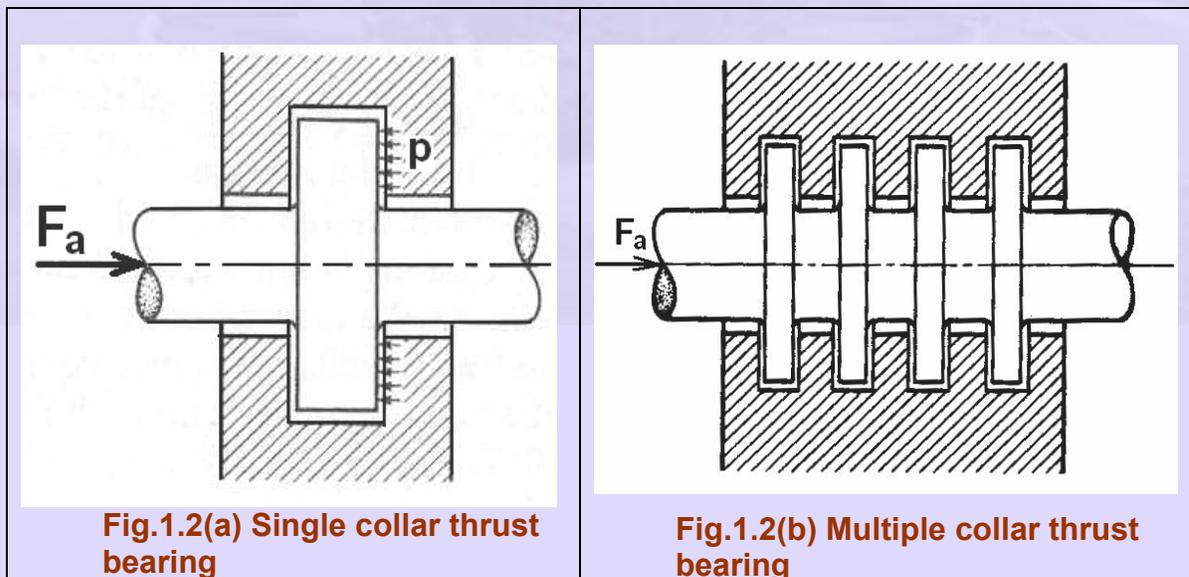


Fig.1.2(a) Single collar thrust bearing

Fig.1.2(b) Multiple collar thrust bearing

1.3.1(c) Radial thrust bearings

These bearings carry both radial and thrust loads.

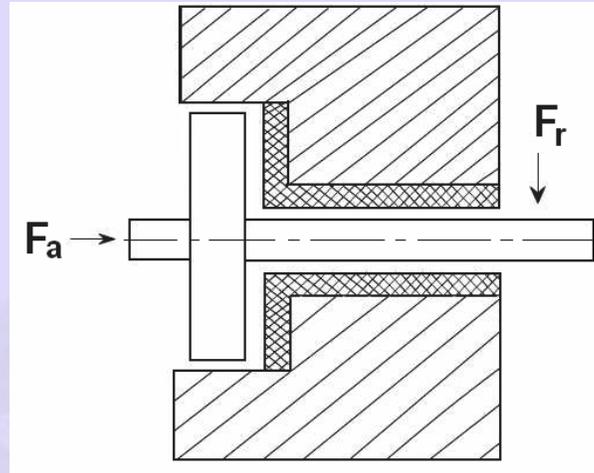


Fig.1.2 (c) Radial thrust bearing

1.3. 2. Bearing classification based on type of lubrication

The type of lubrication means the extent to which the contacting surfaces are separated in a shaft bearing combination. This classification includes

- (a) Thick film lubrication
- (b) Thin film lubrication
- (c) Boundary lubrication

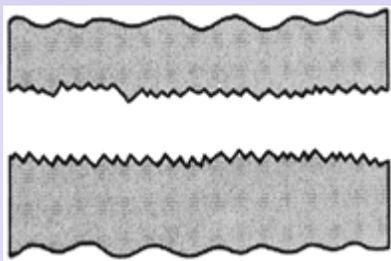


Fig1.4(a) Thick film lubrication

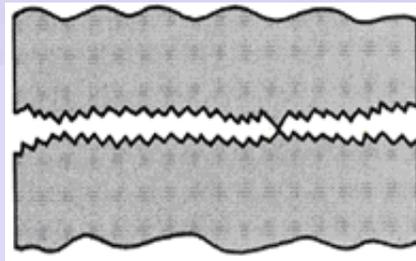


Fig.1.4(b) Thin film lubrication

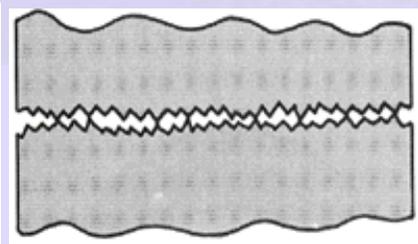


Fig.1.4(c) Boundary lubrication

1.3. 2(a) Thick film lubrication – As in Fig.1.4 (a) the surfaces are separated by thick film of lubricant and there will not be any metal-to-metal contact. The film thickness is anywhere from 8 to 20 μm . Typical values of coefficient of friction are 0.002 to 0.010. Hydrodynamic lubrication is coming under this category. Wear is the minimum in this case.

1.3.2(b) Thin film lubrication – Here even though the surfaces are separated by thin film of lubricant, at some high spots Metal-to-metal contact does exist , Fig.1.4 (b).Because of this intermittent contacts, it also known as mixed film lubrication. Surface wear is mild. The coefficient of friction commonly ranges from 0.004 to 0.10.

1.3.2(c).Boundary lubrication – Here the surface contact is continuous and extensive as Shown in Fig.1.4(c). The lubricant is continuously smeared over the surfaces and provides a continuously renewed adsorbed surface film which reduces the friction and wear. The typical coefficient of friction is 0.05 to 0.20.

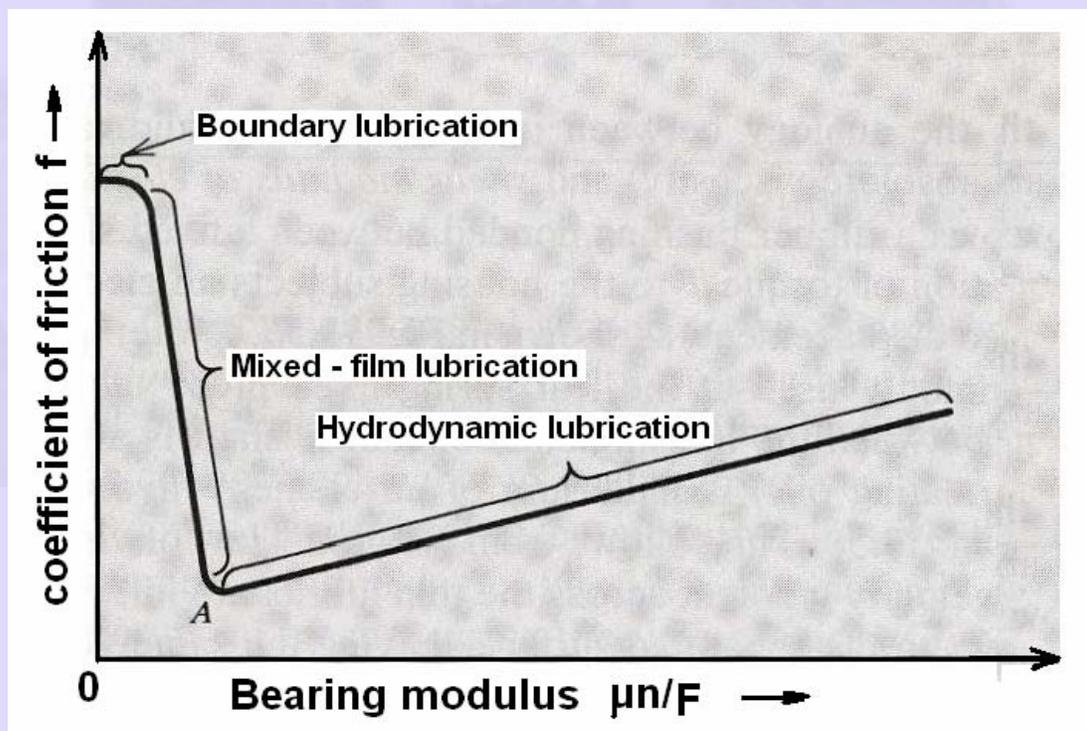


Fig. 1.5 Stribeck curve for bearing friction

1.3.3 Bearing classification based on lubrication mechanism

- Hydrodynamic lubricated bearings
- Hydrostatic lubricated bearings
- Elastohydrodynamic lubricated bearings
- Boundary lubricated bearings
- Solid film lubricated bearings

The operating regimes of different lubrication mechanisms are depicted by Stribeck in Fig.1.5 by plotting coefficient of friction versus the non-dimensional factor known as bearing modulus.

1.3.3(a) Hydrodynamic lubricated bearings

In these bearings the load-carrying surfaces are separated by a stable thick film of lubricant that prevents the metal-to-metal contact. The film pressure generated by the moving surfaces that force the lubricant through a wedge shaped zone. At sufficiently high speed the pressure developed around the journal sustains the load. This is illustrated in Fig.1.6.

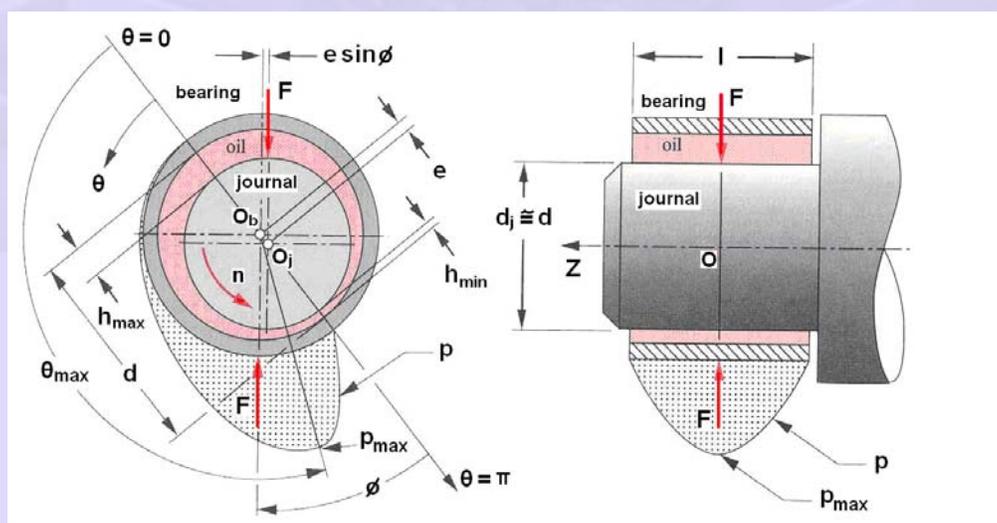


Fig.1.6 Hydrodynamic lubricated bearing

1.3.3(b) Hydrostatic lubricated bearings

In these bearings, externally pressurized lubricant is fed into the bearings to separate the surfaces with thick film of lubricant. These types of bearings do not require the motion of the surfaces to generate the lubricant film. Hence they can operate from very low speed to high speed. This is illustrated in Fig. 1.7

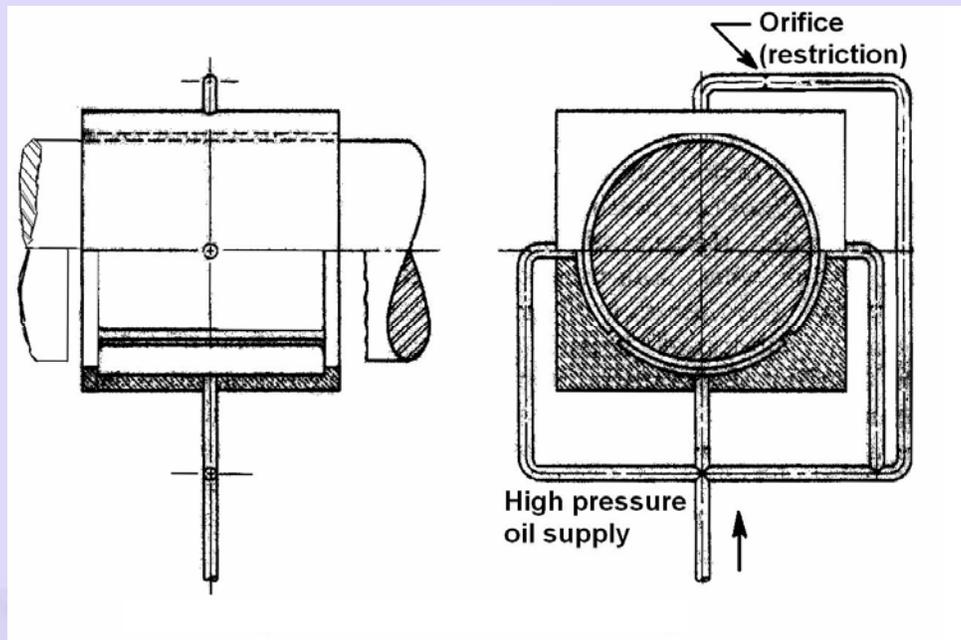
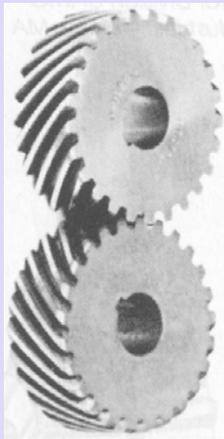


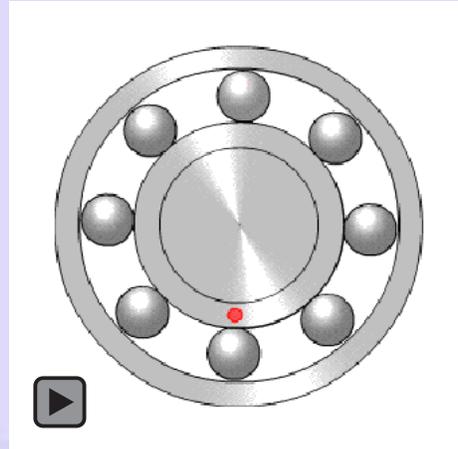
Fig.1.7 Hydrostatic lubricated bearing

1.3.3(c) Elastohydrodynamic lubricated bearings

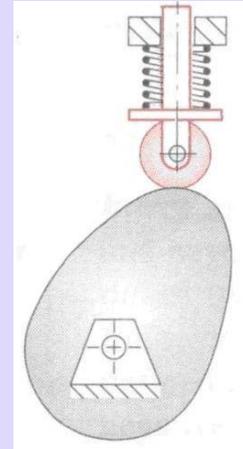
Rolling contact bearings come under this category. The oil film thickness is very small. The contact pressures are going to be very high. Hence to prevent the metal-to-metal contact, surface finishes are to be of high quality. Such a type of lubrication can be seen in gears, rolling contact bearings, cams etc.



(a)



(b)



(c)

Fig.1.8 (a) Gear, (b) Rolling contact bearing and (c) Cam

1.3.3(d) Boundary lubricated bearings

When the speed of the bearing is inadequate, less quantity of lubricant is delivered to the bearing, an increase in the bearing load, or an increase in the lubricant temperature resulting in drop in viscosity – any one of these may prevent the formation of thick film lubrication and establish continuous metal-to-metal contact extensively. Often bearings operating in such situations are called boundary lubricated bearings.

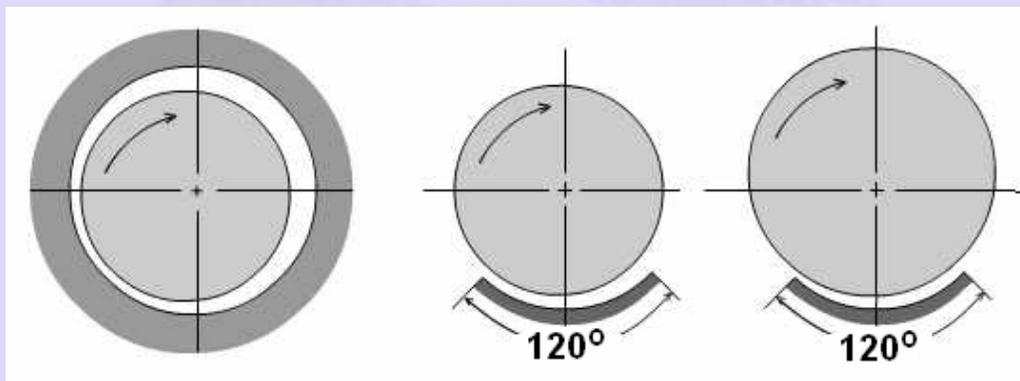
1.3.3(e) Solid film lubricated bearings

For extreme temperature operations ordinary mineral oils are not satisfactory. Solid film lubricants such as graphite, molybdenum disulfide or their combinations which withstand high operating temperature are used. These types of bearings are common in furnace applications, or trunnion bearings of liquid metal handling systems, hot drawing mills etc.

1.4 JOURNAL / SLEEVE BEARINGS

Among the sliding contact bearings radial bearings find wide applications in industries and hence these bearings are dealt in more detail here.

The radial bearings are also called journal or sleeve bearings. The portion of the shaft inside the bearing is called the journal and this portion needs better finish and specific property. Depending on the extent to which the bearing envelops the journal, these bearings are classified as full, partial and fitted bearings. As shown in Fig.1.9.



(a) Full

(b) Partial

(c) Fitted

Fig. 1.9 Various types of journal bearings

1.5 Hydrodynamic lubrication

In 1883 Beauchamp Tower discovered that when a bearing is supplied with adequate oil, a pressure is developed in the clearance space when the journal rotates about an axis that is eccentric with the bearing axis. He exhibited that the load can be sustained by this fluid pressure without any contact between the two members.

The load carrying ability of a hydrodynamic bearing arises simply because a viscous fluid resists being pushed around. Under proper conditions, this resistance to motion will

develop a pressure distribution in the film that can support useful load. Two mechanisms responsible for this are wedge film and squeeze film action.

The load supporting pressure in hydrodynamic bearings arises from either (1) the flow of a viscous fluid in a converging channel, the wedge film, or (2) the resistance of a viscous fluid to being squeezed out from the between approaching surface, the squeeze film.

1.5.1 Stages in hydrodynamic lubrication

Consider a steady load F , a fixed bearing and a rotating journal.

Stage 1 –:

At rest, the bearing clearance space is filled with oil, but the load F has squeezed out the oil film at the bottom. Metal-to-contact exists. The vertical axis of bearing and journal are co-axial. Load and reaction are in line fig.1.10.

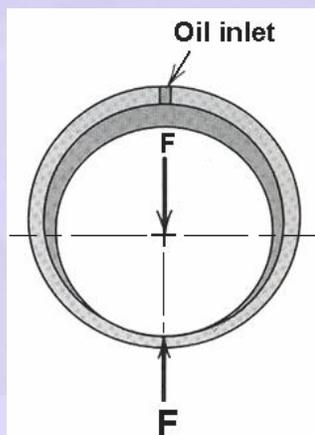
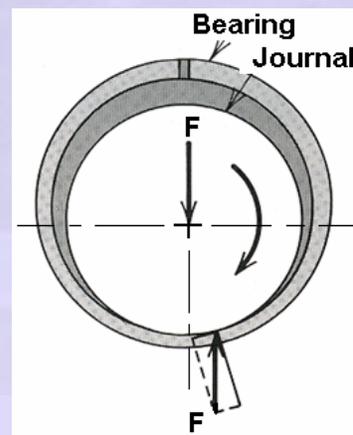


Fig.1.10 At rest



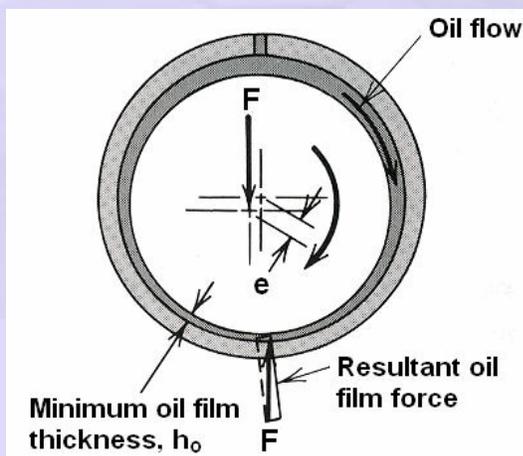
**Fig.1.11 Slow rotation
Boundary lubrication**

Stage 2:

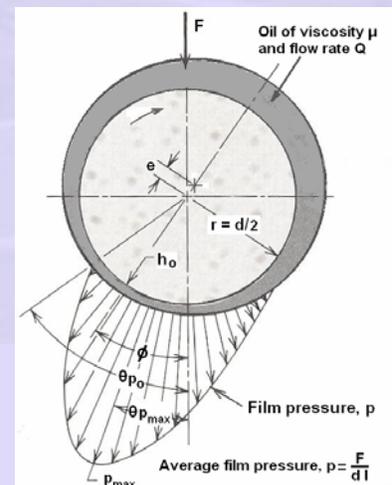
When the journal starts rotating slowly in clockwise direction, because of friction, the journal starts to climb the wall of the bearing surface as in Fig.1.11. Boundary lubrication exists now. The wear normally takes place during this period. However, the journal rotation draws the oil between the surfaces and they separate.

Stage 3:

As the speed increases, more oil is drawn in and enough pressure is built up in the contact zone to float the journal Fig.1.12. Further increase in speed, additional pressure of the converging oil flow to the right of the minimum film thickness position (h_o) moves the shaft slightly to the left of center. As a result full separation of journal and bearing surfaces occurs. In stable operating condition, the pressure distribution on the journal is shown in Fig.1.13. This is known as – Hydrodynamic lubrication or full film/thick film lubrication. At this equilibrium condition, the pressure force on journal balances the external load F . The animation of this lubrication is shown in Fig.1.14.



**Fig.1.12 At running
(hydrodynamic lubrication)**



**Fig.1.13 Stable hydrodynamic
lubrication**

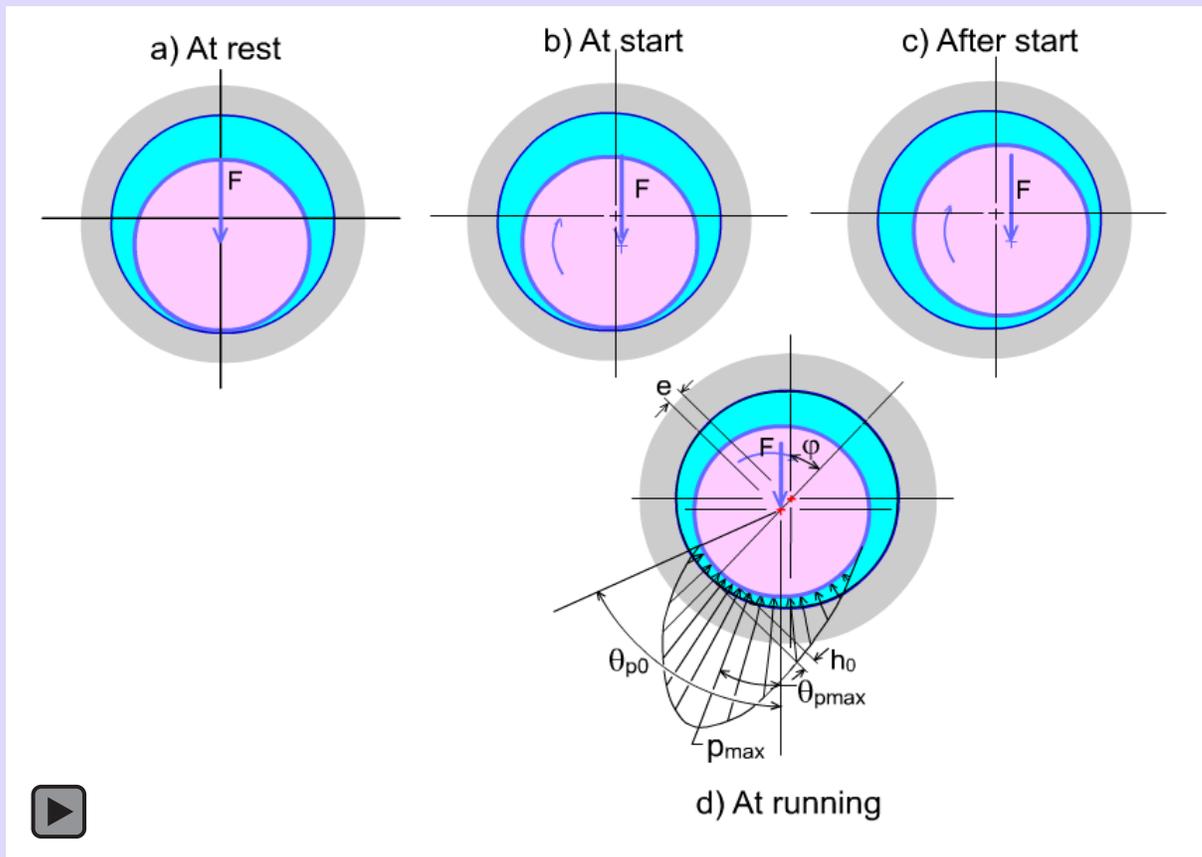


Fig.1.14 Journal position in hydrodynamic lubrication

1.5 .2 HYDRODYNAMIC LUBRICATION - ANIMATION

The various stages of lubrication explained in 1.5.1 can now be perceived from the animation illustrated here.

1.5.3 The friction characteristics of hydrodynamic lubrication of journal bearings

The friction behaviour during hydrodynamic condition is shown in Fig.1.15 and the bearing will operate between point C and D under hydrodynamic lubrication condition.

It can be seen from the graph Fig.1.15 and bearing modulus (μ_n / p) that

- The higher the viscosity, the lower the rotating speed needed to “float” the journal at a given load.

- Any further increase in viscosity produces more bearing friction thereby increasing the forces needed to shear the oil film.
- The higher the rotating speed, the lower the viscosity needed to “float” the journal at a given load. ($\mu n / p$)
- Further increases in rotating speed produces more bearing friction by increasing the time rate at which work is done in shearing the oil film.

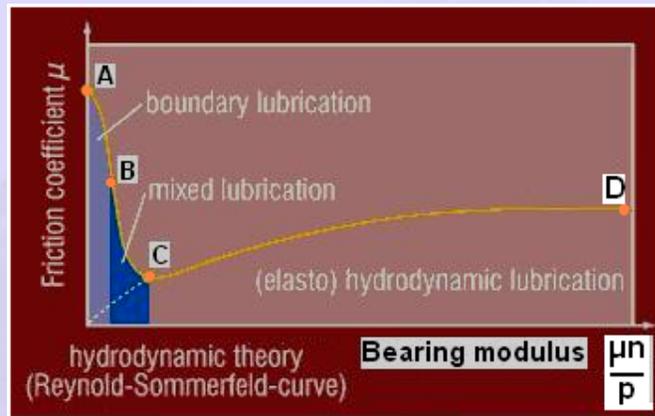


Fig.1.15. Friction behaviour during hydrodynamic lubrication

Average pressure on project surface of the journal: $p = F / (ld)$ (1.1)

Where F – Radial load

l – Length of the journal

d – Journal diameter

ld - Bearing projected area

- The smaller the bearing unit load, the lower the rotating speed and the viscosity needed to “float” the journal.
- Further reductions in bearing load do not produce corresponding reductions in the bearing friction drag force.
- Thus the bearing coefficient of friction, which is the ratio of friction drag force to radial load F, increases.

The basic requirements for achieving Hydrodynamic lubrication are :

1. Surfaces which are in relative motion to be separated.
2. "Wedging," as provided by the shaft eccentricity.
3. The presence of a suitable fluid.

1.6 JOURNAL BEARING - LUBRICANTS

1.6.1 Viscosity:

It is the internal friction that resists the motion in fluids. If an unloaded plate of area $A \text{ m}^2$ moves parallel to a stationary surface with velocity $U \text{ m/s}$ as in Fig.1.16 and the space is filled with fluid, the velocity gradient will be a straight line. The fluid shear stress for Newtonian fluids is proportional to the rate of shear, i.e.,

$$\tau = \mu \frac{U}{h} \quad (1.2)$$

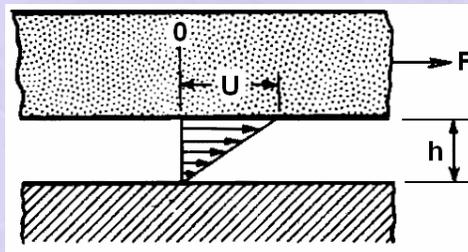


Fig.1.16 Flow between parallel plates

Force required to move the plate is given by

$$F = \mu \frac{U}{h} A \quad (1.3)$$

The unit of viscosity in SI units is Ns/m^2 or Pa.s . Since this is a large unit, it is normally expressed as millipascal second mPa.s or centipoise cp .

One poise is the force, in dynes, required to move one face of a 1cm^3 of liquid at the rate of 1 cm/s relative opposite face. Since this unit is very large one hundredth of it is taken normally and expressed as centipoise or cp.

In FPS unit viscosity is expressed as reyns. $1\text{ reyn} = 6890\text{ Pa.s}$.

Viscosity is also expressed as

SUS (Saybolt Universal Seconds),

SSU (Saybolt Seconds Universal),

SUV (Saybolt Universal Viscosity).

Kinematic viscosity = (absolute viscosity)/(mass density)

Units are $\text{length}^2/\text{time}$, as cm^2/sec , which is named stoke, abbreviated as St.

1.6.2 Viscosity Measurement

Saybolt universal viscometer is widely used for the measurement of viscosity, Fig.1.17.

The time required for a given quantity of the liquid to flow by gravity through a precision opening. Absolute viscosity expressed in saybolt seconds s.

$$\mu \text{ (in cp)} = (0.22 \text{ s} - 180/\text{s}) \rho \quad (1.4)$$

$$\mu \text{ (in } \mu\text{reyn)} = 0.145(22\text{s}-180/\text{s})\rho \quad (1.5)$$

where ρ is the density of oil .

ρ is the mass density of oil in g / cm^3 or numerically equal to specific gravity.. For petroleum oils the density at 60°F or 15.6°C is 0.89 g/cm^3 . At other temperatures, T_o in $^\circ\text{C}$ or T'_o in $^\circ\text{F}$, the density is given by equation 1.6 and 1.7.

$$\rho = 0.89 - 0.00063 (T_o - 15.6) \quad (1.6) \text{ or}$$

$$\rho = 0.89 - 0.00035 (T'_o - 60) \quad (1.7)$$

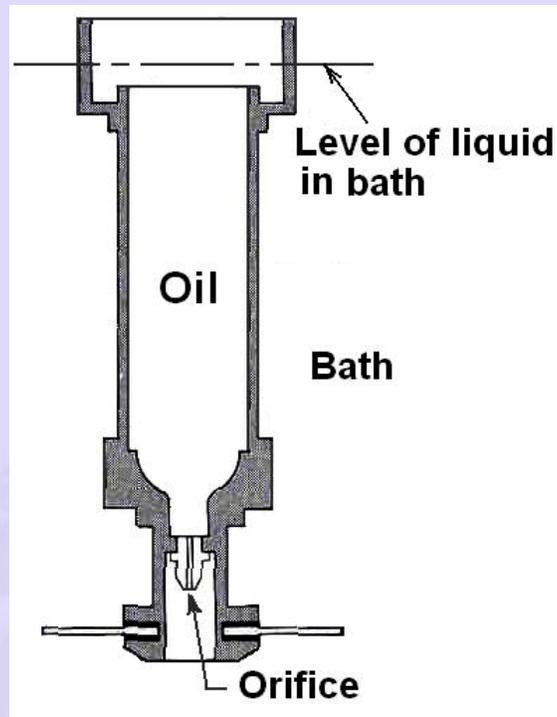


Fig.1.17 Saybolt viscometer

1.6.3 Viscosity Index

- The measure of variation of viscosity with temperature is the *viscosity index* (VI).
- For Pennsylvania crude oils, VI = 100, which undergoes the least change of viscosity with temperature.
- For Gulf coast oils, VI = 0, which undergoes the greatest change with temperature
- Other oils were rated intermediately. VI of Multigrade oils such as SAE 10W-40 is more than that of single grade designation (as SAE 40 or SAE 10W).

$$VI = \frac{L - U}{L - H} \times 100 \quad (6)$$

Where L- viscosity of a standard 0% VI oil at 100°F

H- Viscosity of standard 100% VI oil at 100°F

U - Viscosity of oil with unknown VI oil at 100°F

1.6.4 Temperature Effects on Viscosity Index

- a. Nonpetroleum-base lubricants have widely varying viscosity indices. Silicone oils, for example, have relatively little variation of viscosity with temperature. Thus their viscosity indices substantially exceed 100 on the Dean and Davis scale.
- b. The viscosity index of petroleum oils can be increased by the use of viscosity index improvers known as additives.

1.6.5 Pressure Effects on Viscosity

- a. All lubricating oils experience an increase in viscosity with pressure. This effect is usually significant only at pressures higher than those encountered in sliding bearings.
- b. This effect is important in elastohydrodynamic lubrication.

1.6.5 LUBRICANT PROPERTIES

Properties of a good lubricant are:

1. It should give rise to low friction.
2. It should adhere to the surface and reduce the wear.
3. It should protect the system from corrosion.
4. It should have good cleaning effect on the surface.
5. It should carry away as much heat from the surface as possible.
6. It should have thermal and oxidative stability.
7. It should have good thermal durability.
8. It should have antifoaming ability.
9. It should be compatible with seal materials.
10. It should be cheap and available in plenty.

1.7 LUBRICANT FOR JOURNAL BEARING APPLICATION

1.7.1 Recommended Lubricants for the Bearing Application

1. SAE 10 – spindle oil for light loaded bearings and high speeds.
2. SAE 20 – 40 – Machine oil for bearings of IC engines, machine tools, turbines etc.
3. SAE40-50 – Machine oil for diesel engines heavy load and medium speeds.
4. SAE 60-70 – machine oil for high temperature, heavy load and low speeds.

1.7.3 SAE Specification of Lubrication oils

- a. Viscosity of SAE 30 oil lies in between “thickest” SAE 20 and “thinnest” SAE 40 oil being the thickest.
- b. SAE 20,30,40 and 50 are specified at 100°C.
- c. SAE 5W, 10W and 20W are specified at -18°C.
- d. SAE 10W-40 oil must satisfy the 10W viscosity requirement at -18°C and the 40 requirement at 100°C.

1.7.4 ISO Specification of Lubrication oils

Industrial fluid lubricants are commonly specified in terms of international standards, which appear as

1. ASTM D 2422,
2. American National Standard Z11.232,
3. ISO Standard 3448.

The various viscosity grades are designated as “ISO VG” followed by a number equal to the nominal kinematic viscosity at 40°C.

Eighteen grades are specified, with kinematic viscosities at 40°C of, 3,5,7,10,15,22,46,68,100,150, 220,320,460, 680, 1000 and 1500 cSt (mm²/s).

The properties of various grades of oil against operating temperatures are given Figs. 1.18 to 1.20

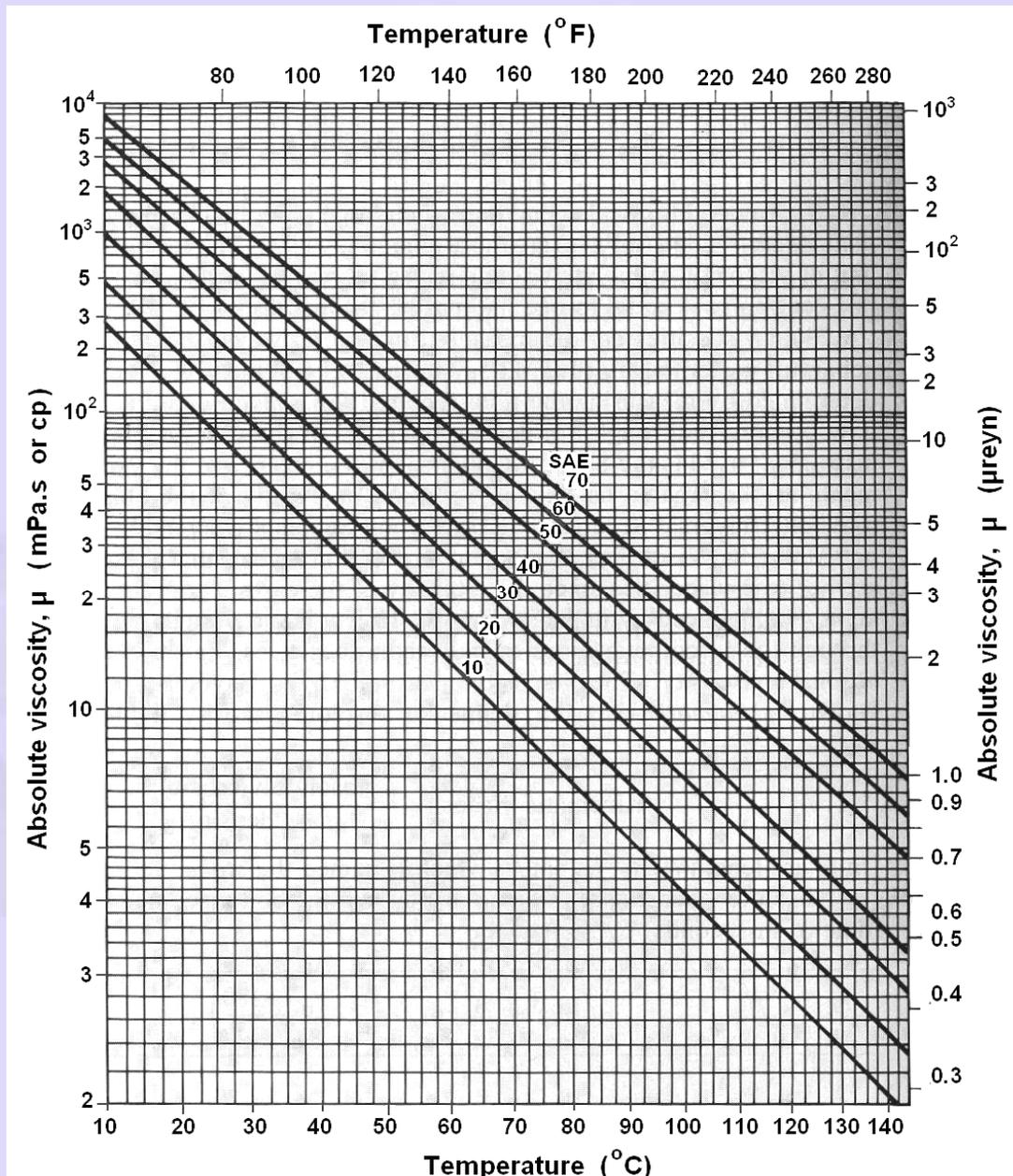


Fig. 1.18 Viscosity - temperature curves of SAE graded oils

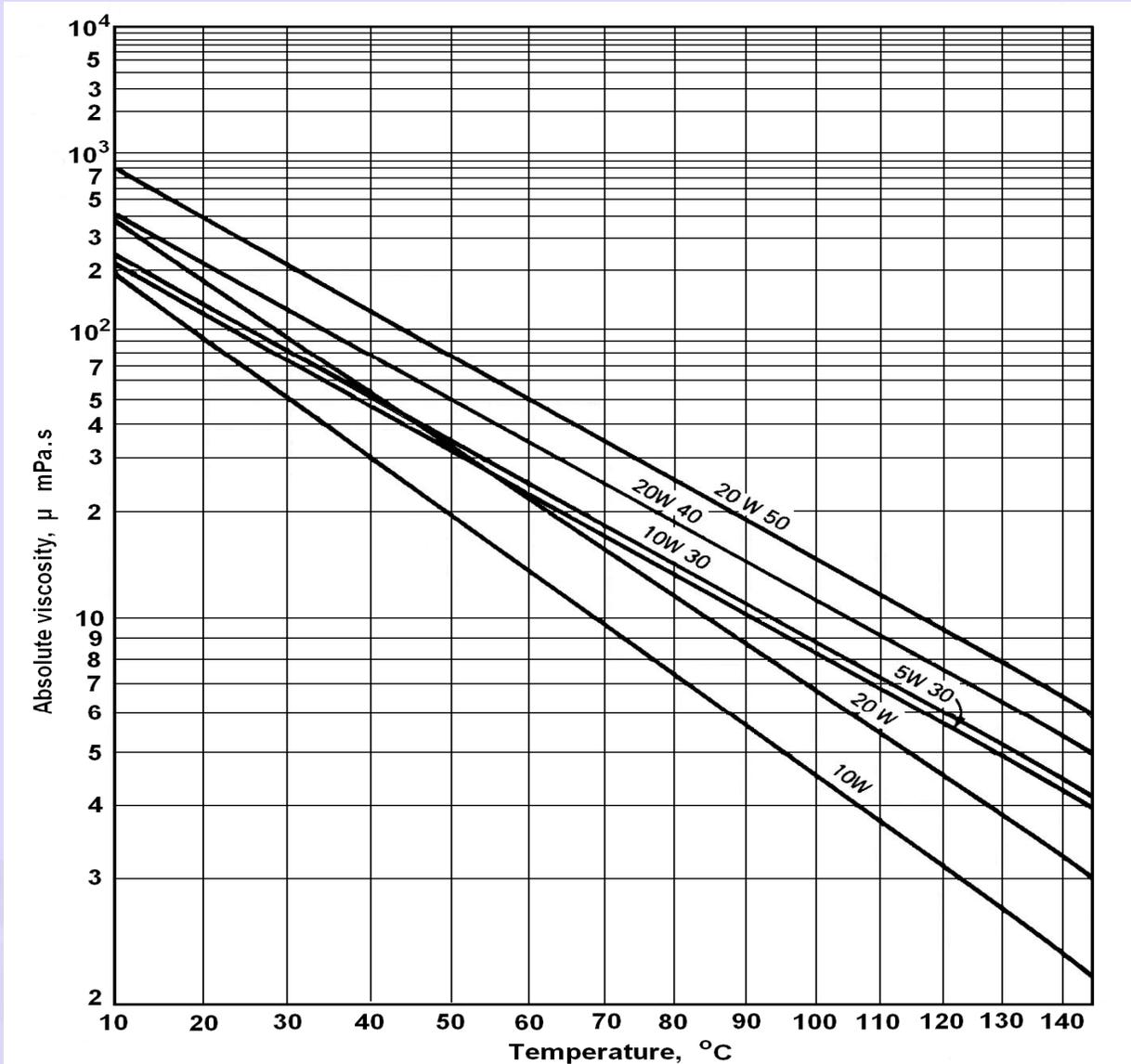


Fig. 1.19 Viscosity temperature chart for multiviscosity lubricants derived from known viscosities at two points, 40 and 100 $^{\circ}\text{C}$

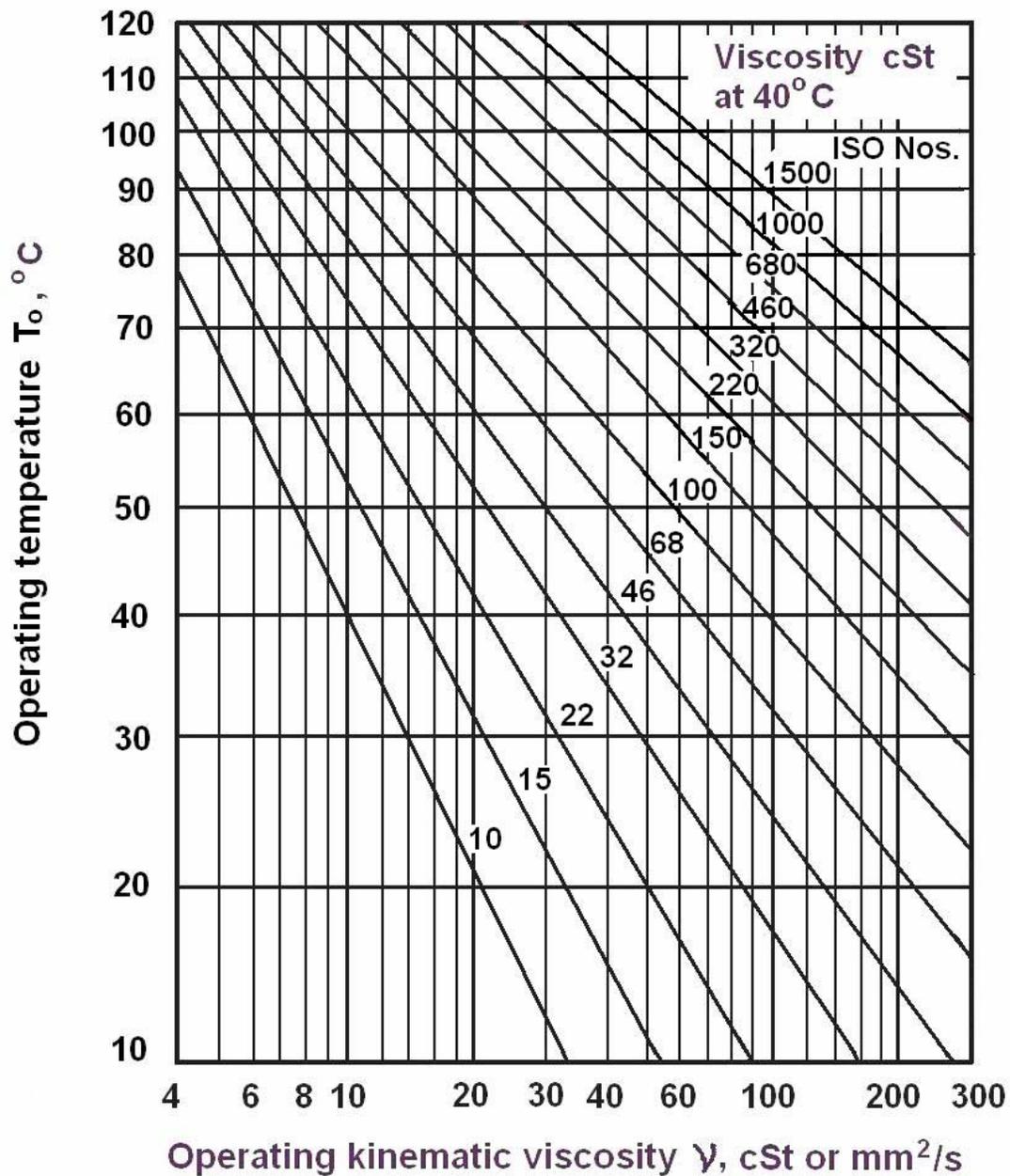


Fig. 1.20 Viscosity – temperature diagram for ISO VG graded oils

1.8 JOURNAL BEARING LUBRICANTS - PROBLEM 1

In a journal bearing application an oil of kinematic viscosity at 100°C corresponding to 46 seconds as found from Saybolt viscometer is used. Determine its absolute viscosity and corresponding oil in SAE and ISO VG grades.

Solution:

$$\gamma = 46 \text{ SUS}$$

$$\begin{aligned} \text{From eqn.(1.6) we have } \rho &= 0.89 - 0.00063(T_o - 15.6) \\ &= 0.89 - 0.00063(100 - 15.6) \\ &= 0.837 \text{ g /cm}^3 \end{aligned}$$

$$\begin{aligned} \text{From eqn (1.4), } \mu &= (0.22 \text{ s} - 180/\text{s}) \rho \\ &= (0.22 \times 46 - 180/46) 0.837 \\ &= 5.19 \text{ cp} \quad \text{Or} \end{aligned}$$

$$\begin{aligned} \text{From eqn (1.5), } \mu &= 0.145(22\text{s} - 180/\text{s})\rho \\ &= 0.145(22 \times 46 - 180/46)0.837 \\ &= 0.735 \text{ } \mu\text{reyn} \end{aligned}$$

SAE oil corresponding to viscosity 5.19cp at 100°C from Fig.1.18a is SAE 20.

$$\begin{aligned} \text{At } 100^\circ\text{C, the kinematic viscosity of the oil from is} \\ = \mu / \rho = 5.19/0.837 = 6.2 \text{ cSt.} \end{aligned}$$

Oil corresponding to the kinematic viscosity 6.3 cSt at 100°C from Fig.1.19a is ISO VG 46.

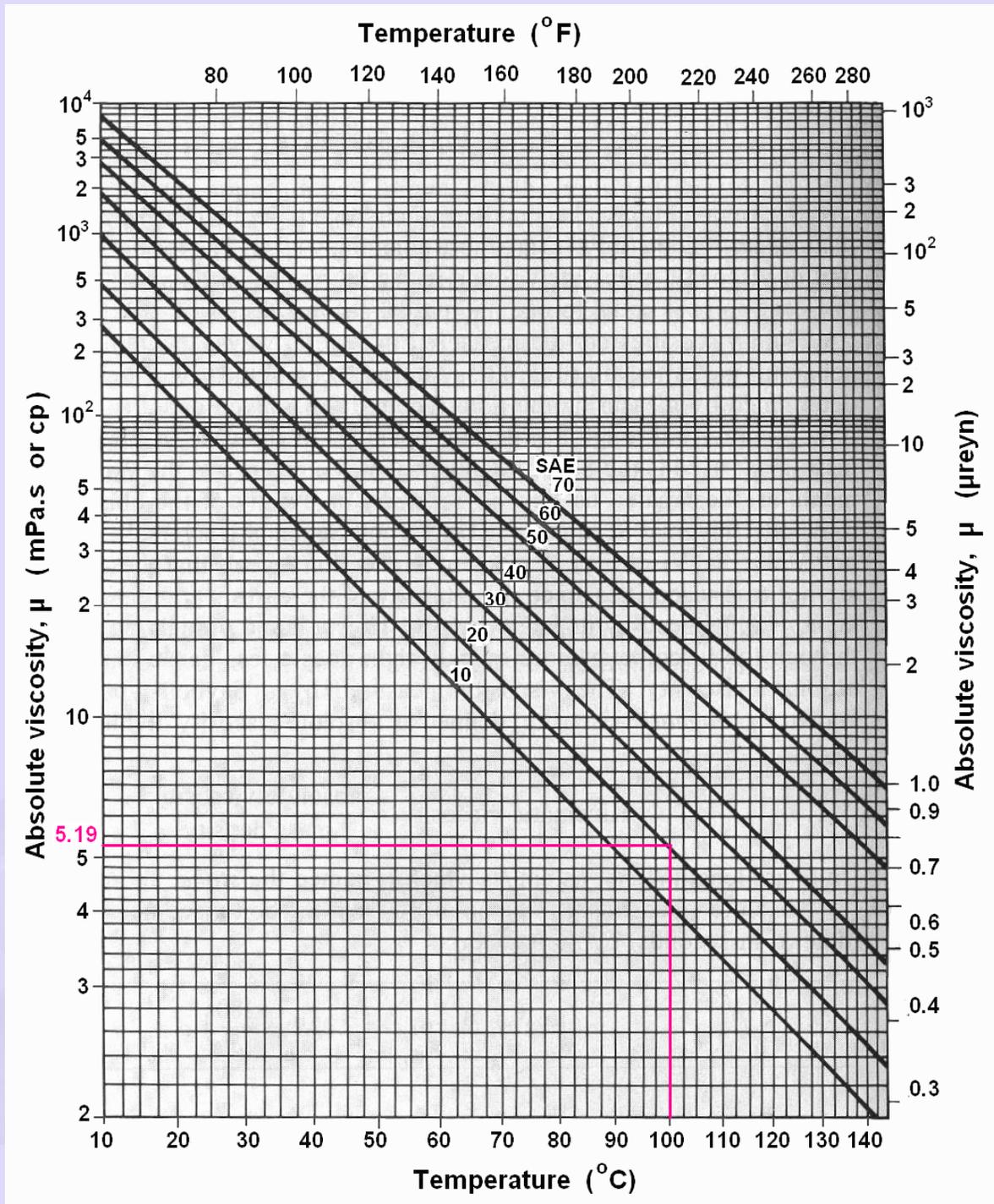


Fig. 1.18a Viscosity – temperature curves SAE graded oils

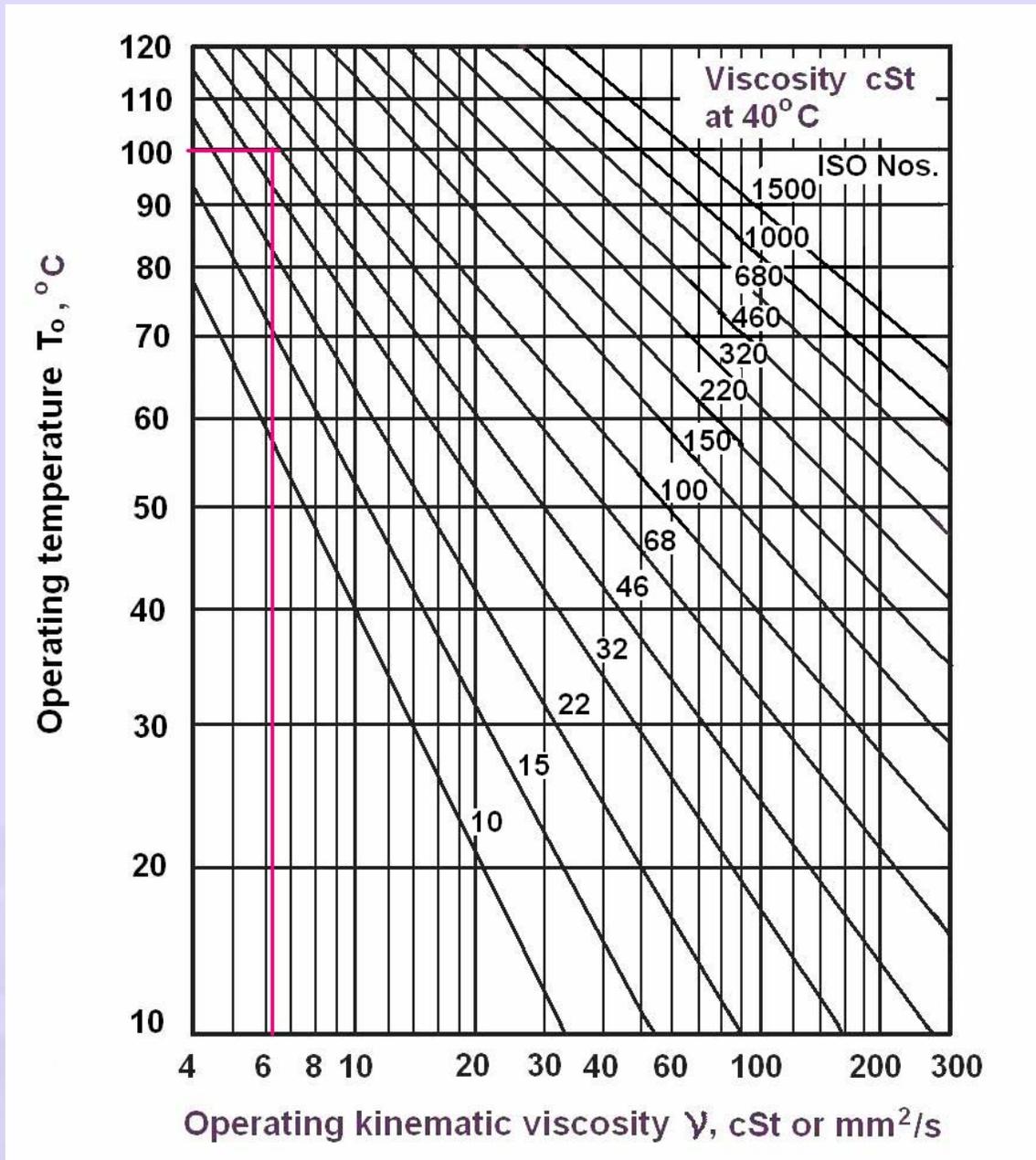


Fig. 1.19a Viscosity – temperature diagram for ISO VG graded oils