ME 308
MACHINE ELEMENTS II

CHAPTER 4

JOURNAL BEARINGS
PART_2

(NO ROLLING ELEMENTS,
ONLY A SHAFT, A HOLE AND SOME LUBRICANT/OIL IN BETWEEN)

The temperature increase ΔT is formulated as:

in American units

$$\Delta T_{^oF} = \frac{0.103P}{\left[1-\frac{1}{2}\times\frac{Q_s}{Q}\right]} \times \frac{\frac{r}{c}f}{\frac{Q}{rcNl}} \qquad P \rightarrow \frac{lb}{in^2}, \quad Q \; and \; Q_s \rightarrow \frac{in^3}{\sec}, \quad r,c,l \rightarrow in \quad \text{etc.}$$

in SI units

$$\Delta T_{^oC} = \frac{8.30P}{\left[1-\frac{1}{2}\times\frac{Q_s}{Q}\right]} \times \frac{\frac{r}{c}f}{\frac{Q}{rcNl}} \quad P \rightarrow \frac{MN}{m^2}, \quad Q \; and \; Q_s \rightarrow \frac{m^3}{\sec}, \quad r,c,l \rightarrow m \quad \text{etc.}$$

- **To be able to determine temperature increase, *ΔT***
- **we need to know parameters:**

$$\frac{r}{c}f, \quad \frac{Q}{rcNl} \quad and \quad \frac{Q_s}{Q}$$

and these parameters are dependent on S and hence on *μ* .

***μ* is also dependent on *ΔT again* !!**

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

or $$\Delta T_{o_C} = T_{\text{var}} \frac{P}{\gamma C_H}$$

where T_{var} is S dependent given in fig. 12.12

To be able to use a correct value of μ in S eqn.

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

we need to know μ which is dependent on ΔT *again*

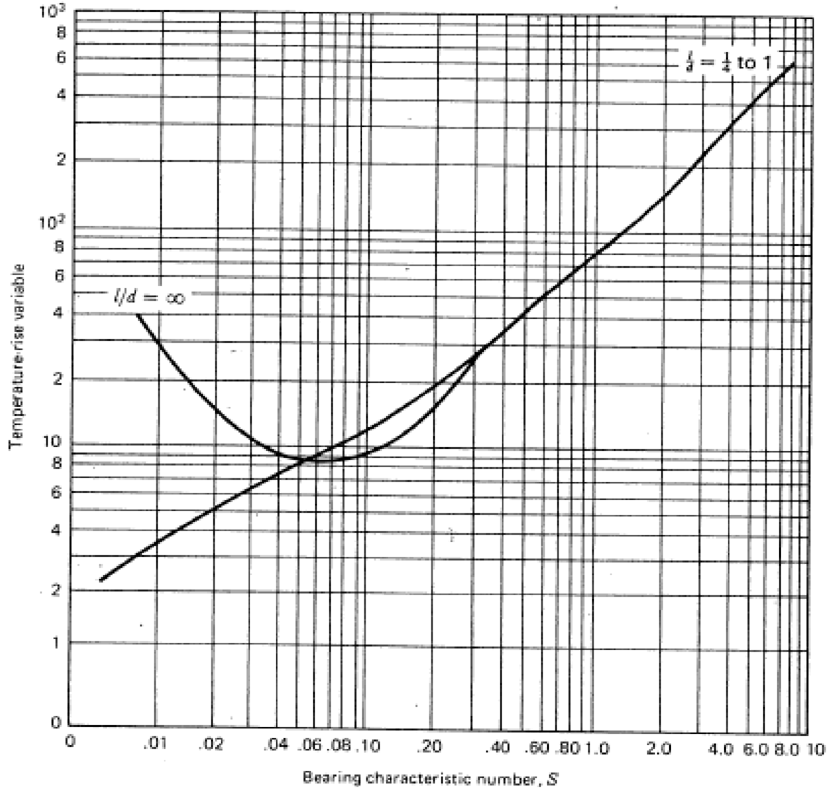


FIGURE 12-12 Chart for temperature-rise variable $T(\text{var}) = \gamma C_H \Delta T/P$. In plotting this chart it was found that the curves for $l/d = \frac{1}{4}, \frac{1}{2}$, and 1 were so close together that they could not be distinguished from a single curve.

Thus there is a situation where we need to start an assumption of either ΔT or μ and check whether the assumption is correct as in the case of tapered roller bearings *(K=1.5)* of TIMKEN.

There are two trial methods:
First one:

1) a) Assume a trial value for the temperature increase (ΔT = *?)*

b) Calculate T_{ave} and determine μ from chart μ vs T_{ave}

c) Calculate S and other related parameters

d) Calculate ΔT_{new} and check if $\Delta T_{new} \approx \Delta T_{old}$; if so OK.

e) If not re-assume ΔT to be between ΔT_{new} & ΔT_{old} and

re-do the calculations from (*b)* to (*d)* until $\Delta T_{new} \approx \Delta T_{old}$

Second trial method is:

2) a) Assume a trial value for the viscosity (μ= *?)* and

b) Then calculate S and related parameters such as $\frac{r}{c} f, \quad \frac{Q_s}{Q}\ etc.$

c) Calculate temperature increase ΔT and average

temperature $T_{ave} = T_{in} + \frac{\Delta T}{2}$

d) Determine μ_{new} from μ vs T_{ave} chart.

e) Check if $\mu_{new} \approx \mu_{old}$; if so design is OK.

f) If not re-assume μ to be between μ_{new} & μ_{old} and

re-do the calculations from (*b)* to (*e)* until $\mu_{new} \approx \mu_{old}$

EXAMPLE 4.1

A sleeve bearing is 10 mm in diameter and 10 mm long. SAE 10 lubricating oil at an inlet temperature of 50^o C is used to lubricate the bearing. The bearing is copper-lead alloy and the journal rotates at 3600 rpm. If the radial load on the bearing is 68 N. Find:

a) temperature rise of the lubricant,
b) the minimum oil-film thickness,
c) the power loss in the bearing.

Given:

l = 10 mm
d= 10 mm
} l/d=1

SAE 10 oil

T_i= 50^o C
T_o= ?
} ΔT= ?

T_{ave}= ?

bearing copper-lead

N= 3600 rev/min= 60 rev/sec

W= 68 N

Required:

ΔT= ? ^{o}C

h_o= ?

P_{loss}= ?

<u>Required</u>:

ΔT = ? °C

h_o = ?

P_{loss} = ?

S dependent

$$\Delta T_{o_C} = \frac{8.30P}{\left[1-\frac{1}{2}\times\frac{Q_s}{Q}\right]} \times \frac{\frac{r}{c}f}{\frac{Q}{rcNl}}$$

or

$$\Delta T_{o_C} = T_{var}\frac{P}{\gamma C_H}$$

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

$$P = \frac{W}{ld}$$

μ dependent

μ is T_{ave} hence ΔT depend.

a) So,

$$\Delta T \rightarrow S \rightarrow \mu \rightarrow T_{ave} \rightarrow \Delta T$$

There is a vicious circle of which the starting point can not be found. Now, we have to use trail and error method of assuming *ΔT* or *μ.*

Let ΔT = 6 °C

$$T_{ave} = T_{in} + \frac{\Delta T}{2} = 50 + \frac{6}{2} = 53\ ^oC \rightarrow SAE \quad 10 \rightarrow \mu = 17\ mPa - \sec$$

$$\Delta T_{^oC} = T_{\text{var}} \frac{P}{\gamma C_H}$$

Where c is unknown → use Table 12.4

For Copper-lead r/c= 500-1000

Let r/c= 600;

Table 12-4 SOME CHARACTERISTICS OF BEARING ALLOYS

Alloy name	Thickness, mm	Clearance ratio r/c	Load capacity	Corrosion resistance
Tin-base babbit	0.559	600–1000	1.0	Excellent
Lead-base babbit	0.559	600–1000	1.2	Very good
Tin-base babbit	0.102	600–1000	1.5	Excellent
Leaded bronze	Solid	500–1000	3.3	Very good
Copper-lead	0.559	500–1000	1.9	Good
Aluminum alloy	Solid	400–500	3.0	Excellent
Silver plus overlay	0.330	600–1000	4.1	Excellent
Cadmium (1.5% Ni)	0.559	400–500	1.3	Good

$$S = \left(\frac{r}{c}\right)^2 \frac{\mu N}{P} \qquad P \rightarrow \frac{W}{ld}$$

So,

$$S = (600)^2 \frac{(17 \times 10^{-3}) \times 60}{[68/(0.01 \times 0.01)]} = (600)^2 \frac{0.017 \times 60}{680\ 000\ Pa} = 0.54$$

Use S-T_{var} chart → for S= 0.54 and l/d=1; T_{var} = 47

$$\Delta T = T_{\text{var}} \frac{P}{\gamma C_H} = 47 \frac{680\ 000}{861 \times 1760} = 21\ ^oC \qquad \Delta T_{ass} = 6\ ^oC < \Delta T_{calc} = 21\ ^oC$$

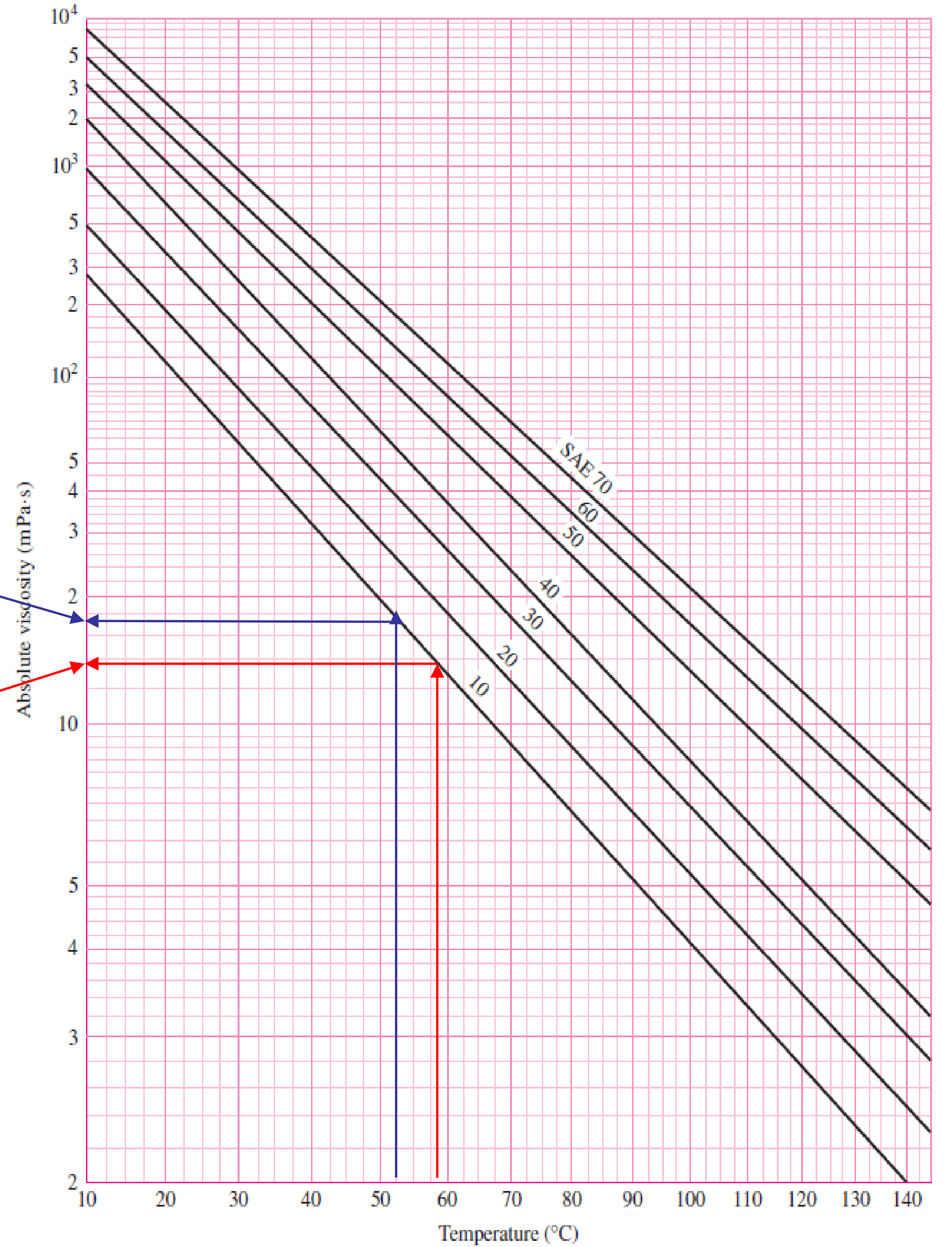
Now from chart Fig. 12.12 pp. 534 (Fig. 12.10 pp. 435)

$$T_{ave} = 53\ {}^{o}C \rightarrow SAE \quad 10$$

$$\rightarrow \mu = 17\ mPa - \text{sec}$$

$$T_{ave} = 58\ {}^{o}C \rightarrow SAE \quad 10$$

$$\rightarrow \mu = 14\ mPa - \text{sec}$$



Let $\Delta T = 16\ ^oC$

$$T_{ave} = T_{in} + \frac{\Delta T}{2} = 50 + \frac{16}{2} = 58\ ^oC \rightarrow SAE \quad 10 \rightarrow \mu = 14\ mPa - \text{sec}$$

$$S = (600)^2 \frac{0.014 \times 60}{680\ 000\ Pa} = 0.444 \longrightarrow$$ T_{var} = 38 and $\Delta T = 17\ ^oC$

$$\Delta T_{ass} \cong \Delta T_{calc}\ ; \quad \Delta T = 16\ ^oC$$

$$For \quad S = 0.54$$

$$\frac{l}{d} = 1$$

$$T_{\text{var}} = 47 \ ^oC$$

$$T_{\text{var}} = 38 \ ^oC$$

$$For \quad S = 0.444$$

$$\frac{l}{d} = 1$$

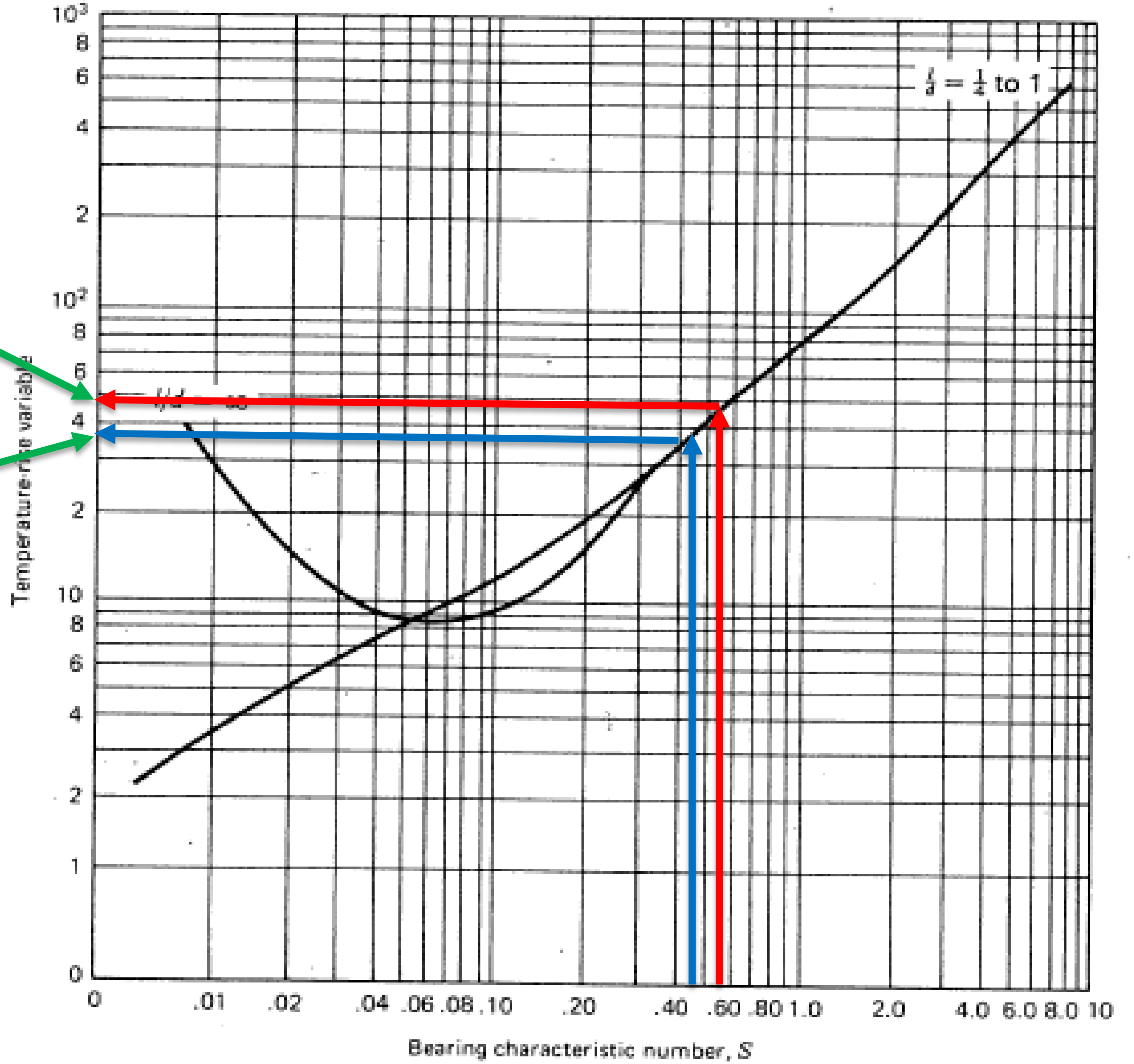


FIGURE 12-12 Chart for temperature-rise variable $T(\text{var}) = \gamma C_H \, \Delta T / P$. In plotting this chart it was found that the curves for $l/d = \frac{1}{4}, \frac{1}{2}$, and 1 were so close together that they could not be distinguished from a single curve.

b) h_o = ? h_o /c = ?

$$For\ \ S = 0.444$$

$$\frac{l}{d} = 1 \qquad \rightarrow \frac{h_o}{c} = 0.72 \quad h_o = 0.72 \times c$$

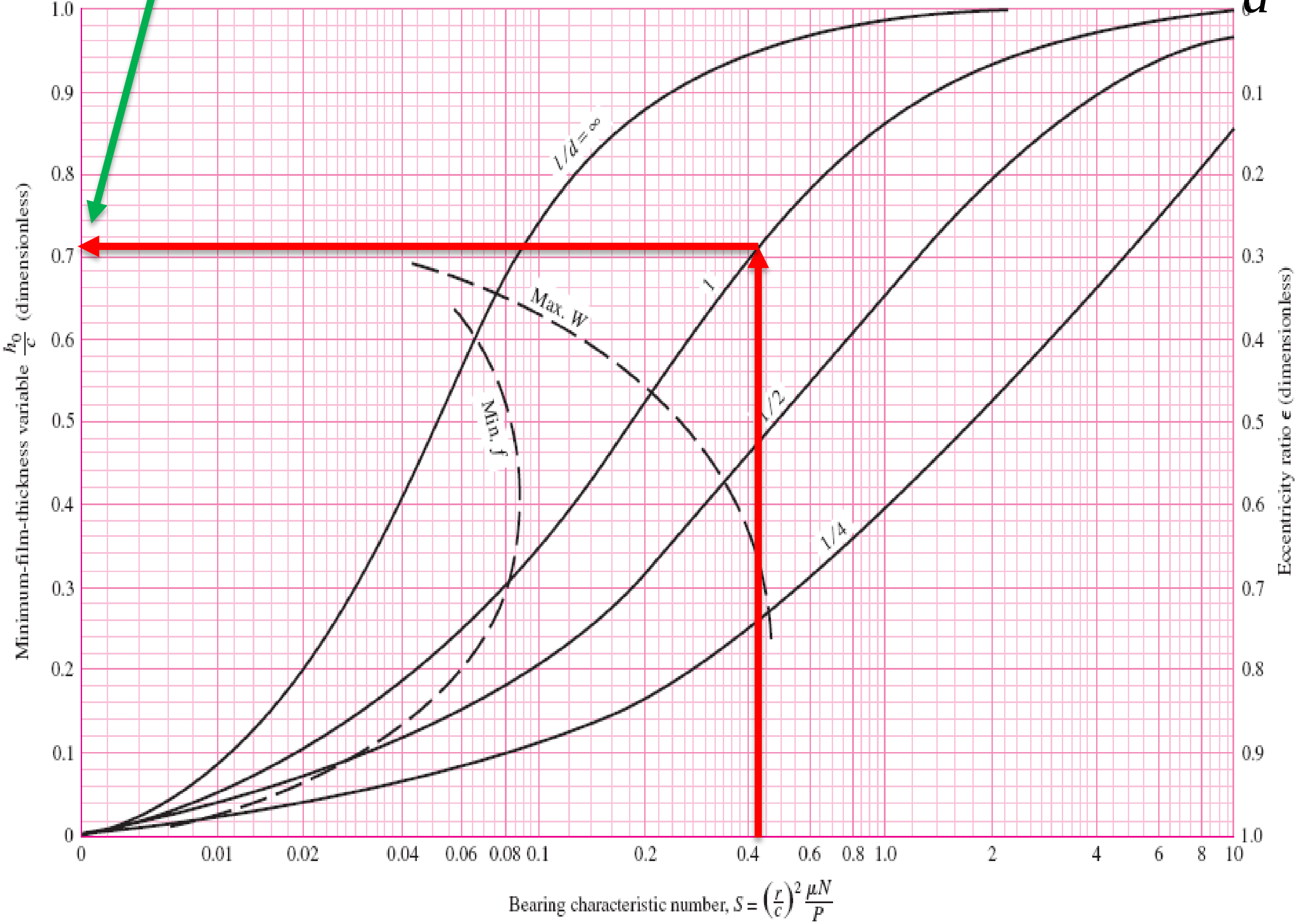
$$\frac{r}{c} = 600 \rightarrow c = \frac{r}{600} = \frac{5}{600} = 0.0083$$

$$h_o = 0.72 \times 0.0083 = \underline{\underline{6\ \mu m}}$$

$$\frac{h_0}{c} = 0.72 \quad (p543)(p446)$$

$$For \;\; S = 0.444$$

$$\frac{l}{d} = 1$$



c)

$$P_{loss} = T_{loss} \times w\left(\frac{rad}{\sec}\right) \qquad T_{loss} = F_{fric} \times r \qquad F_{fric} = W \times f$$

$$For \;\; S = 0.444$$

$$\frac{l}{d} = 1 \qquad \rightarrow \quad \frac{r}{c} f = 9.3 \rightarrow \quad f = \frac{9.3}{600} = 0.0155$$

$$W = 68\ N \quad radial\ load\ on\ bearing$$

$$F_{fric} = W \times f$$

$$F_{fric} = 68 \times 0.0155 = \quad 1.054\ N$$

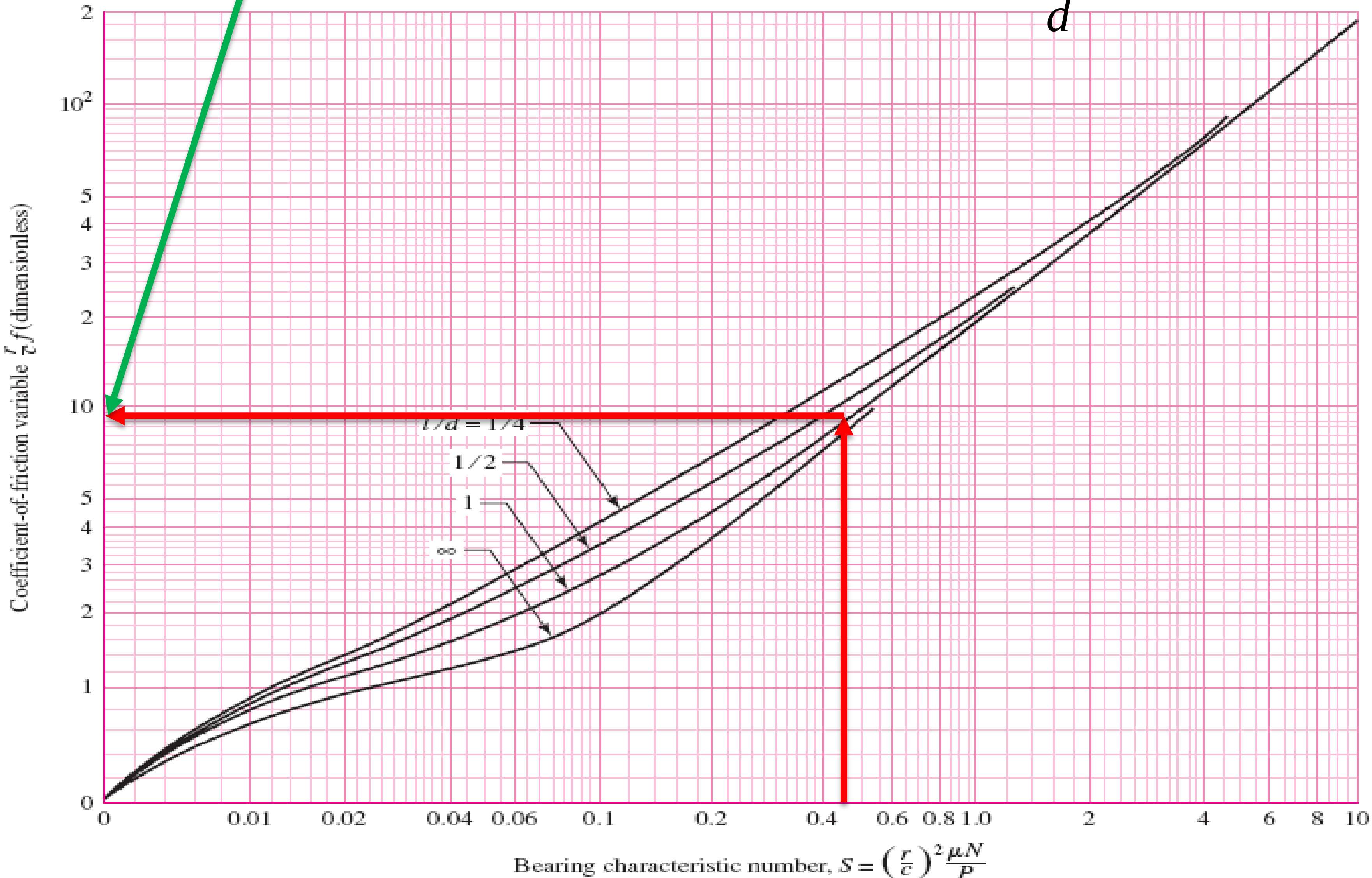
$$T_{loss} = 1.054 \times 0.005 = 5.27 \times 10^{-3}\ Nm$$

$$P_{loss} = 5.27 \times 10^{-3} \times 3600 \frac{rev}{\min} \times \frac{2\pi rad}{rev} \times \frac{\min}{60 \sec} = \underline{\underline{1.986\ watt}}$$

$$\left(\frac{r}{c}\right) f = 9.3 \quad (p540)(p444)$$

$$For \;\; S = 0.444$$

$$\frac{l}{d} = 1$$



EXAMPLE 4.2

An *80 mm* diameter full bearing on an air compressor is to be designed for a load of *6.6 kN*, *N*= *300 rpm*. It is desired that the bearing operates at a reasonable steady-state temperature (average temperature) of 50 ^{o}C. The journal is ground and operates in reamed-cast-bronze bearing.

a) Compute all unknown parameters of the bearing:
(d, l, c), h_0, f, P_{max}, Q, Q_s, ΔT
b) Specify the oil to be used (SAE ??)
c) Compute the amount of power loss in the bearing

SOLUTION:

GIVEN:

W= 6.6 kN
N= 300 rpm = 300/60= 5 rps
T_{ave}= 50 ^{o}C
Ground journal
Reamed-cast-bronze bearing
Air compressor

1) Since application type is known then *P* can be limited in table 12-2 pp.558 (12.3 pp. 457)
P= 1-2 MPa for **air compressor** main bearings

$$P = \frac{W}{l \times d} \qquad l = \frac{W}{P \times d} = \frac{6600}{1 \times 10^6 \times 0.08} = 0.0825\ m$$

$$l = 82.5\ mm$$

For *d= 80 mm l= 80 mm* $P = \frac{W}{ld}$

$$P = \frac{6600}{80 \times 80} = 1.031\ MPa \qquad \text{(in the range OK)}$$

Thus $\frac{l}{d} = \frac{80}{80} = 1$

Table 12-3 RECOMMENDED UNIT LOADS FOR SLEEVE BEARINGS

Application	Unit load, kPa	Application	Unit load, kPa
Air compressors:		Diesel engines:	
Main bearings	1 000– 2 000	Main bearings	6 000–12 000
Crankpin	2 000– 4 000	Crankpin	8 000–15 000
Automotive engines:		Wristpin	14 000–15 000
Main bearings	4 000– 5 000	Electric motors	800– 1 500
Crankpin	10 000–15 000	Gear reducers	800– 1 500
Centrifugal pumps	600– 1 200	Steam turbines	800– 1 500

2) Since bearing type is known the clearance value can be determined from Fig.12.29 pp.559 curve C. (Fig.12.27 pp.458 curve C.)

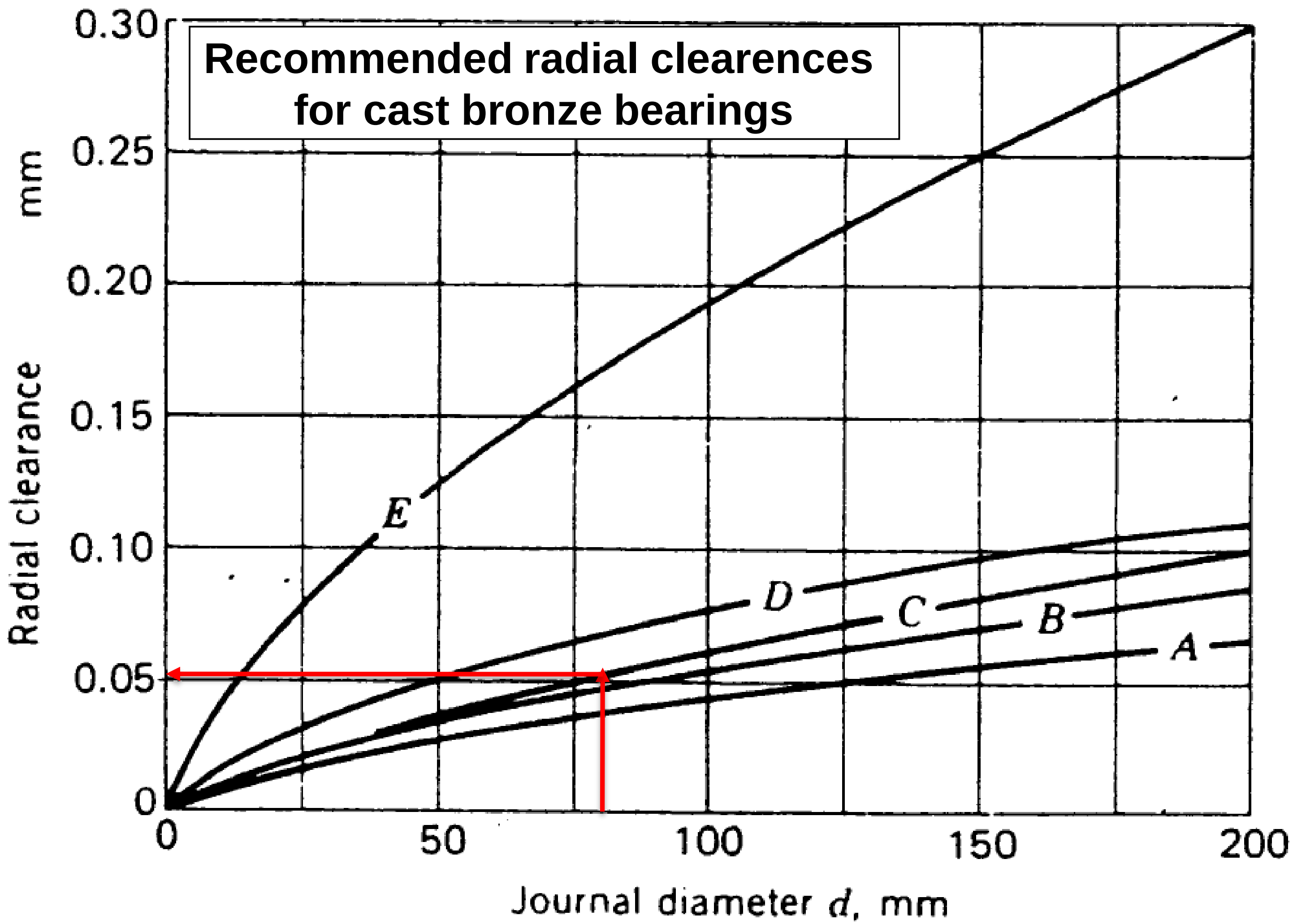


FIGURE 12-27. Recommended radial clearances for cast-bronze bearings. The curves are identified as follows:

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)

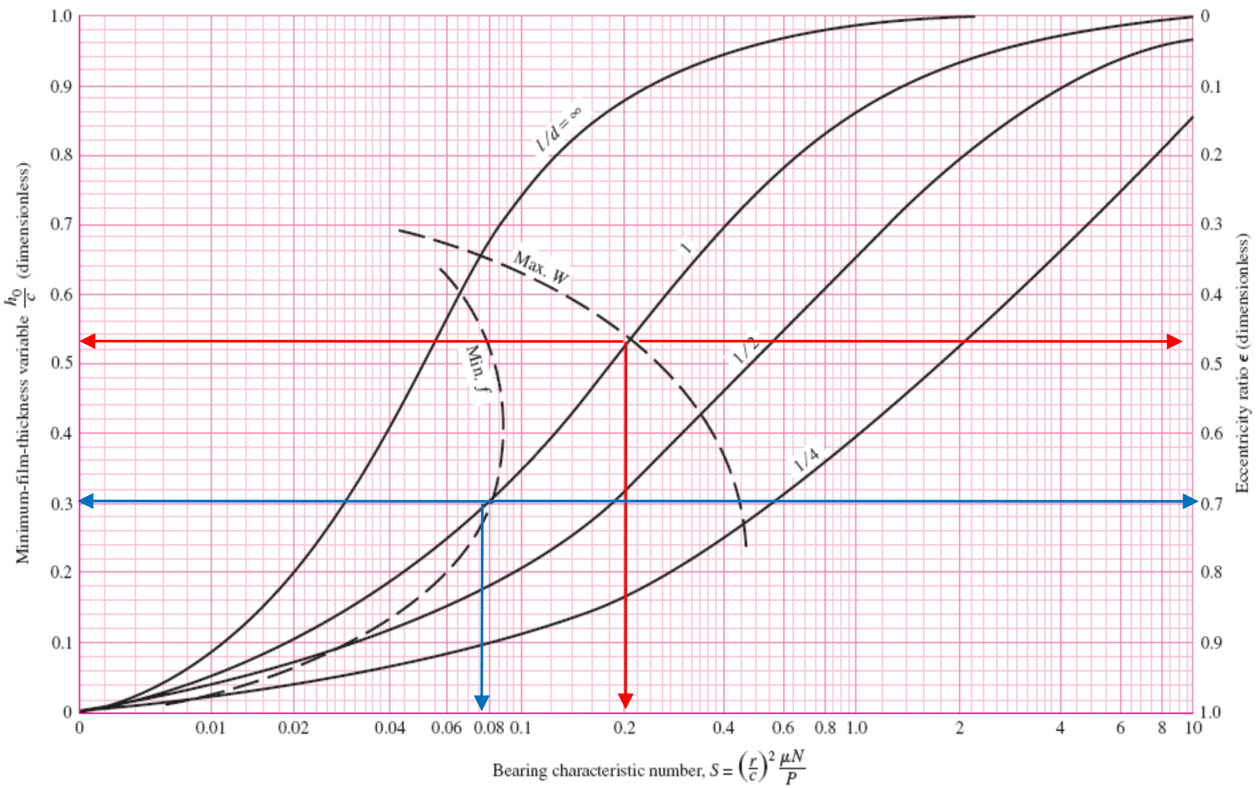
D—general machinery which continuously rotates or reciprocates and uses turned or cold-rolled steel journals in bored and reamed cast-bronze bearings (0.8- to 1.6-μm rms finish)

E—rough-service machinery having turned or cold-rolled steel journals operating on cast-bronze bearings (1.6- to 3.2-μm rms finish)

3) Now since $\frac{l}{d} = 1$ and $c = 0.05\ mm$

And none of the parameters are known we will make use of chart (f.12.14) of $\frac{h_0}{c}$ *vs* S *for maximum load carrying capacity cond.*

and for minimum friction cond.



For max. W

$$\frac{h_0}{c} = 0.53$$

$$S = 0.21$$

$$\varepsilon = 0.47$$

For min. Fric.

$$\frac{h_0}{c} = 0.3$$

$$S = 0.08$$

$$\varepsilon = 0.7$$

4) Since

$$S = \left(\frac{r}{c}\right)^2 \frac{\mu N}{P} = 0.21$$

Thus for max load carrying capacity condition

$$\mu = \frac{S \times P}{N}\left(\frac{c}{r}\right)^2 = \frac{0.21 \times 1.031 \times 10^6}{5}\left(\frac{0.05}{40}\right)^2 = 0.0676 \quad Pa - \mathrm{sec}$$

$$\mu = 67.6 \quad mPa - \mathrm{sec}$$

and for min friction condition

$$S = \left(\frac{r}{c}\right)^2 \frac{\mu N}{P} = 0.08$$

$$\mu = \frac{S \times P}{N}\left(\frac{c}{r}\right)^2 = \frac{0.08 \times 1.031 \times 10^6}{5}\left(\frac{0.05}{40}\right)^2$$

$$\mu = 25.75 \quad mPa - \mathrm{sec}$$

5) Now from chart Fig. 12.12 pp. 534 (Fig. 12.10 pp. 435)

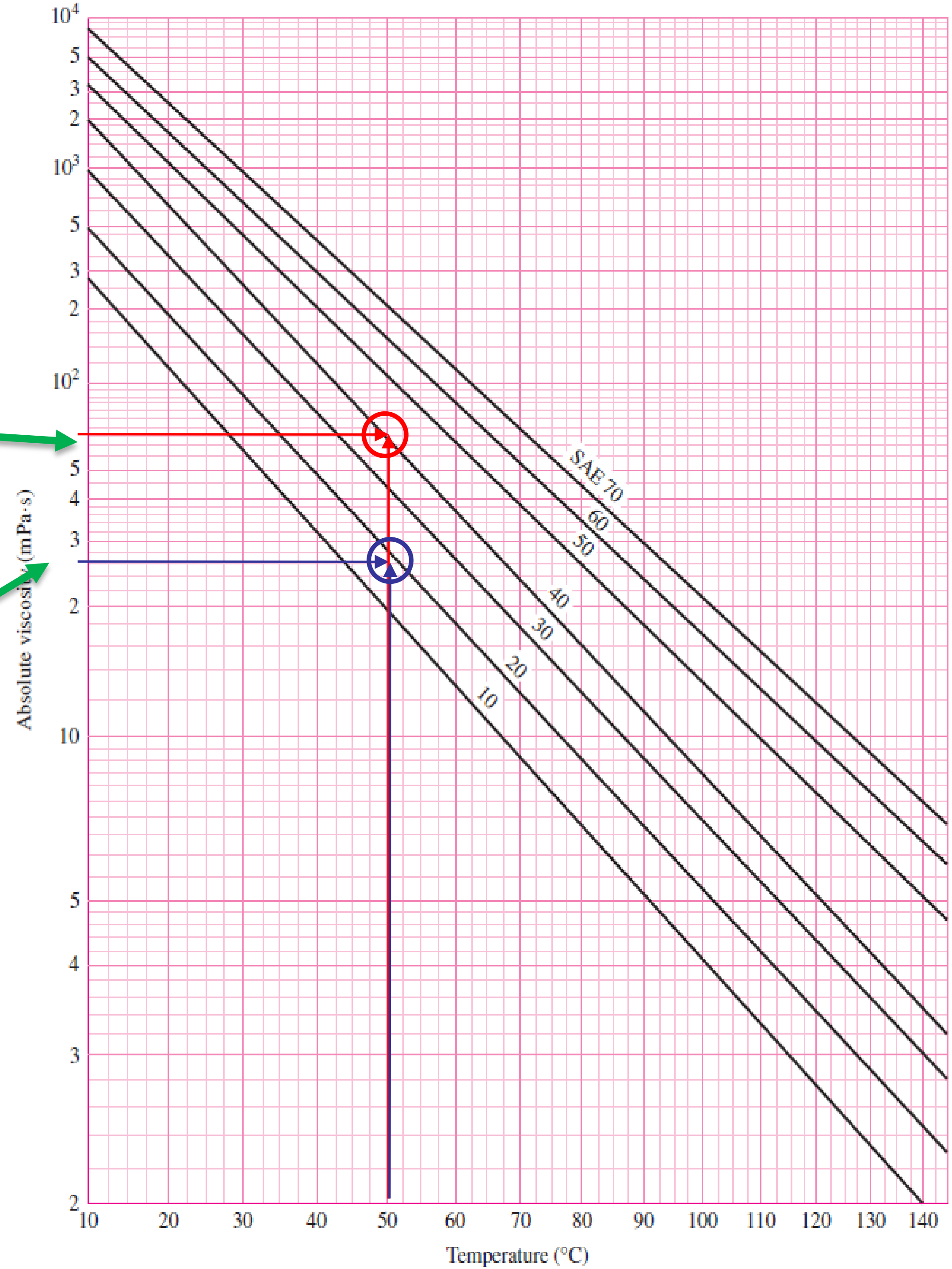
T_{ave}= 50 °C for both cases

$$\mu = 67.6 \ \ mPa - \sec$$

SAE 40 is the most suitable one for max. W

$$\mu = 25.75 \ \ mPa - \sec$$

and SAE 20 is suitable for f_{min}



6) Now because *S* and *l/d* are known most other parameters can be determined from Boyd chart (S=0.21 and l/d=1):

$$\left(\frac{r}{c}\right)f = 4.8 \quad \text{(p540) (p444)} \rightarrow\rightarrow \; f = 4.8\left(\frac{c}{r}\right) = 4.8\frac{0.05}{40} = 0.006$$

For max. load carrying capacity

$$\left(\frac{r}{c}\right)f = 2.4 \quad \textit{(p540)(p444)} \rightarrow\rightarrow \; f = 2.4\left(\frac{c}{r}\right) = 2.4\frac{0.05}{40} = 0.003$$

For min. friction condition

$$\frac{Q}{rcNl} = 4.1 \quad \textit{(p541)(p445)} \rightarrow\rightarrow Q = 4.1rcNl = 4.1\times 40\times 0.05\times 5\times 80 = 3280 \quad mm^3/\text{sec}$$

$$\frac{Q_s}{Q} = 0.55 \quad \text{(p542) (p446)} \rightarrow\rightarrow Q_s = 0.55\times 3280 = 1804 \quad mm^3/\text{sec}$$

$$\frac{P}{P_{\max}} = 0.46 \quad \text{(p543) (p446)} \rightarrow\rightarrow P_{\max} = \frac{P}{0.46} = \frac{1.031MPa}{0.46} = 2.241 \quad MPa$$

$$\frac{h_0}{c} = 0.53 \quad \textit{(p543)(p446)} \rightarrow\rightarrow h_0 = 0.53\times 0.05 = 0.0265\, mm = \underline{\underline{26.5\ \mu m}}$$

For $S = 0.08$

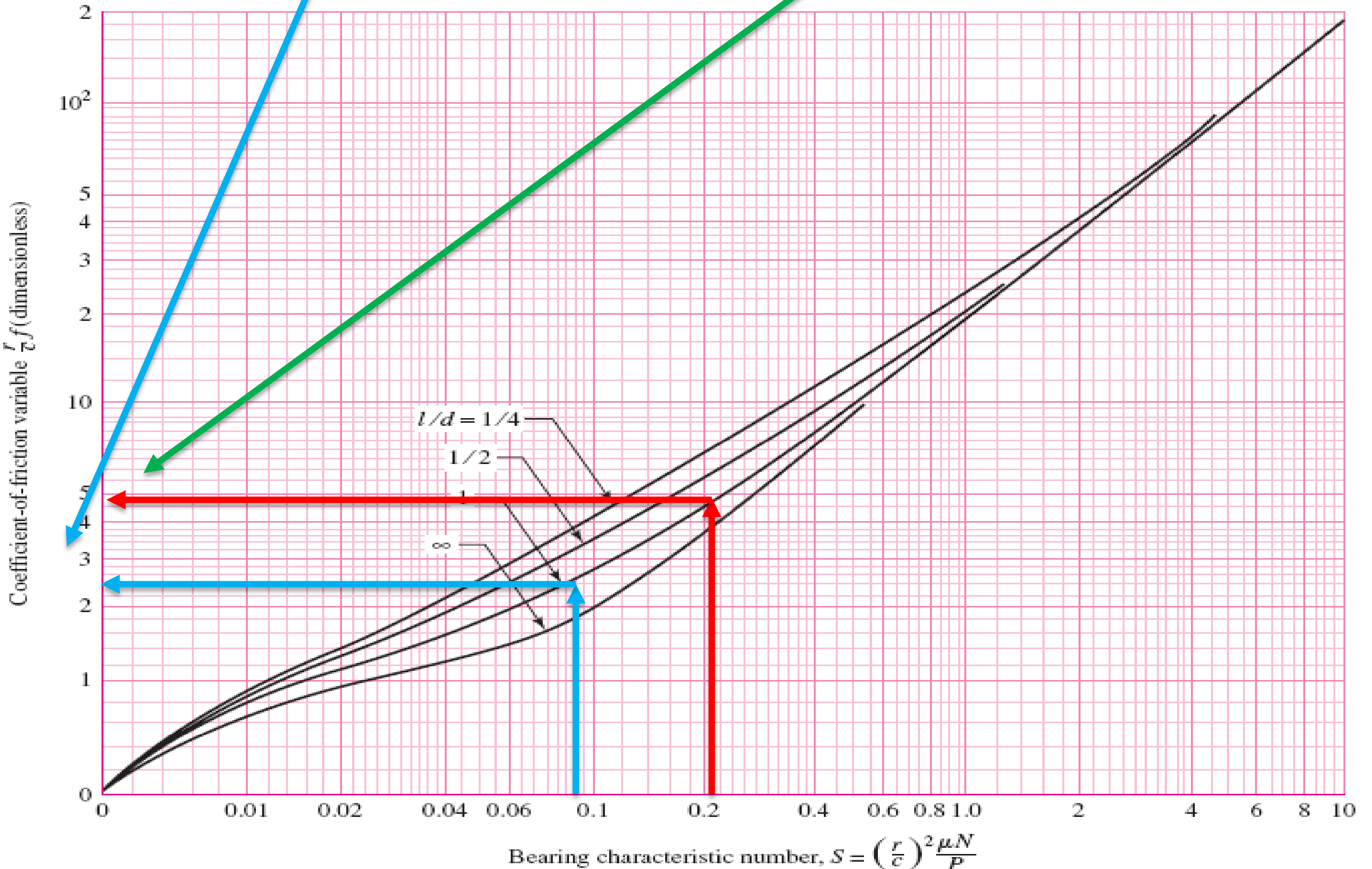
$\frac{l}{d} = 1$

$\left(\frac{r}{c}\right) f = 2.4 \quad (p540)(p444)$

$\left(\frac{r}{c}\right) f = 4.8 \quad (p540)(p444)$

For $S = 0.21$

$\frac{l}{d} = 1$



$$\frac{Q}{rcNl} = 4.1 \quad (p541)(p445)$$

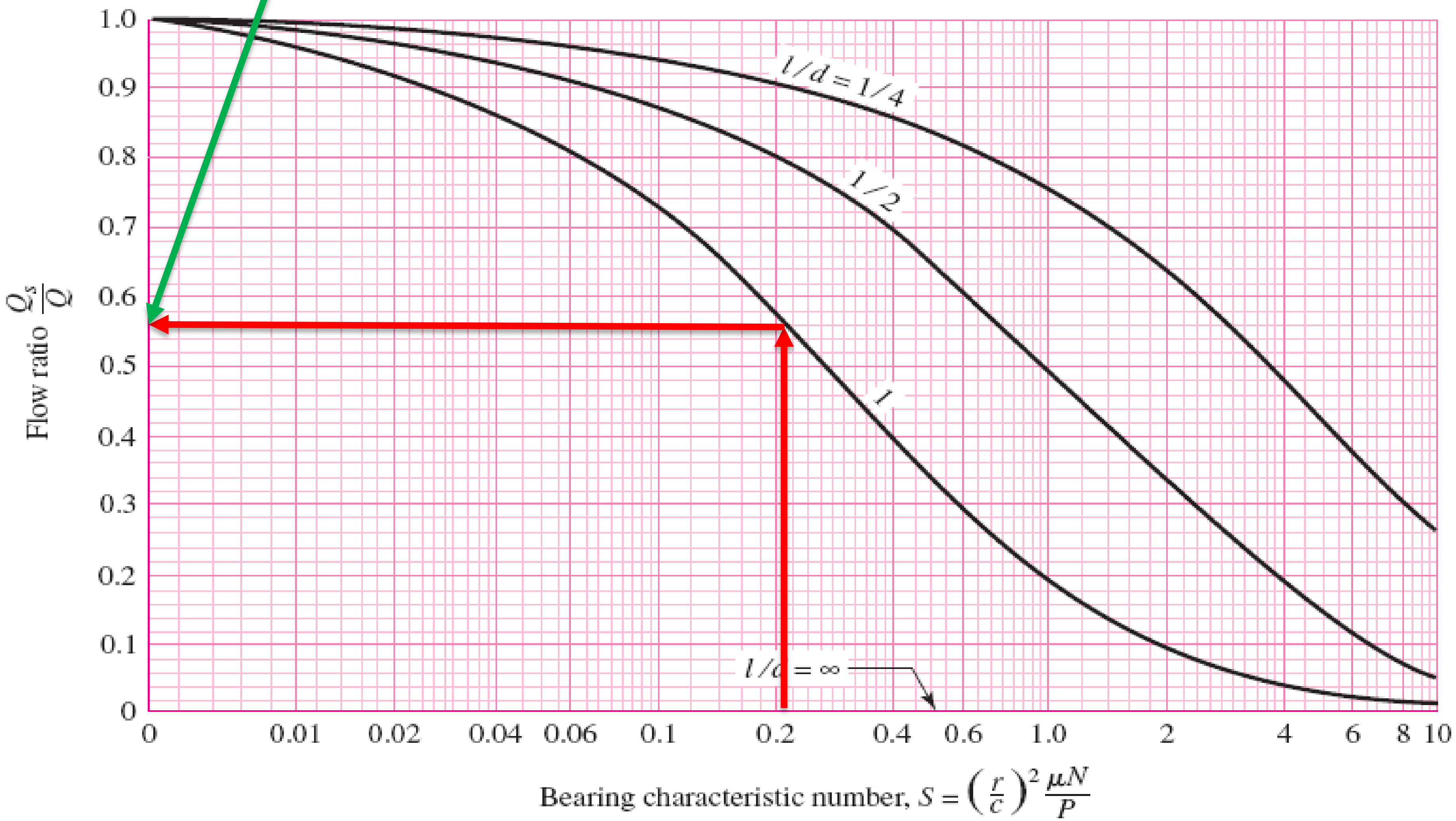
$$For \;\; S = 0.21$$

$$\frac{l}{d} = 1$$

$$\frac{Q_s}{Q} = 0.55 \quad (p542)(p446)$$

$$For \;\; S = 0.21$$

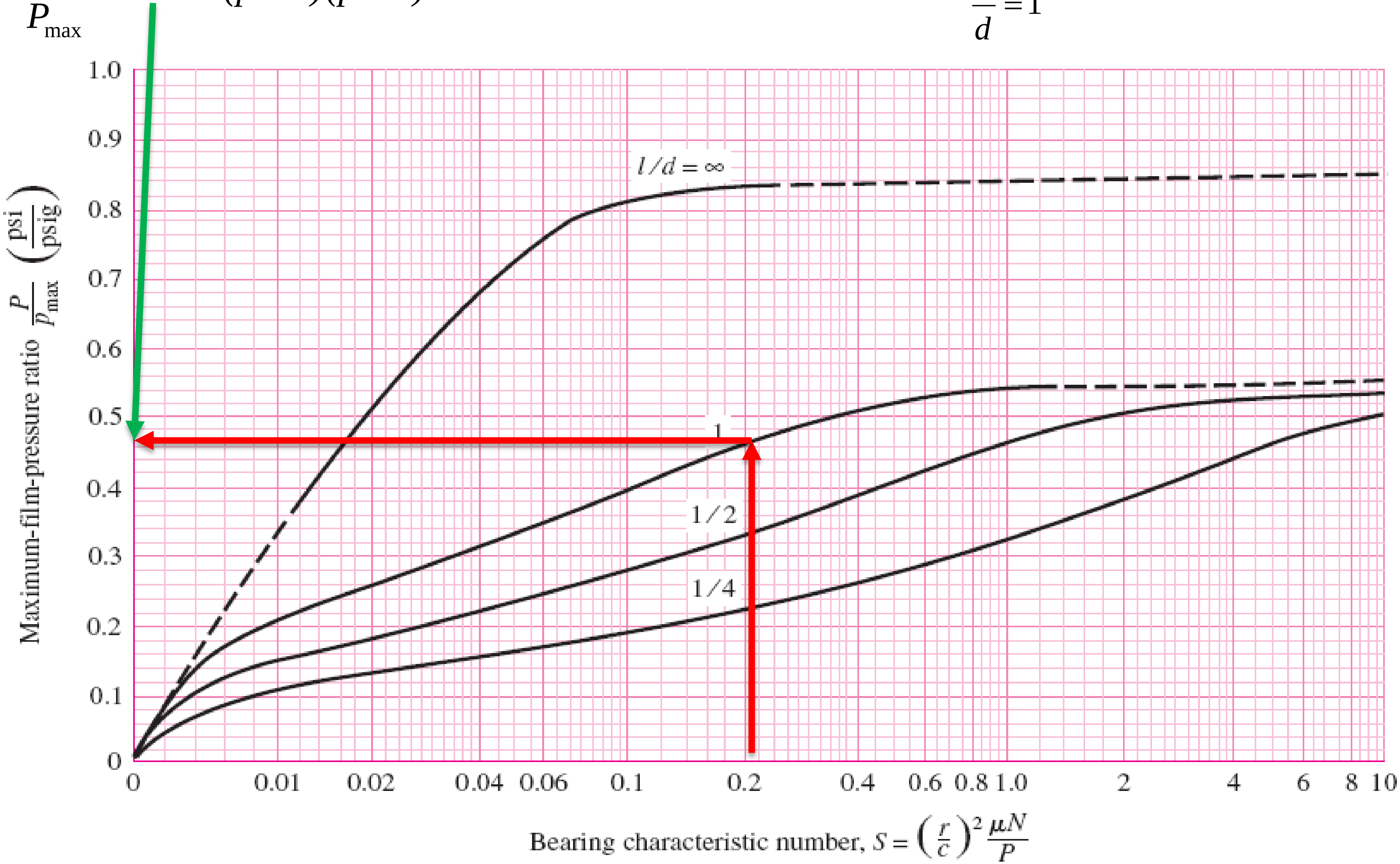
$$\frac{l}{d} = 1$$



$$\frac{P}{P_{max}} = 0.46 \quad (p543)(p446)$$

$$For \;\; S = 0.21$$

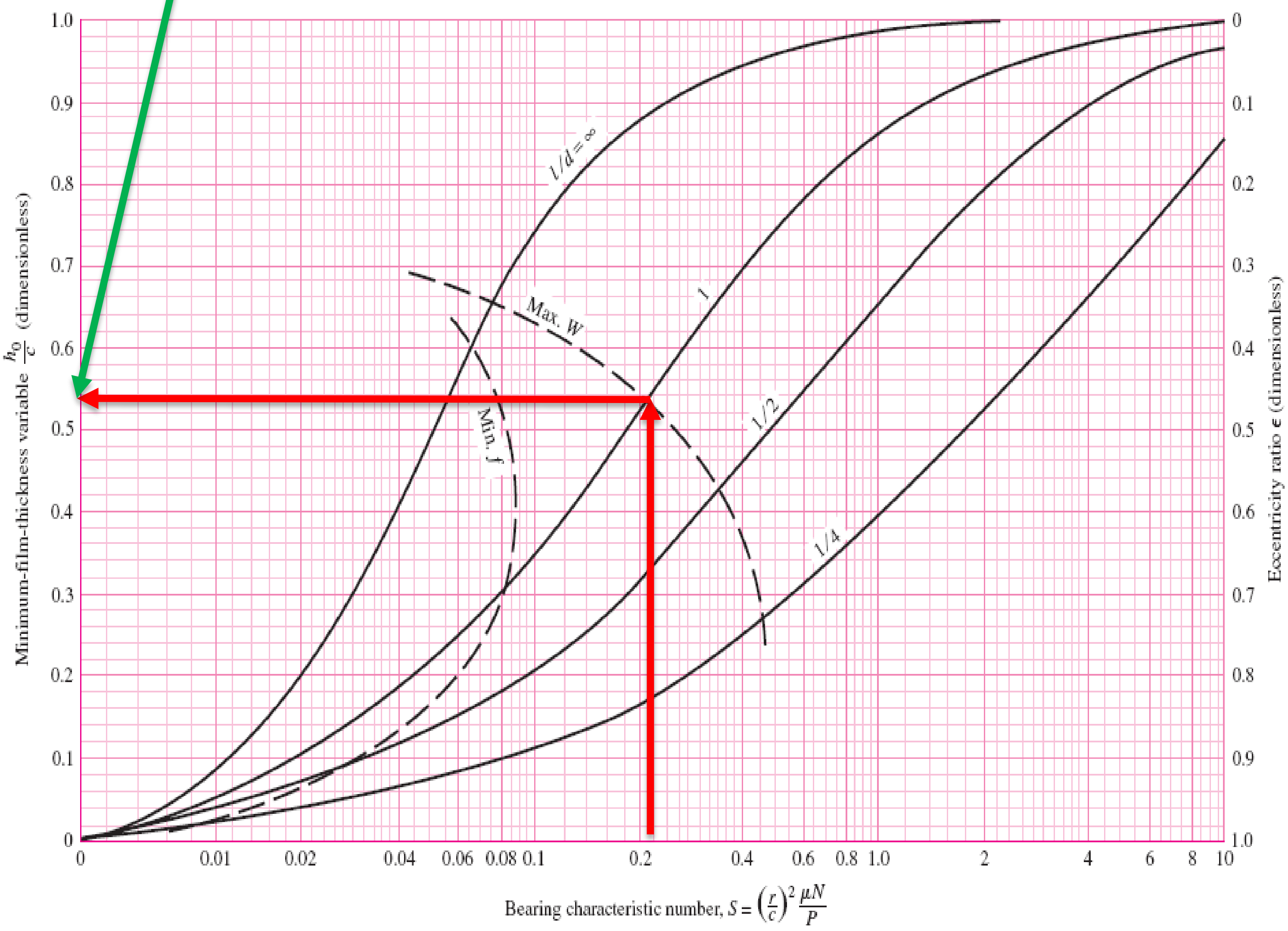
$$\frac{l}{d} = 1$$



$$\frac{h_0}{c} = 0.53 \quad (p543)(p446)$$

$$For \;\; S = 0.21$$

$$\frac{l}{d} = 1$$



7)

$$\Delta T_{^oC} = \frac{8.30P}{\left[1-\frac{1}{2}\times\frac{Q_s}{Q}\right]}\times\frac{\frac{r}{c}f}{\frac{Q}{rcNl}} = \frac{8.3\times 1.031MPa}{1-\frac{1}{2}\times 0.55}\times\frac{4.8}{4.1} = 13.8\ \ ^oC$$

$$\Delta T_{^oC} = T_{\text{var}}\frac{P}{\gamma C_H} = ??\ ^oC \qquad T_{\text{var}} = ?$$

$$Fig.12.12\,(S=0.21),\ \frac{l}{d}=1\ \ \rightarrow T_{\text{var}} = 20\ ^oC$$

$$\Delta T_{^oC} = 20\times\frac{1.031\times 10^6}{861\times 1760} = 13.7\ ^oC$$

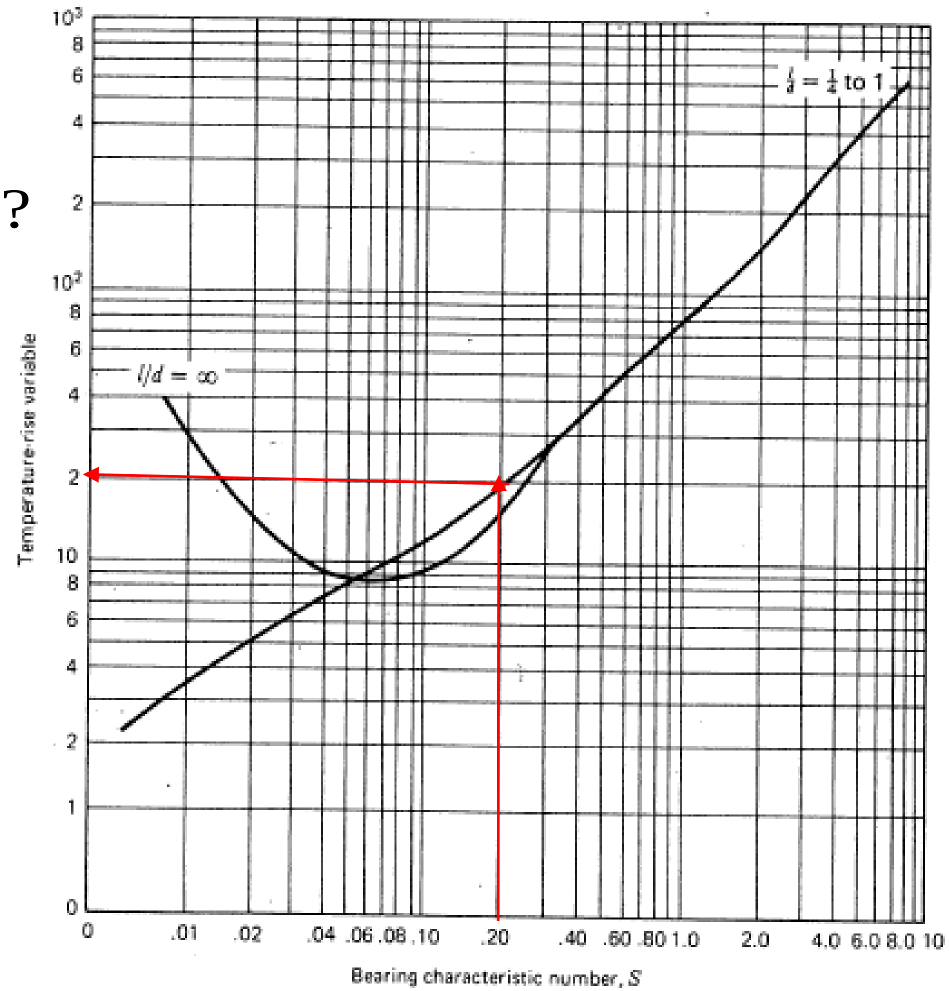


FIGURE 12-12 Chart for temperature-rise variable $T(\text{var}) = \gamma C_H\,\Delta T/P$. In plotting this chart it was found that the curves for $l/d = \frac{1}{4}, \frac{1}{2}$, and 1 were so close together that they could not be distinguished from a single curve.

8) The torque required to overcome the friction is

$$T_f = f \times W \times r = 0.006 \times 6600 \times 0.04 = 1.584\ Nm$$

Thus for max load capacity condition

$$T_f = f \times W \times r = 0.003 \times 6600 \times 0.04 = 0.792\ Nm$$

again for min friction condition

The power loss is

$$P_{loss} = T_f \times \omega_{\frac{rad}{\sec}} = 1.584 \times 300\ \frac{rev}{\min} \times \frac{\min}{60\sec} \times \frac{2\pi rad}{rev} = 49.76\ Nm/\sec$$

Thus for max load capacity condition

$$P_{loss} = 49.76\ watt$$

again for min friction condition

$$P_{loss} = 24.88\ watt$$

As a last check

$$PV_{\max} = 1750\ kPa \cdot m/s$$

$$PV_{\max} = 50\,000\ \ psi\!-\ fpm$$

for cast bronze bearings for boundary lubrication in Table 12.4 pp.565, or (12.5 pp.463)

Table 12-5 SOME MATERIALS FOR BOUNDARY-LUBRICATED BEARINGS AND THEIR OPERATING LIMITS

Material	Maximum pressure, MPa	Maximum temperature, °C	Maximum speed, m/s	Maximum *PV* value, kPa · m/s
Cast bronze	31.0	165	7.5	1750
Porous bronze	31.0	65	7.5	1750
Porous iron	55.0	65	4.0	1750
Phenolics	41.0	95	13.0	530
Nylon	7.0	95	5.0	100
Teflon	3.5	260	0.5	35
Reinforced teflon	17.0	260	5.0	350
Teflon fabric	410	260	0.3	900
Delrin	7.0	80	5.0	100
Carbon-graphite	4.2	400	13.0	530
Rubber	0.4	65	20.0	. . .
Wood	14.0	65	10.0	530

$$P = 1.031\ MPa\left(\times 0.000145\right) = 150\ \ psi$$

$$V = N \times \pi d = 5\frac{rev}{\sec} \times \pi \times 0.08m = 1.256\frac{m}{\sec}\left(\frac{60\sec}{1\min} \times \frac{ft}{0.3048m}\right) = 247\frac{ft}{\min}$$

$$\text{in } \ lb - in \quad \rightarrow PV = 150 \times 247 = 37050\ psi -\ fpm < 50\,000\ psi -\ fpm \qquad \text{OK}$$

$$\text{in } \ SI \quad \rightarrow PV = 1031\,kPa \times 1.256 \cong 1295\,kPa \cdot m/s < 1750\,kPa \cdot m/s \qquad \text{OK}$$

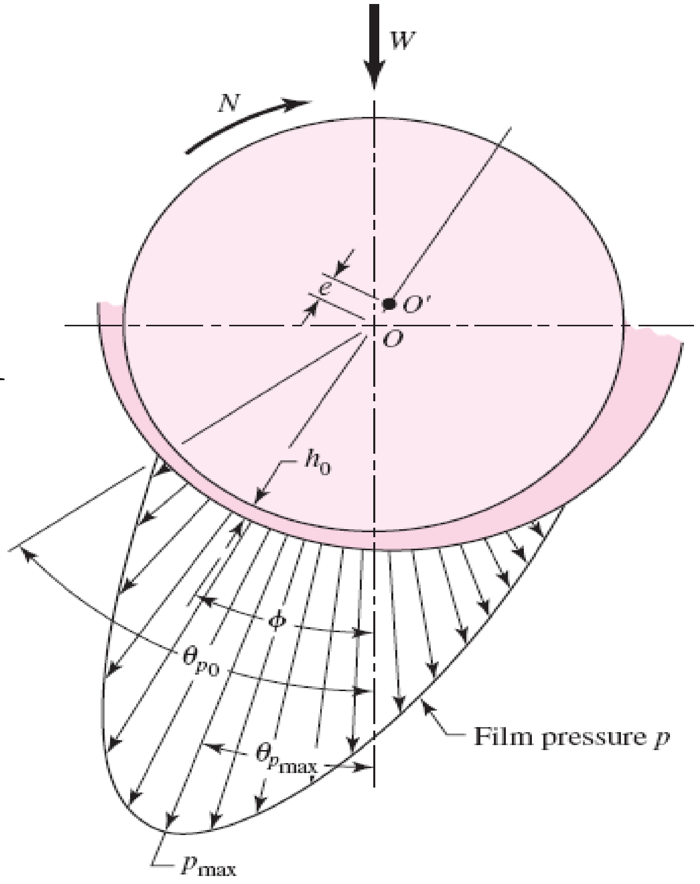
Thus for max load capacity condition

$$if\ required\ \left(\theta_{P_0}\ \&\ \theta_{P_{max}}\right)\quad for\quad S=0.21,\quad \frac{l}{d}=1$$

$$from\ Fig.12.21\ \ pp.554$$

$$\theta_{P_0}=85^o\ \&\ \theta_{P_{max}}=17.5^o$$

again for min friction condition



$$if\ required\ \left(\theta_{P_0}\ \&\ \theta_{P_{max}}\right)\quad for\quad S=0.08,\quad \frac{l}{d}=1$$

$$from\ Fig.12.21\ \ pp.554$$

$$\theta_{P_0}=64^o\ \&\ \theta_{P_{max}}=19^o$$

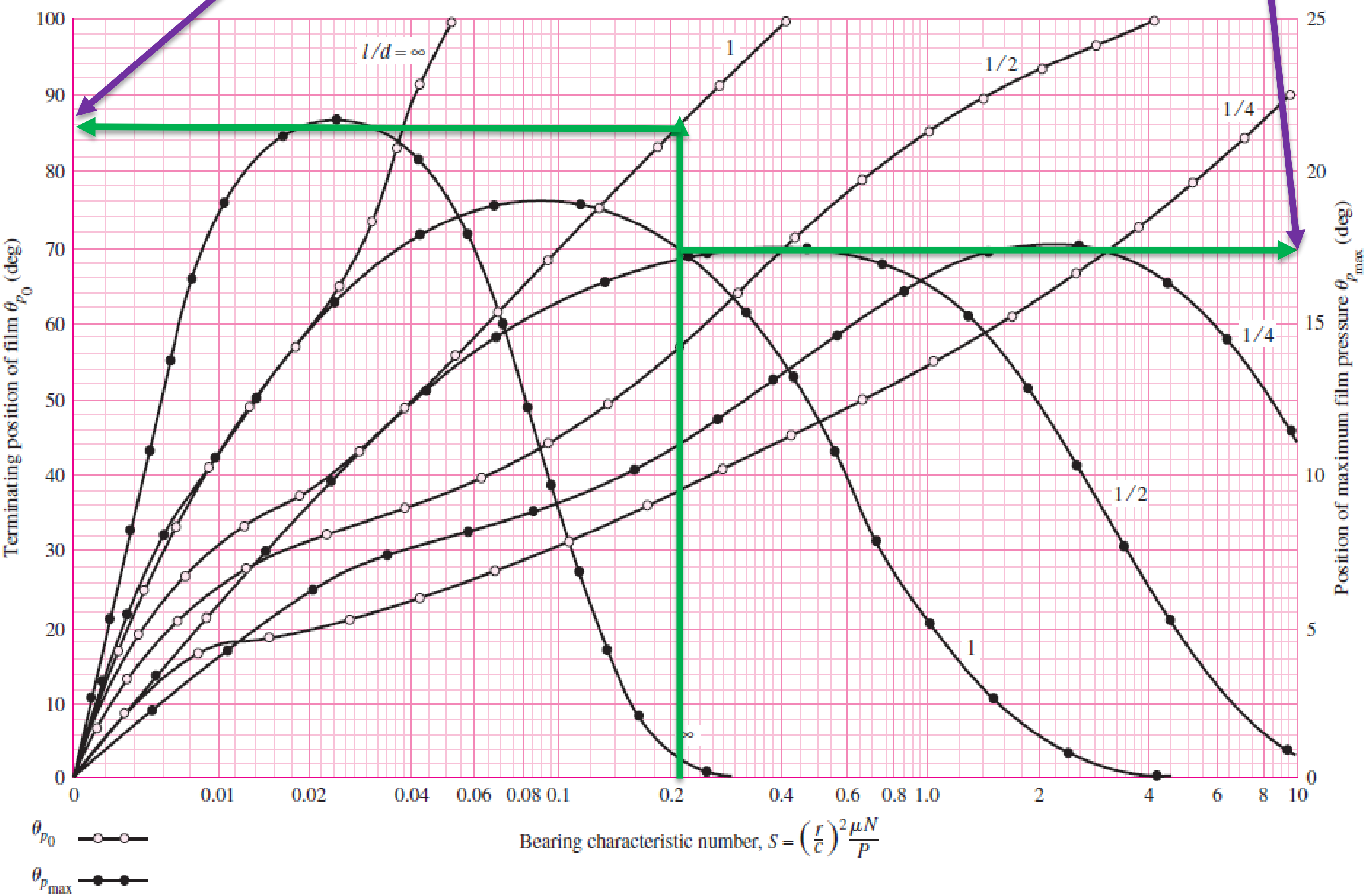
Thus for max load capacity condition

$\theta_{P_0} = 85^o$

$For \ \ S = 0.21$

$\frac{l}{d} = 1$

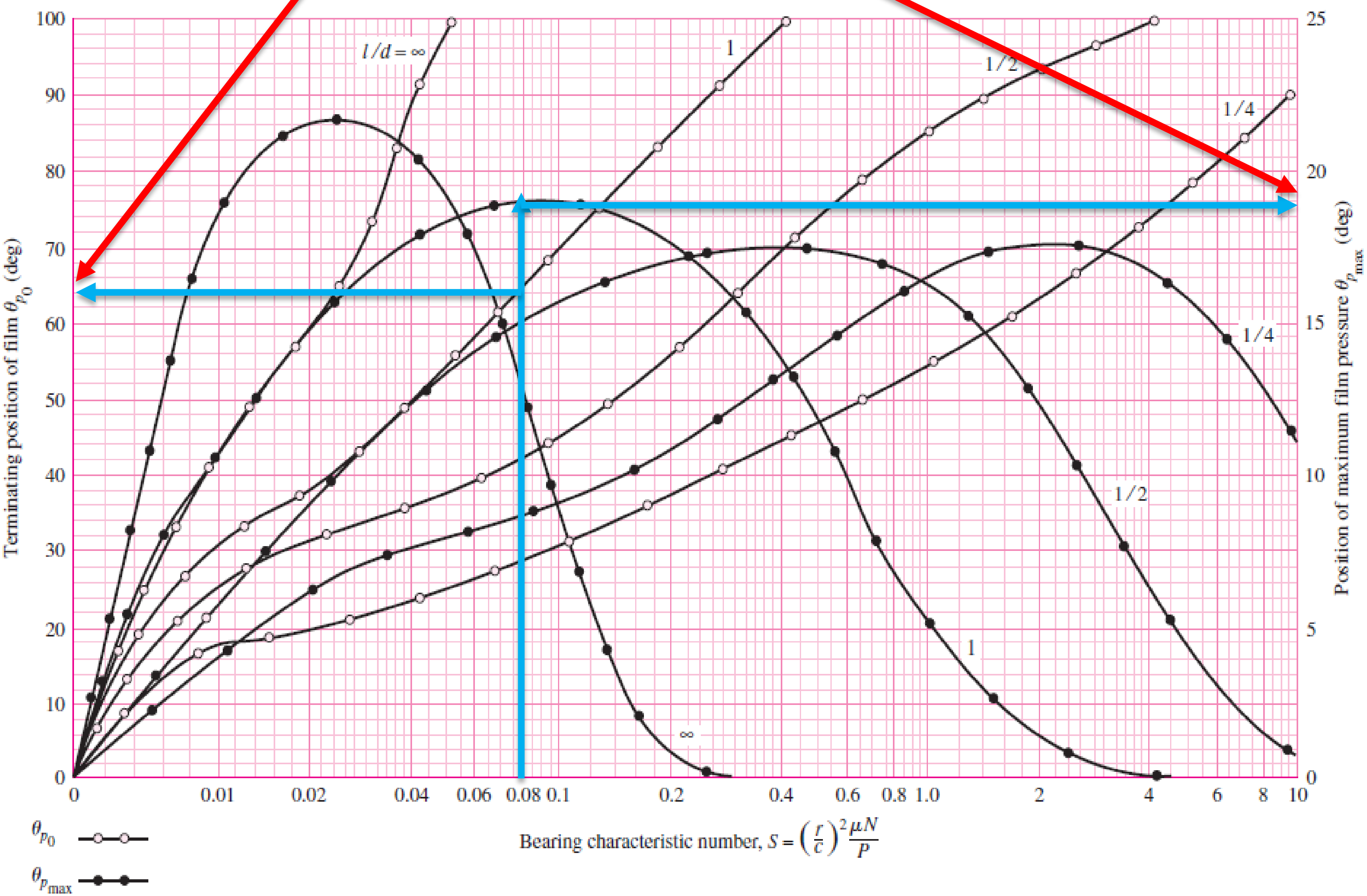
$\theta_{P_{max}} = 17.5^o$



for min friction condition

$$For \;\; S = 0.08$$

$$\theta_{P_0} = 64^o \qquad \frac{l}{d} = 1 \qquad \theta_{P_{max}} = 19^o$$



4.11 OPTIMIZATION OF J.B's

In designing a journal bearing for thick-film lubrication,
the engineer must :

- select the grade of oil to be used and
- determine suitable values for P, N, r, c, and l.

A poor selection of these parameters or inadequate control of them during manufacture or in use may result in a film that is too thin, so that the oil flow is insufficient, causing the bearing to overheat and, eventually, fail.

Furthermore, the radial clearance c is difficult to hold accurate during manufacture, and it may increase because of wear.

What is the effect of an entire range of radial clearances, expected in manufacture, and what will happen to the bearing performance if c increases because of wear?

Most of these questions can be answered and the design optimized by plotting curves of the performance as functions of the quantities over which the designer has control.

Figure shows the results obtained when the performance of a particular bearing is calculated for a whole range of radial clearances and is plotted with clearance as the independent variable.

The graph shows that if the clearance is too tight, the outlet temperature T_2 will be too high and the minimum film thickness (h_o) too low.

High temperatures may cause the bearing to fail by fatigue. If the oil film is too thin, dirt particles may be unable to pass without scoring or may embed themselves in the bearing.

In either event, there will be excessive wear and friction, resulting in high temperatures and possible seizing.

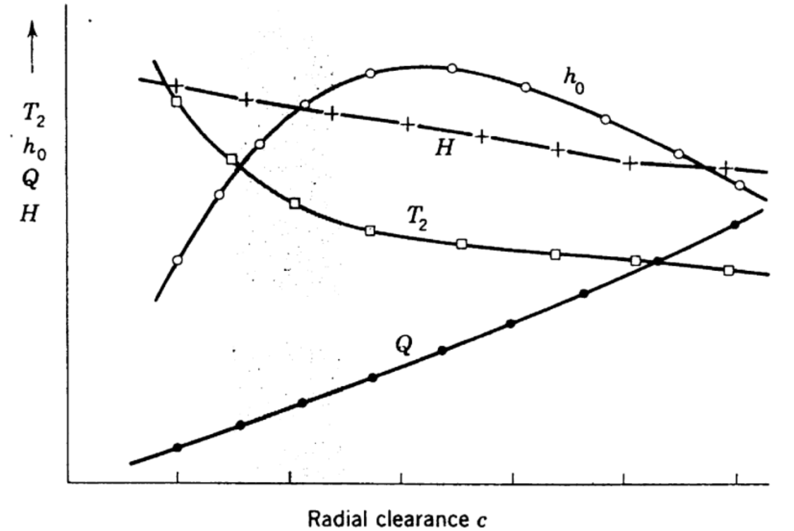


FIGURE 12-21 A plot of some performance characteristics. The bearing outlet temperature is T_2. New bearings should be designed for the shaded zone because wear will move the operating point to the right.

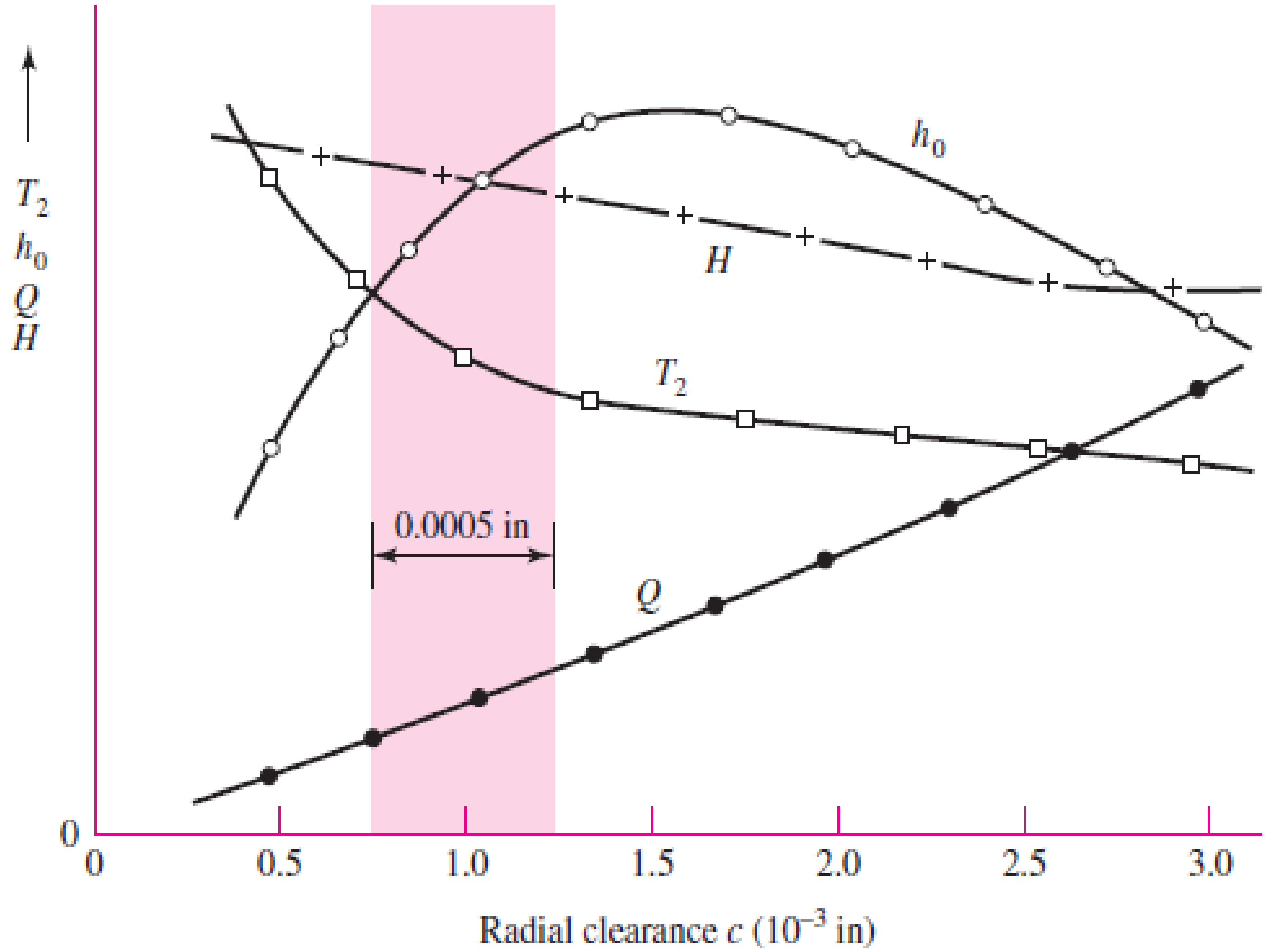


Fig 12.25 A plot of some performance characteristics of the bearing of Exs. 12–1 to 12–4 for radial clearances of 0.0005 to 0.003 in. The bearing outlet temperature is designated T2. New bearings should be designed for the shaded zone, because wear will move the operating point to the right.

A tight clearance results in a high temperature. It would seem that a large clearance will permit the dirt particles to pass through and also will permit a large flow of oil. This lowers the temperature and increases the life of the bearing. However, if the clearance becomes too large, the bearing becomes noisy and the minimum film thickness begins to decrease again.

In between these two limitations there exists a rather large range of clearances that will result in satisfactory bearing performance.

When both the production tolerance and the future wear on the bearing are considered, it is seen, from Fig. 12–25, that the best compromise is a clearance range slightly to the left of the top of the minimum-film-thickness curve.

In this way, future wear will move the operating point to the right and increase the film thickness and decrease the operating temperature

4.12 PRESSURE - FED BEARINGS (HYDRO-STATIC JOURNAL BEARING)

The load-carrying capacity of self-contained natural-circulating journal bearings is quite restricted.

The factor limiting better performance is the heat-dissipation capability of the bearing.

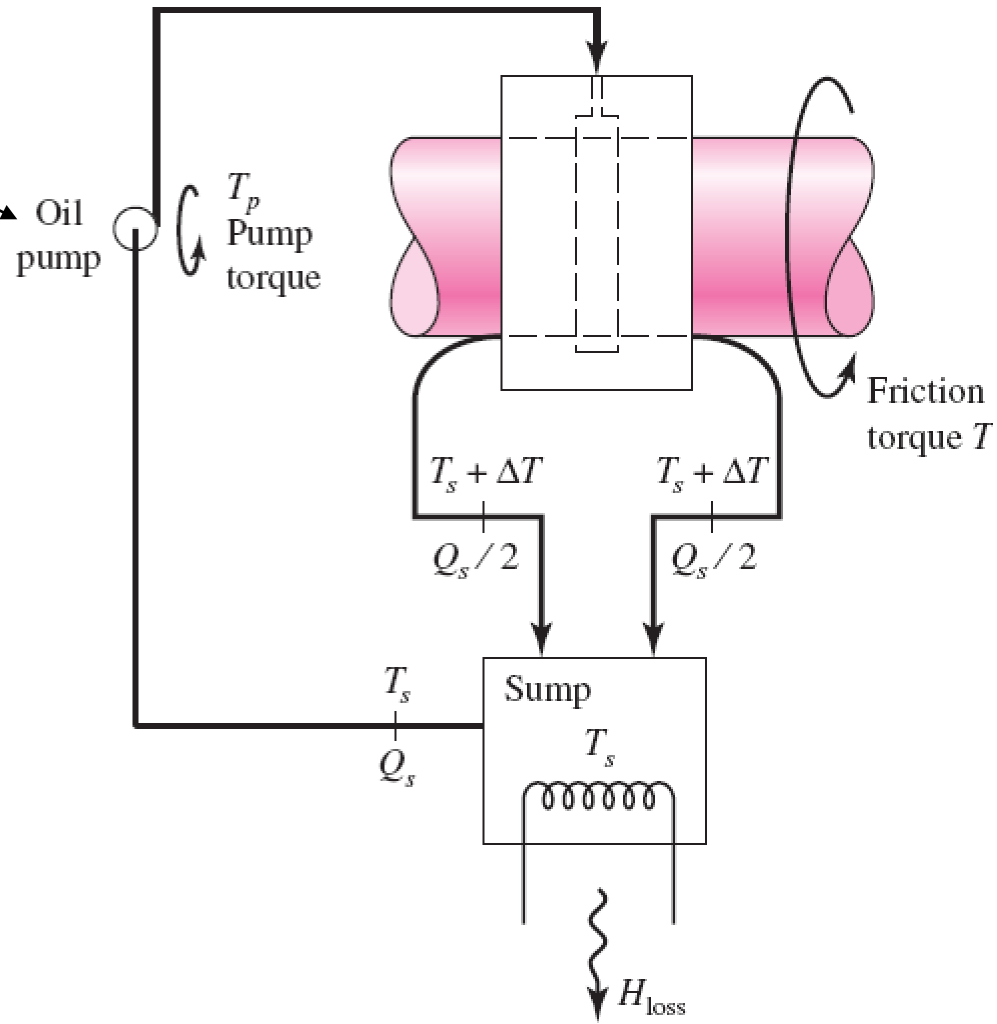
A first thought of a way to increase heat dissipation is to cool the sump with an external fluid such as water. The high-temperature problem is in the film where the heat is generated but cooling is not possible in the film until later. This does not protect against exceeding the maximum allowable temperature of the lubricant.

A second alternative is to reduce the *temperature rise* in the film by dramatically increasing the rate of lubricant flow.

To increase lubricant flow, <u>an external pump</u> must be used with lubricant supplied at pressures higher than atmospheric pressure.

Because the <u>lubricant is supplied to the bearing under pressure</u>, such bearings are called:

<u>pressure-fed bearings</u>.



To force a greater flow through the bearing and thus obtain an increased cooling effect, a common practice is to use a circumferential groove at the center of the bearing, with an oil-supply hole located opposite the load-bearing zone.

Such a bearing is shown in Fig. 12–27. The effect of the groove is to create two half-bearings, each having a smaller *l/d* ratio than the original.

The groove divides the pressure-distribution curve into two lobes and reduces the minimum film thickness, but it has wide acceptance among lubrication engineers because such bearings carry more load without overheating.

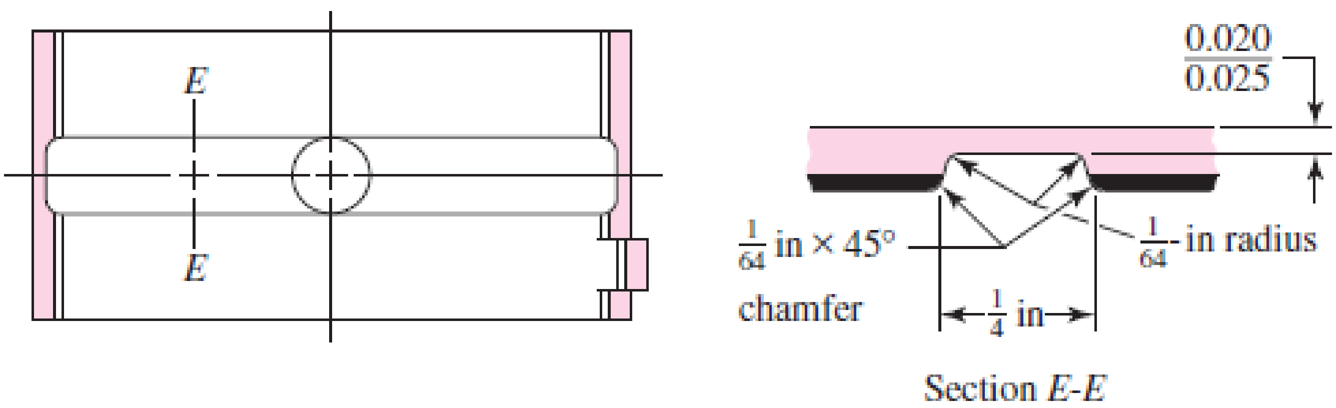
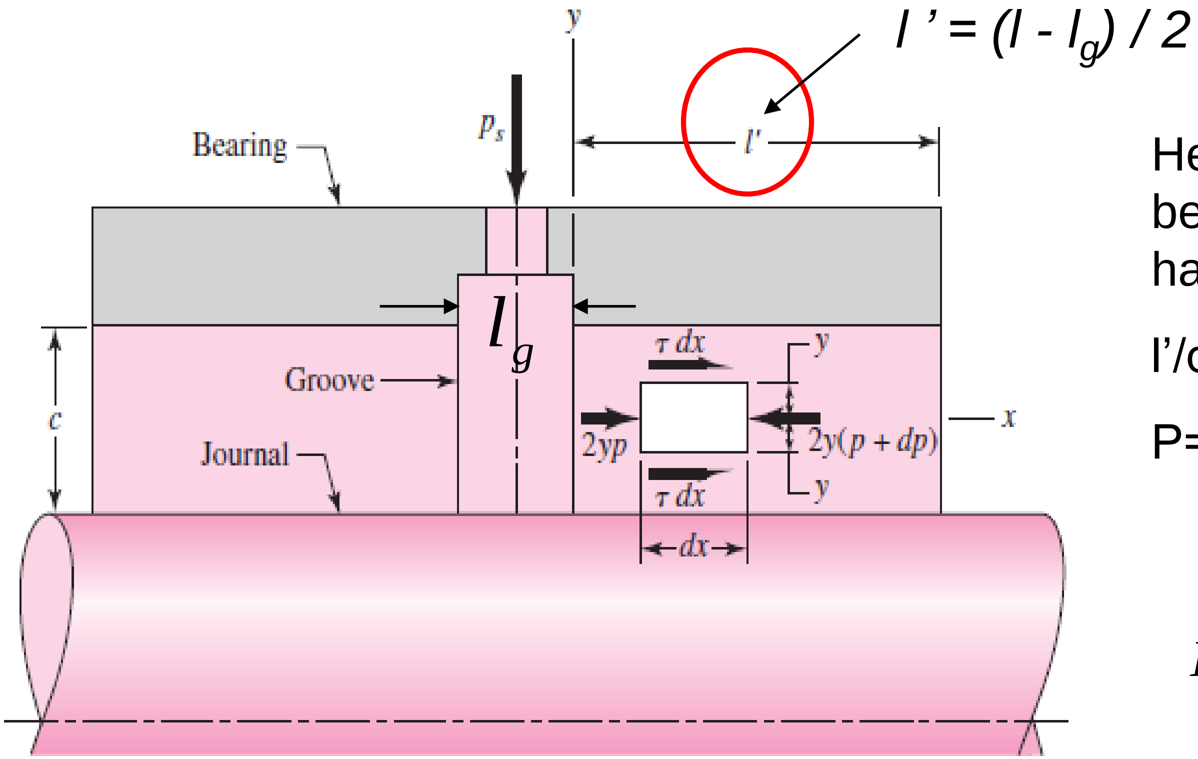


Fig. 12.27 Centrally located full annular groove.



Here we assume that the bearing is two fold each having properties:

l'/d (instead of l/d) and

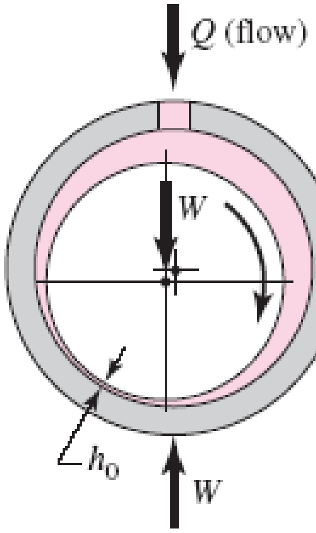
P=(W/2)/(l'd)

$$P = \frac{W/2}{l' \times d}$$

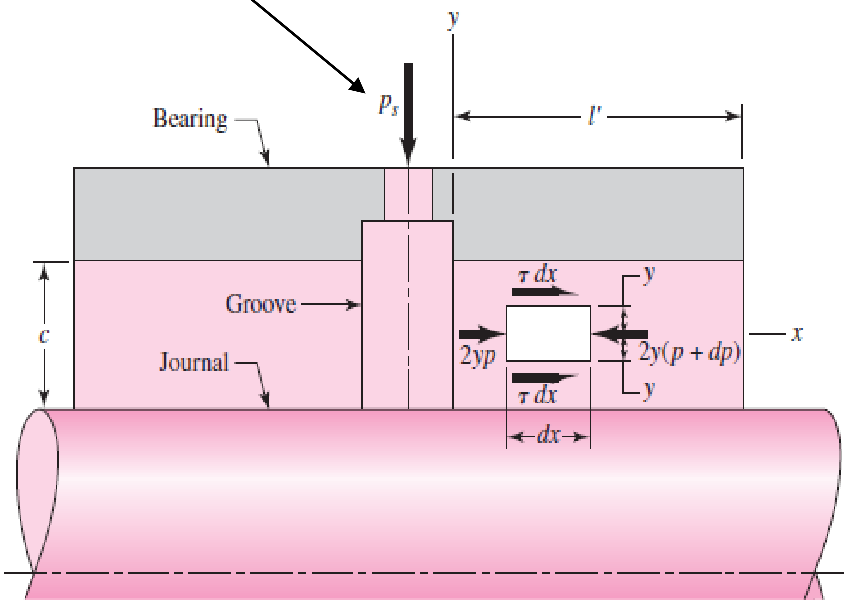
Fig. 12.28 Flow of lubricant from a pressure-fed bearing having a central annular groove.

These pressure-fed (hydro-static) bearings do not rely on wedging action of the lubricant by the rotation of journal only but lubricant is forced into the bearing clearance under a pressure p_s

This is the method used in application where the normal lubricant flow Q is not sufficient to carry away the heat generated by the hydrodynamic action within the bearing clearance for the load carried is high to be hold up.



(b) Lubricated



To set up a method of solution for oil flow rate Q, we shall assume a groove ample enough that the pressure drop in the groove itself is small.

Initially we will neglect eccentricity and then apply a correction factor for this condition.

The oil flow, then, is the amount that flows out of the two halves of the bearing in the direction of the concentric shaft.

If we neglect the rotation of the shaft, the flow of the lubricant is caused mainly by the supply pressure p_s.

Figure 12–29 shows a graph of this relation fitted into the clearance space *c* so that you can see how the velocity of the lubricant varies from the journal surface to the bearing surface.

The distribution is parabolic, as shown, with the maximum velocity occurring at the center, where *y* = 0. The magnitude is, from Eq. (12–21),

$$u_{\max} = \frac{p_s c^2}{8\mu l'} \qquad (i)$$

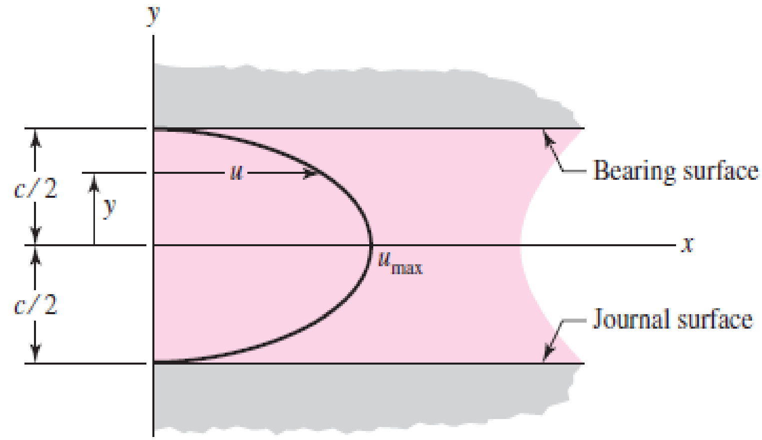


Fig.12.29 Parabolic distribution of the lubricant velocity.

To consider eccentricity, as shown in Fig. 12–30, the film thickness is $h = c - e\cos\theta$. Substituting *h* for *c* in Eq. (*i*), with the average ordinate of a parabola being two-thirds the maximum, the average velocity at any angular position *θ* is

$$u_{ave} = \frac{2}{3}\frac{p_s h^2}{8\mu l'} = \frac{p_s}{12\mu l'}\left(c - e\cos\theta\right)^2 \qquad (j)$$

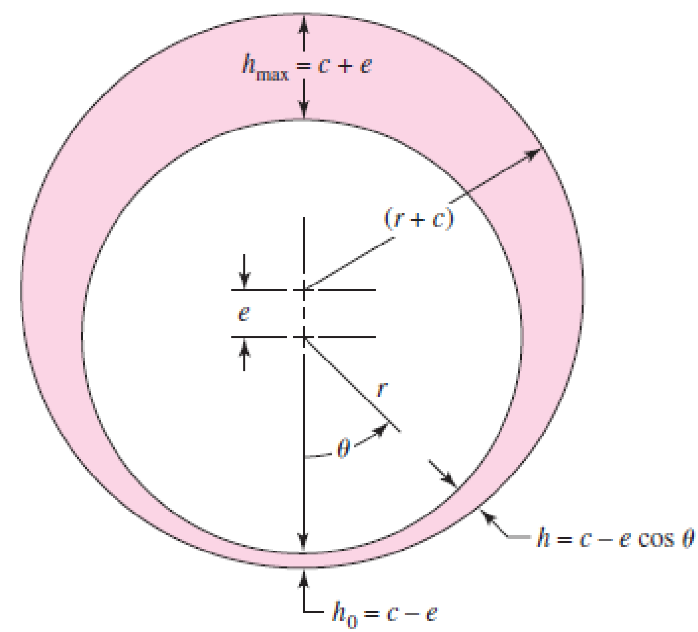


Fig. 12.30

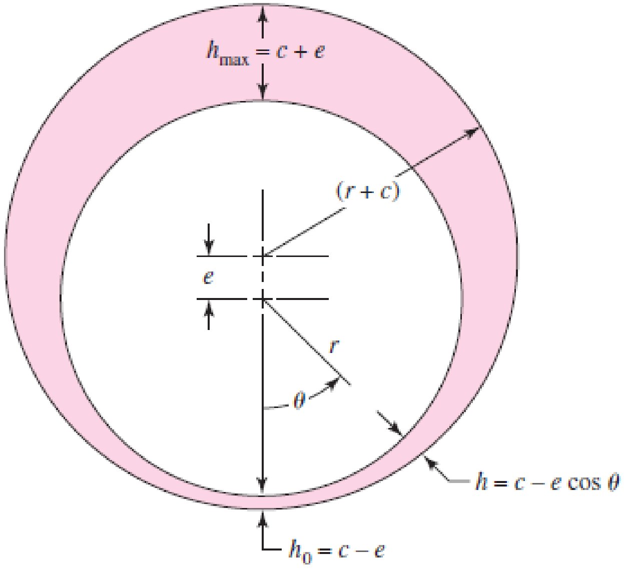


Fig. 12.30

$$h_o = c(1-\varepsilon)$$

$$\varepsilon = \frac{e}{c}$$

We can compute the amount of lubricant that flows out both ends;

$$dQ_s = u_{ave} \times dA \qquad dA = h \times r d\theta$$

Total side flow is twice that value and *dA* is the elemental area;

$$dQ_s = 2 \times u_{ave} \times h r d\theta$$

$$u_{ave} = \frac{p_s}{12\mu l'}(c - e\cos\theta)^2$$

$$dQ_s = \frac{p_s r}{6\mu l'}(c - e\cos\theta)^3 d\theta$$

Integrating around the bearing gives the total side flow as

$$Q_s = \int dQ_s = \frac{p_s r}{6\mu l'} \int_0^{2\pi} (c - e\cos\theta)^3 d\theta = \frac{p_s r}{6\mu l'} (2\pi c^3 - 3\pi c^3)$$

Rearranging, with *ε* = *e/c*, gives

$$Q_s = \frac{\pi p_s r c^3}{3\mu l'} (1 + 1.5\varepsilon^2)$$

<u>Flow rate equation for pressure-fed bearings</u>

The charts for flow variable *(Q/rcNl)* and flow ratio *(Q_S/Q)* (Figs. 12–19 and 12–20) do not apply to pressure-fed bearings.

In analyzing the performance of pressure-fed bearings, the bearing length should be taken as l'.

The characteristic pressure in each of the two bearings that constitute the pressure-fed bearing assembly P is given by

$$P = \frac{W/2}{2rl'} = \frac{W}{4rl'}$$

Also, the maximum film pressure P_{max} given by Fig. 12–21 must be increased by the oil supply pressure p_s to obtain the total film pressure.

$$P_{max} = P_{max} + P_s$$

Since the oil flow has been increased by forced feed, Eq. (12–14) will not give a correct temperature rise (will give too high temperature rise) because the side flow carries away all the heat generated. A new temperature rise equation (ΔT) is derived for pressure fed bearings.

To calculate heat gain of the fluid passing through the bearing the following steps are followed:

1) $$\frac{l'}{d} = \frac{\left(l - l_g\right)/2}{d}$$

2) $$P = \frac{W/2}{2rl'} = \frac{W}{4rl'}$$

The Sommerfeld number may be expressed as

3) $$S = \left(\frac{r}{c}\right)^2 \frac{\mu N}{P} = \left(\frac{r}{c}\right)^2 \frac{\mu N}{\frac{W}{4rl'}} = \left(\frac{r}{c}\right)^2 \frac{4rl'\mu N}{W}$$

4) The corresponding temperature rise (ΔT) equation in SI units uses the bearing load *W in kN*, lubricant supply pressure p_s in *kPa*, and the journal radius *r* in mm:

$$\Delta T_{^oC} = \frac{978 \times 10^6}{1 + 1.5\varepsilon^2} \times \frac{\left(\frac{r}{c} f\right) S W^2}{p_s r^4}$$

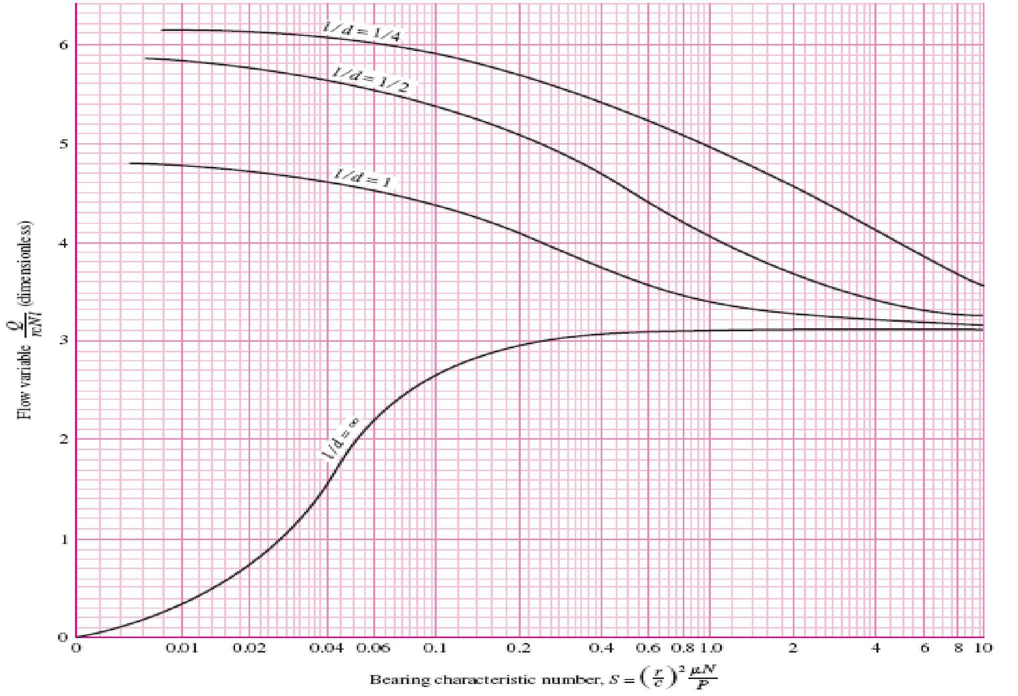
Where ΔT in ^{o}C (temperature increase)
W= bearing load in kN
p_s= supply pressure in kPa
r= journal radius in mm

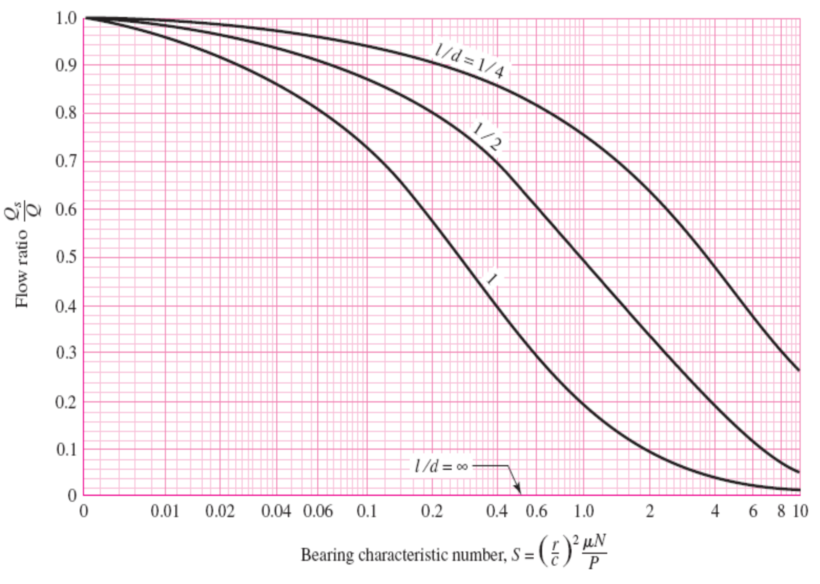
$$\Delta T_{^oC} = \frac{1.5\left(\frac{r}{c} f\right)}{\gamma C_H \left(1 + 1.5\varepsilon^2\right)} \times \frac{S W^2}{p_s r^4}$$

Where γ= 861 kg/m^3
C_H= 1760 J/kg ^{o}C
W= bearing load in N
p_s= supply pressure in Pa
r= journal radius in m

5) Do not use figures 12.18 (17) of flow rate variable

and 12.19 (18) flow rate





Instead use the equation

$$Q_s = \frac{\pi p_s r c^3}{3\mu l'}(1+1.5\varepsilon^2)$$

6) Use figures 12.20 (40) but to find maximum pressure add two values P_{max} *and* p_s .

$$P_{\max act} = P_{\max} + p_s$$

EXAMPLE 4.3 (12-20) (prob. 12.16)

An 8- cylinder diesel engine has a front main bearing 90 mm in diameter and 51 mm long. The bearing has a central annular oil groove 6 mm wide.
It is pressure lubricated with SAE 30 oil at on inlet temperature of 82 ^oC and at a supply pressure of 345 kPa.
Corresponding to a radial clearance of 0.064 mm, a speed of 2800 rpm and a radial load of 20500 N, find;

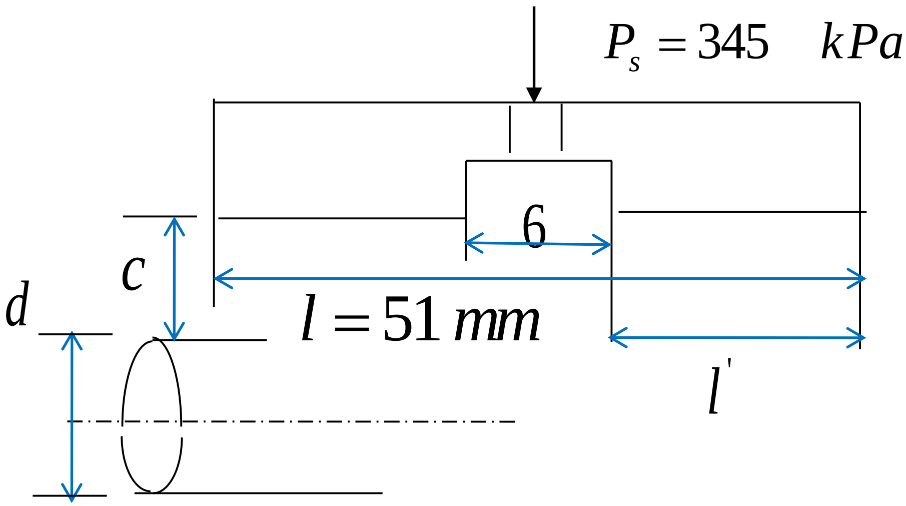
a) The temperature rise

b) Minimum oil film thickness

c) maximum oil film pressure

d) oil flow rate

e) θ_{h_0} *and* $\theta_{P_{max}}$



$$l = 51\ mm$$

$$d = 90\ mm \rightarrow \quad r = 45\ mm$$

$$l_g = 6\ mm \rightarrow l' = \frac{1}{2}\left(l - l_g\right) = 22.5\ mm$$

$$\frac{l'}{d} = \frac{22.5}{90} = \frac{1}{4}$$

$T_i = 82\ ^oC$

$T_o = ?$ $\quad \rightarrow \mu$ at T_{ave} Since ΔT and μ are unknowns we need to use trail and error method

$P_s = 345\ kPa$

$c = 0.064\ mm$

$r = 45\ mm$

$N = 2800\ rpm \rightarrow N = 46.7\ rev/\sec$

$W = 20500\ N$

$$P = \frac{W}{4rl'} = \frac{20500}{4 \times 45 \times 22.5} = 5.06\ \ MPa$$

$(see\ pp.558\ \ Table\ 12.2,\ 6 - 12\ \ MPa)$

Table 12-3 RECOMMENDED UNIT LOADS FOR SLEEVE BEARINGS

Application	Unit load, kPa	Application	Unit load, kPa
Air compressors:		Diesel engines:	
Main bearings	1 000– 2 000	Main bearings	6 000–12 000
Crankpin	2 000– 4 000	Crankpin	8 000–15 000
Automotive engines:		Wristpin	14 000–15 000
Main bearings	4 000– 5 000	Electric motors	800– 1 500
Crankpin	10 000–15 000	Gear reducers	800– 1 500
Centrifugal pumps	600– 1 200	Steam turbines	800– 1 500

a)

$$\Delta T_{o_C} = \frac{978\times10^6}{1+1.5\varepsilon^2}\times\frac{\left(\frac{r}{c}f\right)SW^2}{p_s r^4}$$

$$\text{but } S = \left(\frac{r}{c}\right)^2\frac{\mu N}{P} \quad \text{depends on } \mu$$

$$\text{also } \mu \quad \text{depends on } T_i + \frac{\Delta T}{2}$$

$$\text{where } \varepsilon \;\&\; \frac{r}{c}f \quad \text{depends on } S \;\&\; \frac{l}{d}$$

Start by assuming ΔT= 20 ^{o}C (1)

$$T_{ave} = T_i + \frac{\Delta T}{2} = 82 + \frac{20}{2} = 92\ {}^oC \rightarrow \mu = 8.5\times10^{-3}\,Pa-\sec$$

$$\text{Then}\ \ S = \left(\frac{45}{0.064}\right)^2 \frac{8.5\times10^{-3}\times46.7}{5.06\times10^6} = 0.038$$

$$\text{For}\ \ S = 0.038 \qquad \frac{h_0}{c} = 0.075 \quad \& \quad \varepsilon = 0.925 \quad (\text{Fig.}12.14\ \text{or}\ 12.16)$$

$$\frac{l'}{d} = \frac{1}{4} \qquad \frac{r}{c}f = 2.1 \quad (\text{Fig.}12.17\ \text{or}\ 12.18)$$

$$\Delta T_{o_C} = \frac{978\times10^6}{1+1.5\times(0.925)^2}\times\frac{2.1\times0.038\times(20.5)^2}{345\times(45)^4} = 10.15\ {}^oC < 20\ {}^oC$$

$$\Delta T_{o_C} = \frac{978\times10^6}{1+1.5\varepsilon^2}\times\frac{\left(\frac{r}{c}f\right)SW^2}{p_s r^4}$$

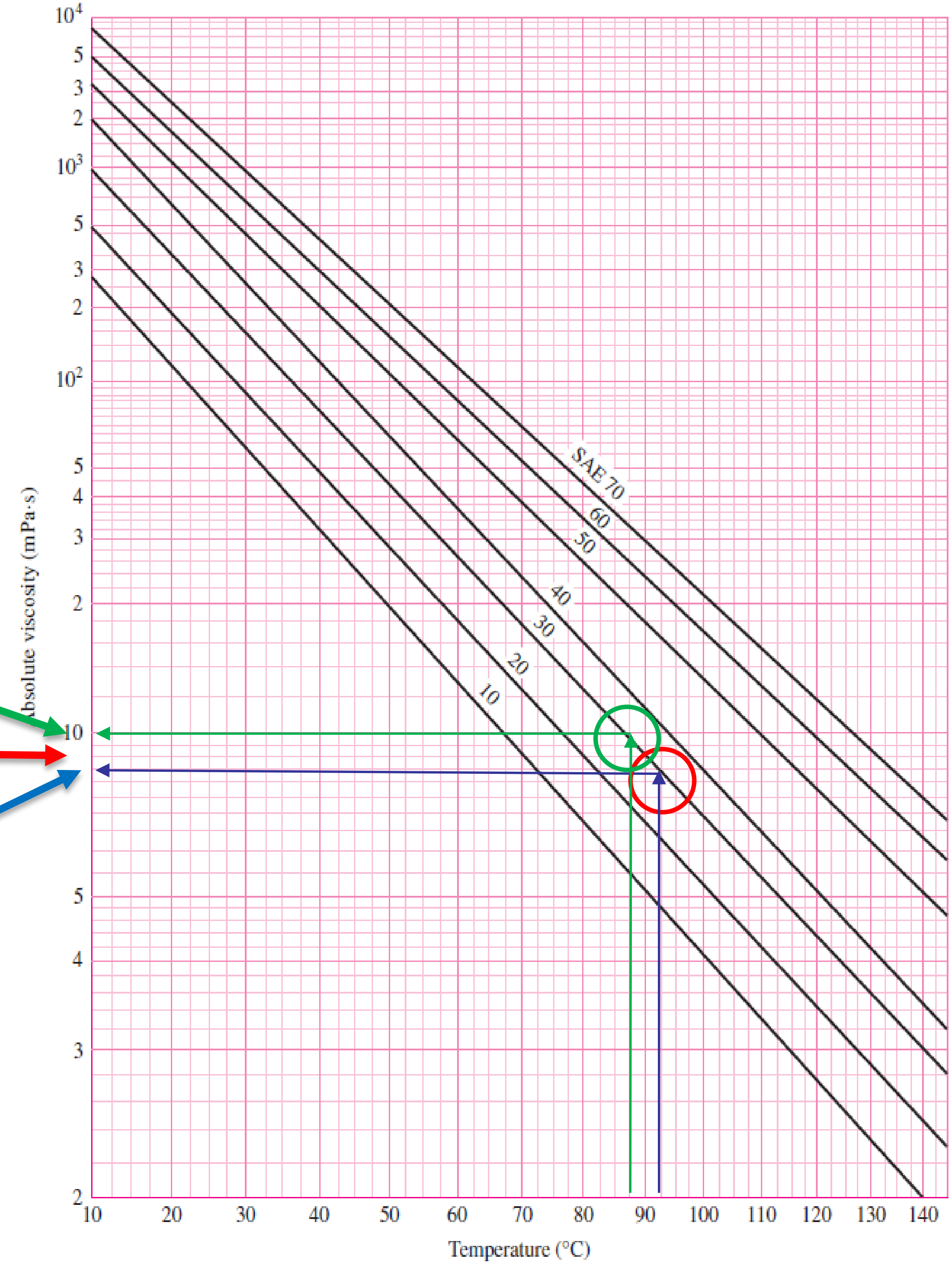
Now from chart Fig. 12.12 pp. 534 (Fig. 12.10 pp. 435)

Lubrication Oil is
SAE 30

$$(2)\quad T_{ave} = 87\ ^{o}C$$
$$\rightarrow \mu = 10\times10^{-3}\,Pa-\text{sec}$$

$$(3)\quad T_{ave} = 87.5\ ^{o}C$$
$$\rightarrow \mu = 9.5\times10^{-3}\,Pa-\text{sec}$$

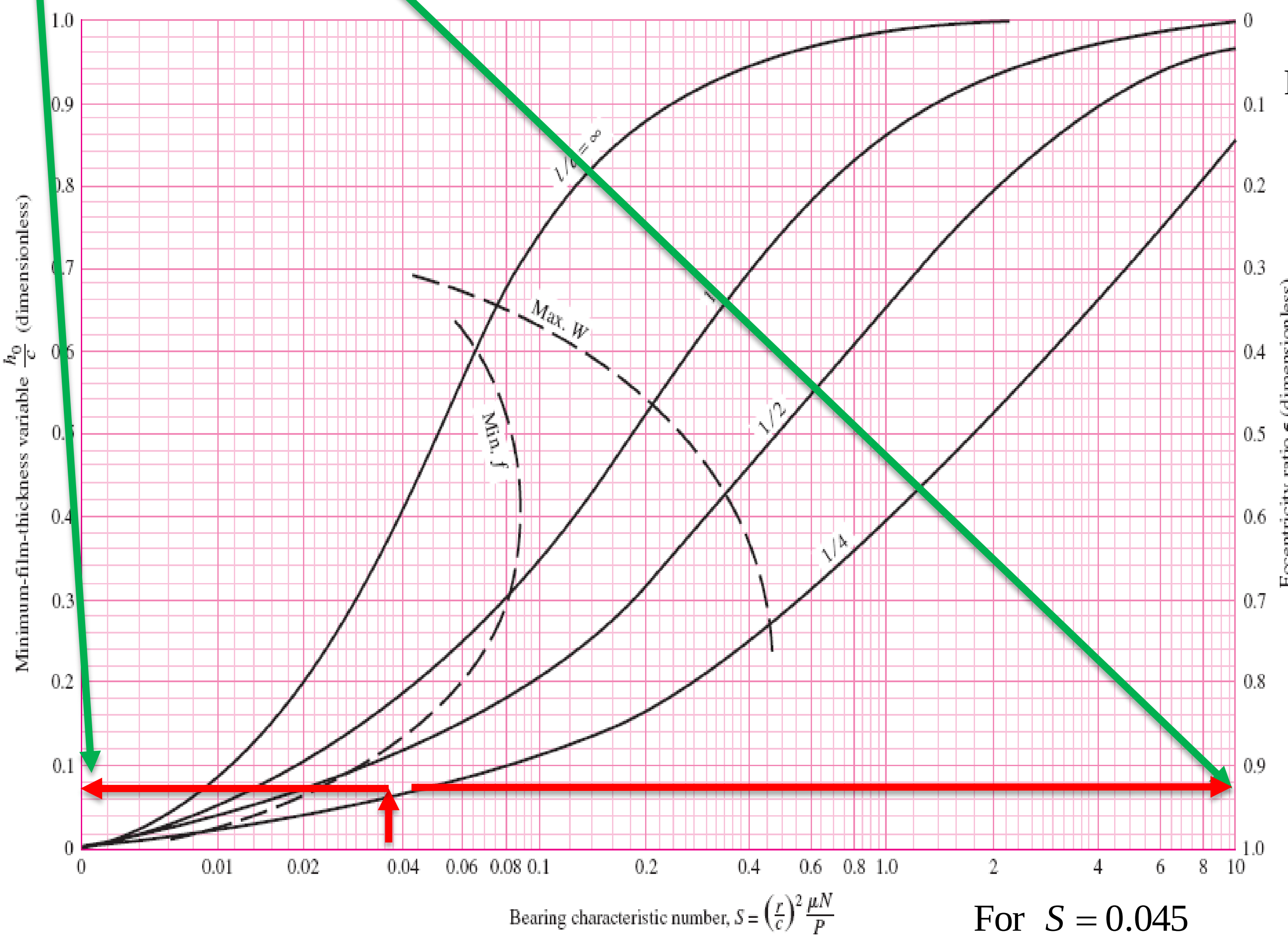
$$(1)\quad T_{ave} = 92\ ^{o}C$$
$$\rightarrow \mu = 8.5\times10^{-3}\,Pa-\text{sec}$$



For $S = 0.038$

$\frac{l'}{d} = \frac{1}{4}$

(1) $\frac{h_0}{c} = 0.075$ & $\varepsilon = 0.925$ *(p543)(p446) (Fig.12.14 or 12.16)*



For $S = 0.042$

$\frac{l'}{d} = \frac{1}{4}$

$\frac{h_0}{c} = 0.075$

$\varepsilon = 0.925$

$\frac{h_0}{c} = 0.08$

$\varepsilon = 0.92$

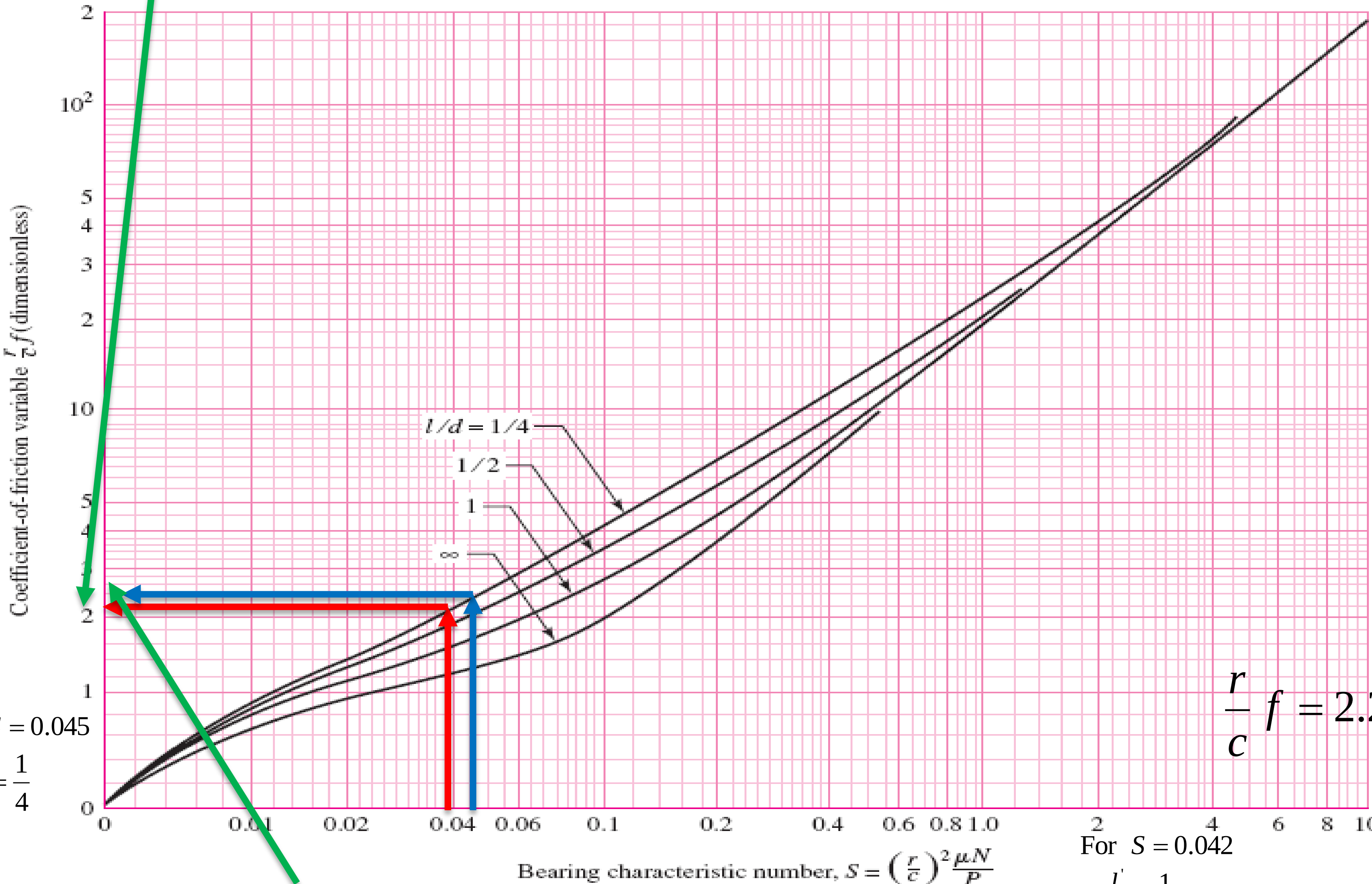
For $S = 0.045$

$\frac{l'}{d} = \frac{1}{4}$

(1) $\frac{r}{c} f = 2.1$ *(Fig.12.17 or 12.18)(p540)(p444)*

For $S = 0.038$

$\frac{l'}{d} = \frac{1}{4}$



For $S = 0.045$

$\frac{l'}{d} = \frac{1}{4}$

$\frac{r}{c} f = 2.2$

For $S = 0.042$

$\frac{l'}{d} = \frac{1}{4}$

$\frac{r}{c} f = 2.3$

Re-try ΔT= 10^{o}C (2)

$$T_{ave} = T_i + \frac{\Delta T}{2} = 82 + 5 = 87\ ^oC \rightarrow \mu = 10 \times 10^{-3}\, Pa - \sec$$

$$\text{Then } S = \left(\frac{45}{0.064}\right)^2 \frac{10 \times 10^{-3} \times 46.7}{5.06 \times 10^6} = 0.045$$

$$\text{For } S = 0.045 \qquad \frac{h_0}{c} = 0.08 \quad \& \quad \varepsilon = 0.92 \qquad (\text{Fig.}12.16)$$

$$\frac{l'}{d} = \frac{1}{4} \qquad \frac{r}{c} f = 2.3 \quad (\text{Fig.}12.18)$$

$$\Delta T_{^oC} = \frac{978 \times 10^6}{1 + 1.5 \times (0.92)^2} \times \frac{2.3 \times 0.045 \times (20.5)^2}{345 \times (45)^4} = 13.25\ ^oC > 10\ ^oC$$

$$\Delta T_{^oC} = \frac{978 \times 10^6}{1 + 1.5\varepsilon^2} \times \frac{\left(\frac{r}{c} f\right) S W^2}{p_s r^4}$$

Re-try ΔT= 11^{o}C; (3)

$$T_{ave} = T_i + \frac{\Delta T}{2} = 82 + 5.5 = 87.5\ {}^oC \rightarrow \mu = 9.5\times 10^{-3} \quad Pa-\sec$$

$$\text{Then}\ \ S = \left(\frac{45}{0.064}\right)^2 \frac{9.5\times 10^{-3}\times 46.7}{5.06\times 10^6} = 0.042$$

$$\text{For}\ \ S = 0.042 \qquad \frac{h_0}{c} = 0.075 \quad \& \quad \varepsilon = 0.925 \quad (\text{Fig.}12.16)$$

$$\frac{l'}{d} = \frac{1}{4}$$

$$\frac{r}{c} f = 2.2 \quad (\text{Fig.}12.18)$$

$$\Delta T_{o_C} = \frac{978\times 10^6}{1+1.5\times(0.925)^2}\times\frac{2.2\times 0.042\times(20.5)^2}{345\times(45)^4} = 11.75\ {}^oC \cong 11\ {}^oC$$

$$\Delta T_{o_C} = 11.75\ {}^oC \cong 11\ {}^oC$$

OK…

$$\Delta T_{o_C} = \frac{978\times 10^6}{1+1.5\varepsilon^2}\times\frac{\left(\frac{r}{c}f\right)SW^2}{p_s r^4}$$

$$b) \quad h_0 = 0.075 \times c = 0.075 \times 0.064$$

$$h_0 = 4.8 \times 10^{-3}\ mm = 4.8\ micron$$

$$c) \quad P_{\max} = ? \quad P_{\max} + P_s \quad \text{from Fig.12.21 for S} = 0.042 \rightarrow \frac{P}{P_{\max}} = 0.15$$

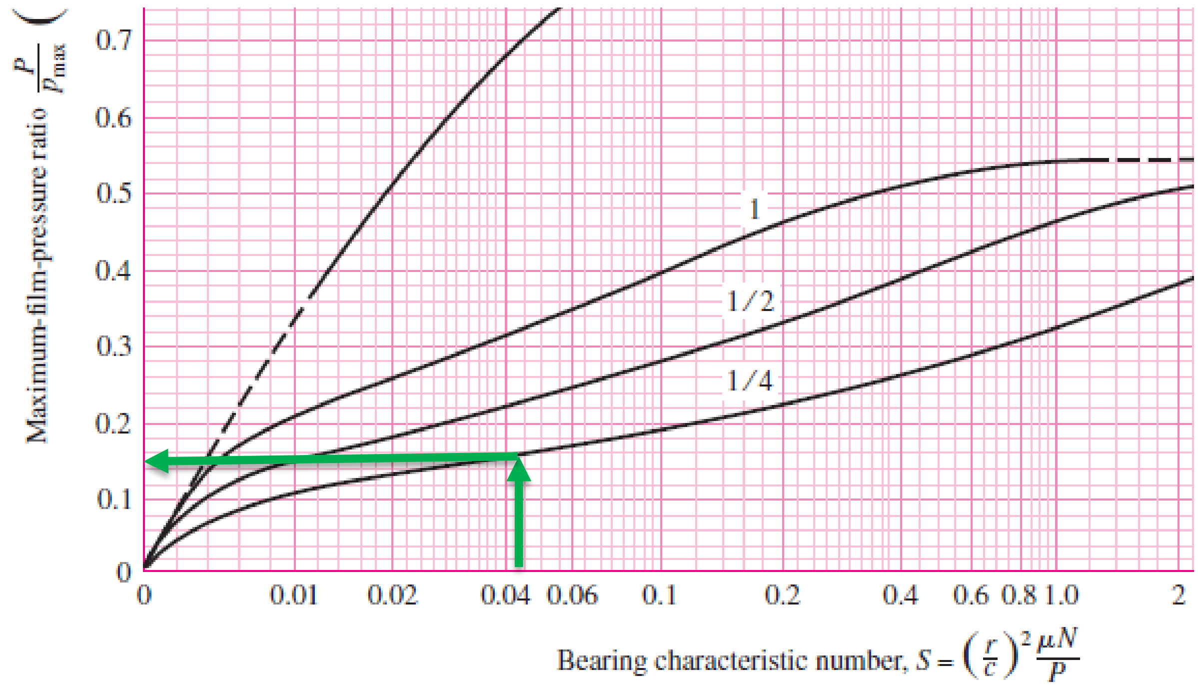
$$So, \quad P_{\max} = \frac{P}{0.15} = \frac{5.06}{0.15} = 33.73\ \ MPa$$

$$P_{\max_{act}} = 33.73 \times 10^6 + 345 \times 10^3 = 34075\ \ kPa = 34.075\ MPa$$

$$P_{\max\, act} = P_{\max} + p_s$$

For $S = 0.042$

$$\frac{l'}{d} = \frac{1}{4}$$



d) Q_s =?

$$Q_s = \frac{\pi P_s r c^3}{3\mu l'}(1+1.5\varepsilon^2) = \frac{\pi \times 345\times10^3 \times 45 \times (0.064)^3}{3\times 9.5\times10^{-3}\times 22.5}\left(1+1.5\times(0.925)^2\right)$$

$$Q_s = 45528\ mm^3/\sec$$

$$Q_s = 0.045\ lt/\sec = 2.73\ lt/\min$$

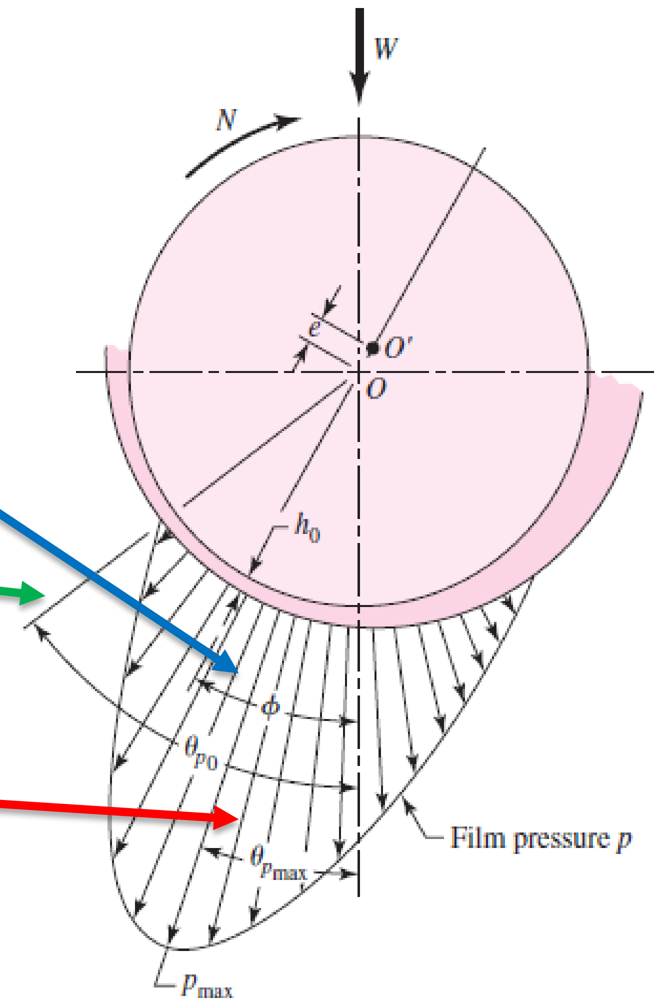
e) For $\theta_{h_0}, \theta_{P_0}$ & $\theta_{P_{\max}}$

use related Figures

$$\theta_{h_0} = 18^o$$

$$\theta_{P_0} = 24^o$$

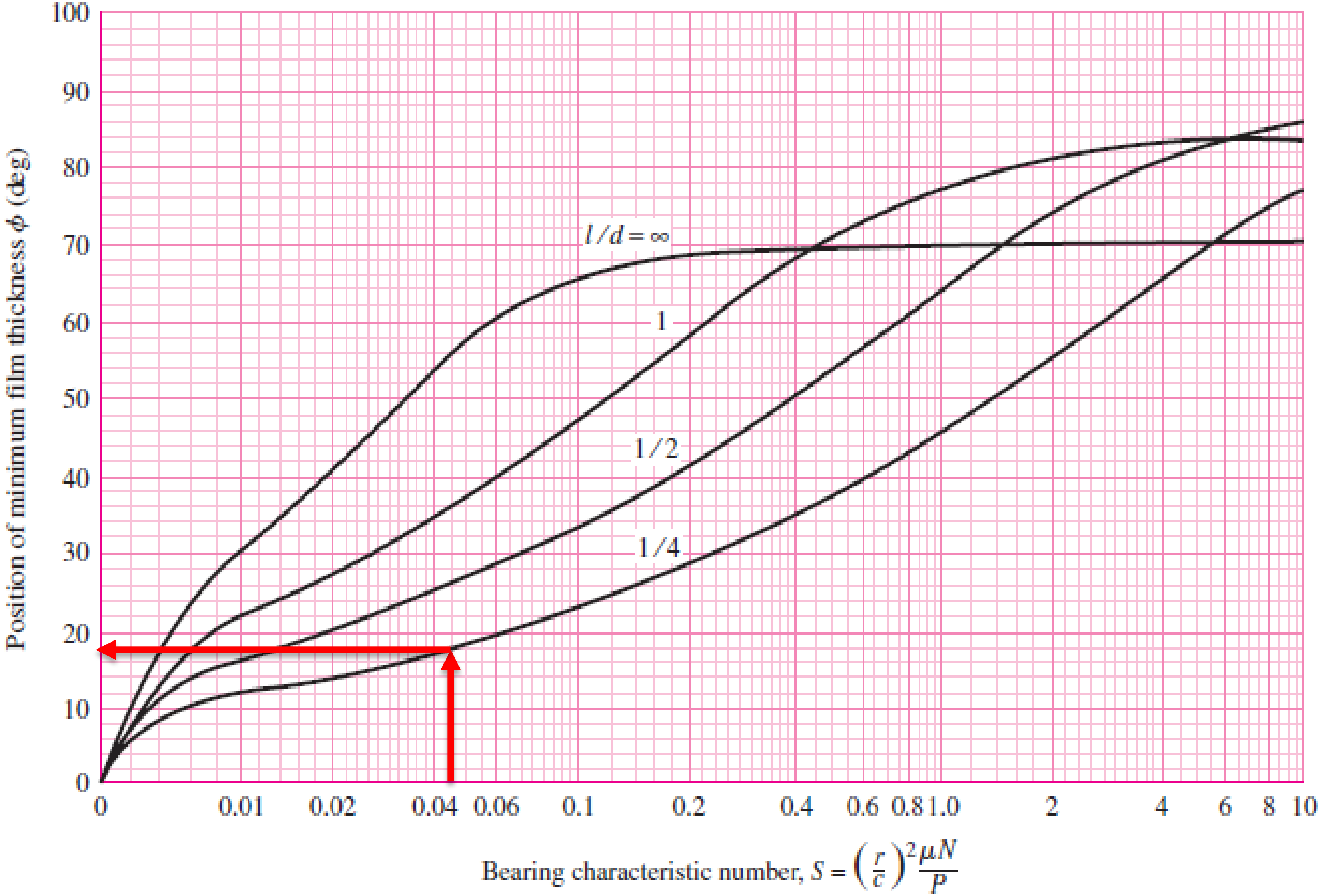
$$\theta_{P_{\max}} = 7.5^o$$



e) For θ_{h0} *use the Fig. 12.17* For $S = 0.042$

$$\theta_{h_0} = 18^o$$

$$\frac{l^{'}}{d} = \frac{1}{4}$$

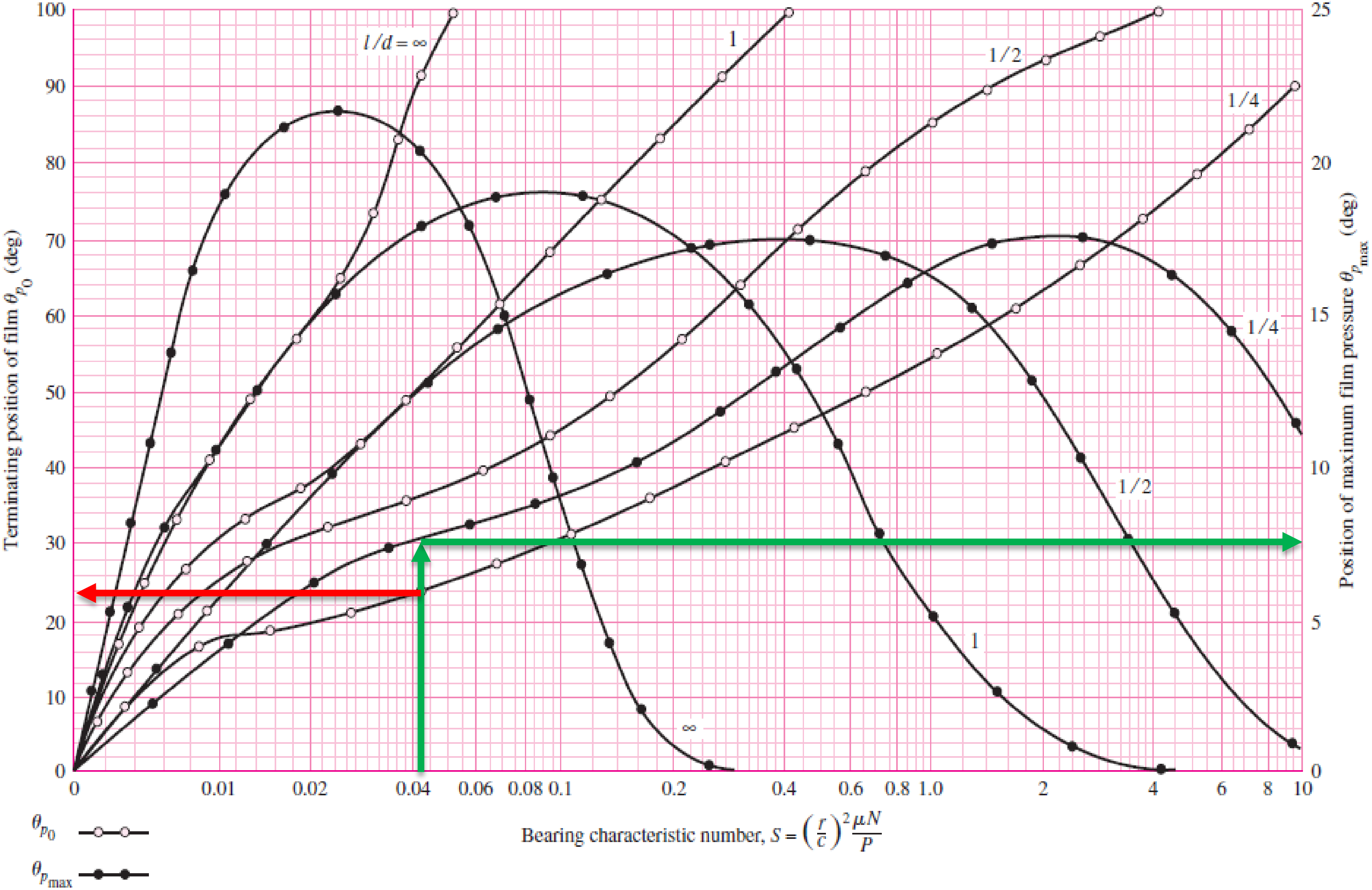


For θ_{P_0} & $\theta_{P_{max}}$ *use the Fig. 12.22*

For $S = 0.042$

$\frac{l'}{d} = \frac{1}{4}$

$\theta_{P_0} = 24^o$ $\theta_{P_{max}} = 7.5^o$



BEARINGS