Boundary & mixed lubricated sleeve bearings

- bearings That run with little or no lubrication
- usually no hydrodynamic film (wedge)... some greased or porous (bronze) bearing can have hydrodynamic action

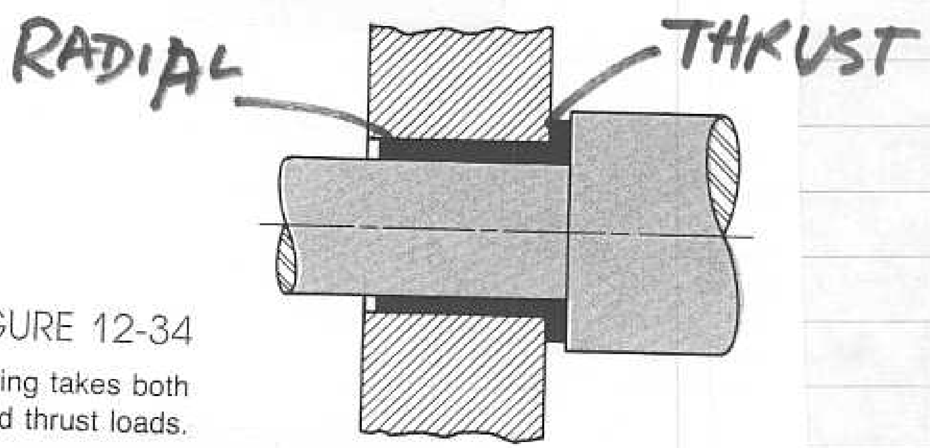


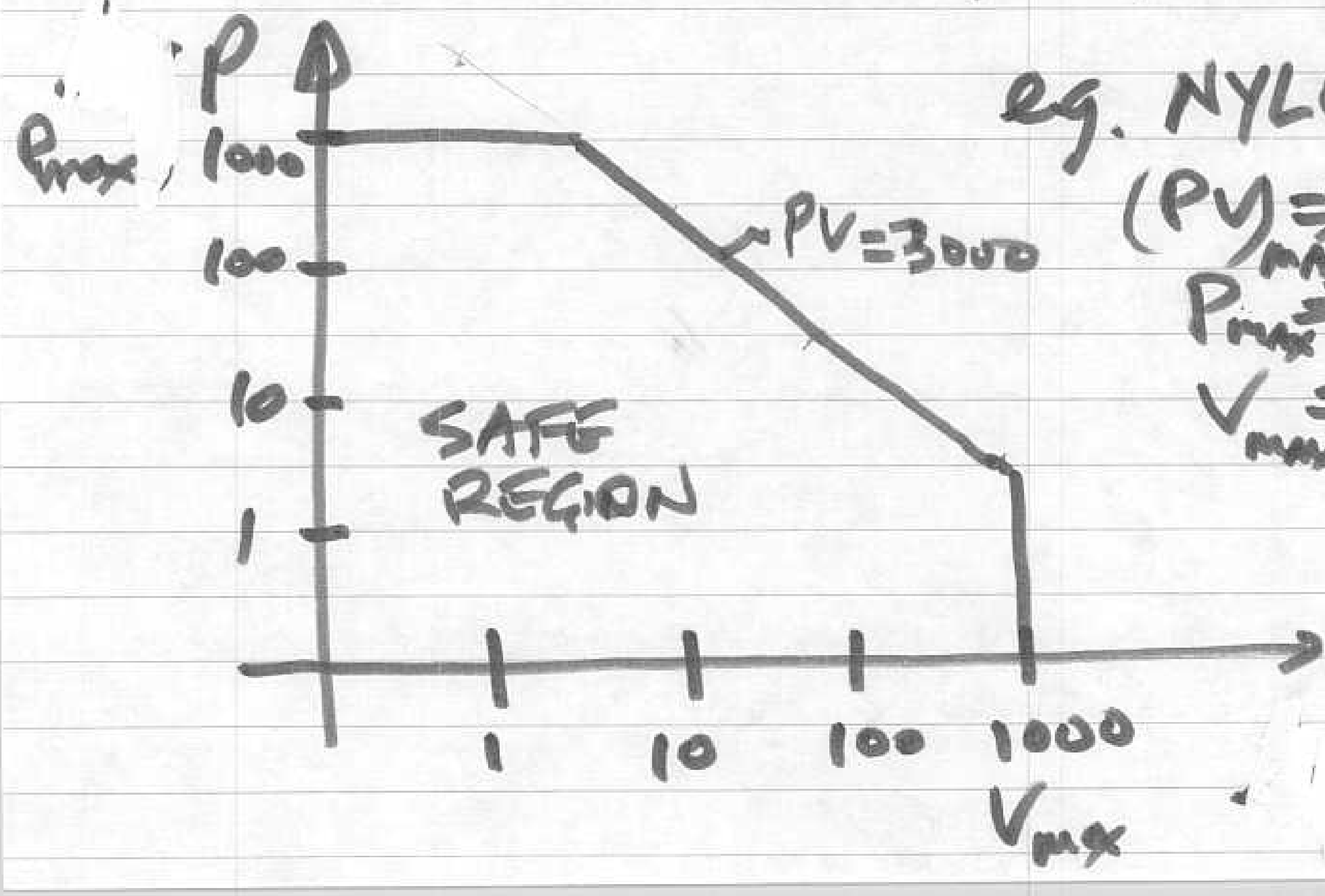
FIGURE 12-34
Flanged sleeve bearing takes both radial and thrust loads.

MAX P | MAX T | MAX SPEED | MAX PV

MATERIAL	MAXIMUM LOAD, psi	MAXIMUM TEMPERATURE, °F	MAXIMUM SPEED, fpm	MAXIMUM *PV* VALUE*
Cast bronze	4 500	325	1 500	50 000
Porous bronze	4 500	150	1 500	50 000
Porous iron	8 000	150	800	50 000
Phenolics	6 000	200	2 500	15 000
Nylon	1 000	200	1 000	3 000
Teflon	500	500	100	1 000
Reinforced Teflon	2 500	500	1 000	10 000
Teflon fabric	60 000	500	50	25 000
Delrin	1 000	180	1 000	3 000
Carbon-graphite	600	750	2 500	15 000
Rubber	50	150	4 000	
Wood	2 000	150	2 000	15 000

*P = load, psi; V = speed, fpm.

$\frac{W}{DL}$ $P \Rightarrow$ psi $V \Rightarrow$ feet/sec



eg. NYLON

$(PV)_{max} \Rightarrow 3000$

$P_{max} \Rightarrow 1000$

$V_{max} \Rightarrow 1000$

$P_{max} \rightarrow$ static load

$V_{max} \Rightarrow$ softening, melting

$PV_{max} \Rightarrow$ related to heat generation.

$$\underbrace{\dot{h}}_{\frac{\text{heat}}{\text{area}}} = \underbrace{f_m}_{\substack{\text{friction coeff.} \\ \simeq 0.1}} \underbrace{P}_{\text{oper. press}} \underbrace{V}_{\text{speed}}$$

Need model to calc actual operating temp if needed e.g. eq 12-30

Assignment #8

Due 4/18

12-1

12-7

12-12* (iteration req'd)

12-17

11-2

11-6

* is this a good/acceptable design based on the criteria discussed in class ... see notes

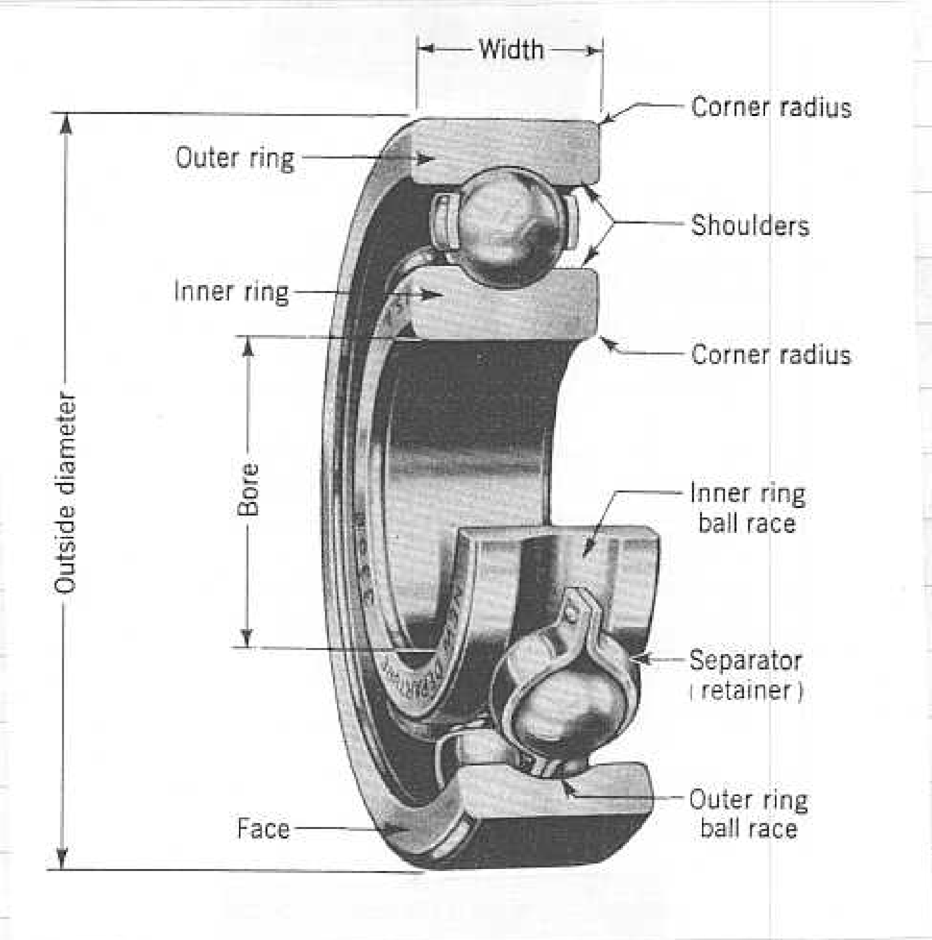
Ball and Roller Bearings

A.K.A. "anti-friction bearings"

(AFBMA) ... industry assoc.

1. GEOMETRY + INTRO
2. LUBRICATION -- Elastohydrodynamic --
   I viscosity ↑↑
   II elastic def. (like Hertz)
3. FAILURE -- fatigue, surface-initiated, due to fluctuating <u>contact</u> stresses.

TERMINOLOGY



- Low friction .... esp at low (zero) speed
- Similar load capacity to hydrodynamic bearings @ moderate speeds (few 1000 rpm)
- Room needed, cost, etc, more than j-b

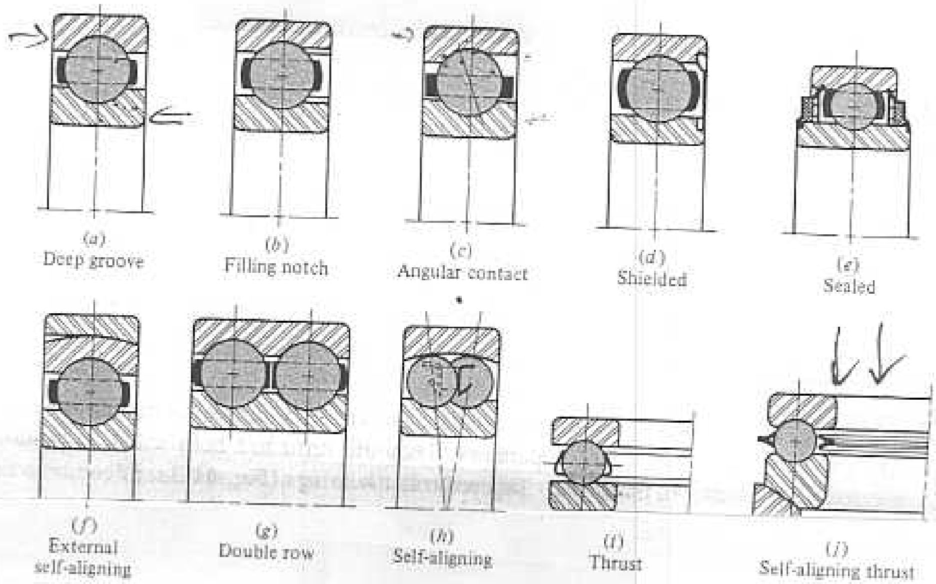


FIGURE 11-2

Various types of ball bearings.

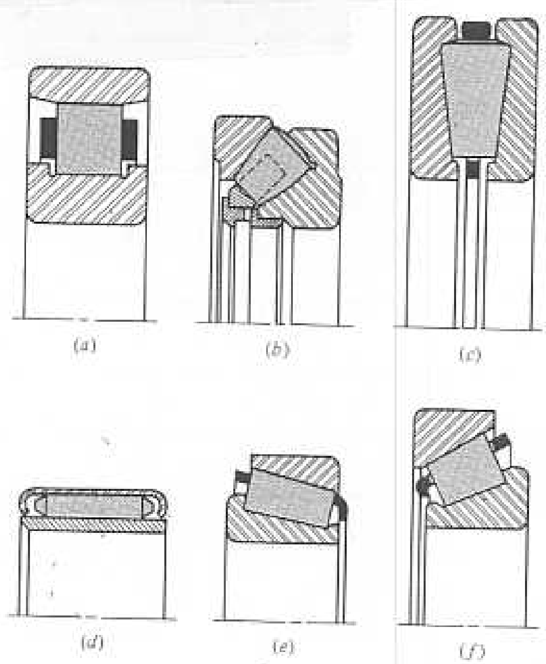


FIGURE 11-3

Types of roller bearings: (a) straight roller; (b) spherical roller thrust; (c) tapered roller thrust; (d) needle; (e) tapered roller; (f) steep-angle tapered roller. *(Courtesy of The Timken Company.)*

- Lubrication (very thin films $\approx 0.1 \rightarrow 1^{+}$ μm) keeps surfaces apart ... no metal to metal contact... FATIGUE FAILURE (see illust)

Fatigue behavior

$$\frac{L_1}{L_2} = \left(\frac{F_2}{F_1}\right)^a$$

$a = 3 \Rightarrow$ ball bearing $\frac{10}{3} \Rightarrow$ roller bearings

Use L_{10} life .... 10% of bearings fail under given load

L measured in $\overbrace{\text{cycles}}^{\text{millions of}}$ or hours

FATIGUE FAILURES

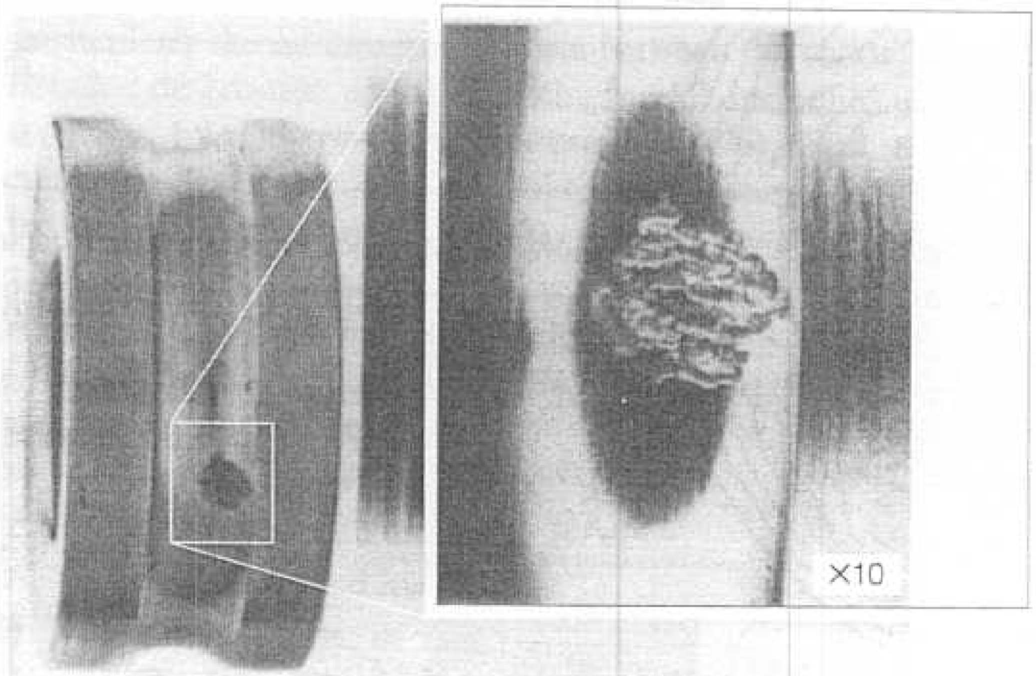


Figure 13.20 Typical fatigue spall. [B. J. Hamrock and W. J. Anderson, *Rolling-Element Bearings,* NASA Reference Publication 1105, 1983.]

- Both races & balls/rollers see "0" (out of contact) to max Hertzian stress (within contact) during operation.
- Most failures (past couple of decades) are surface initiated due to low internal defects in bearing steels

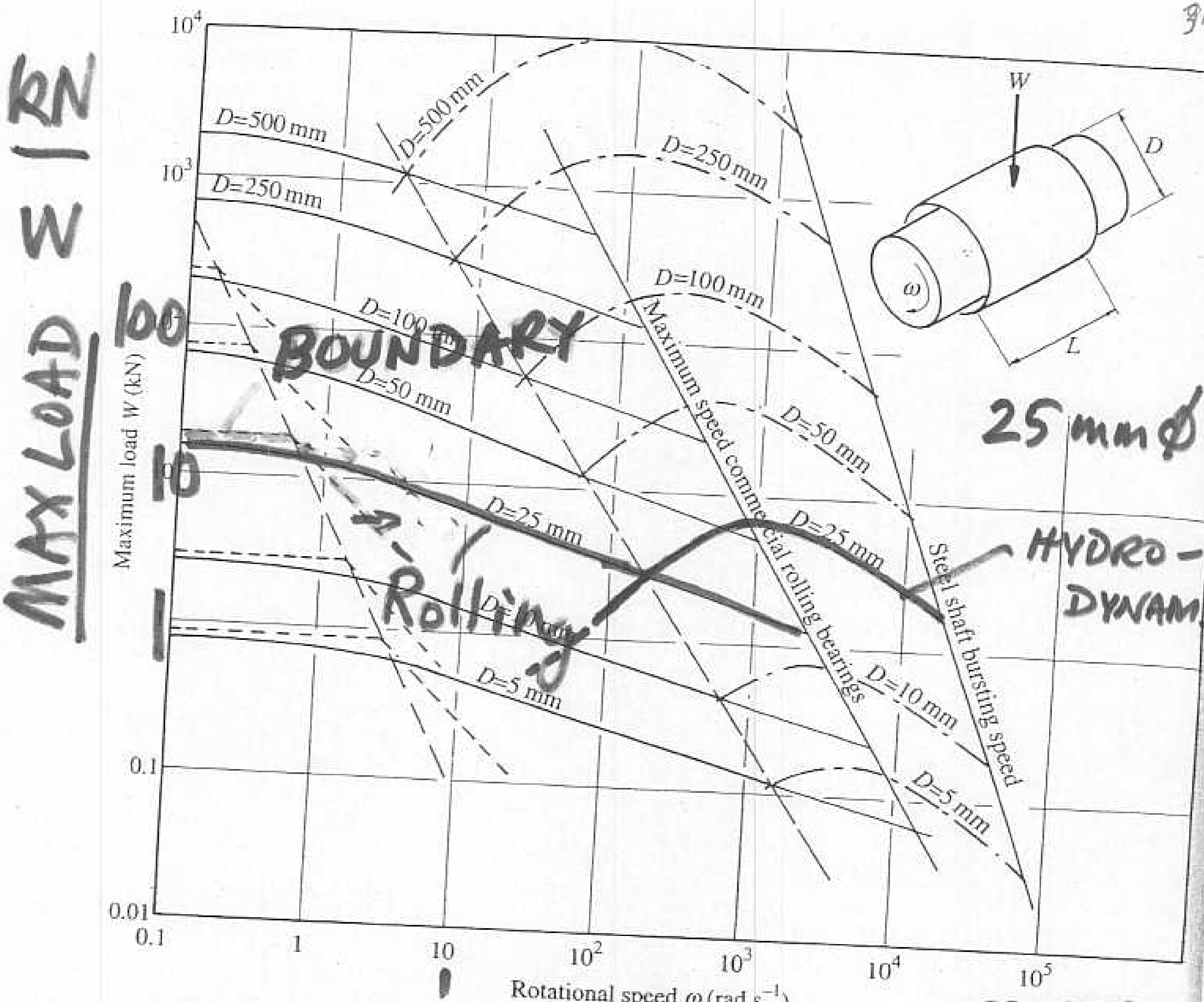


Fig. 1.4 Selection of journal bearings for continuous rotation (from Neale 1973): dry and marginally lubricated bearings (– – – – – – –); fluid film bearings (— – — – –); rolling-element bearings (————————).

100 1000 10000 rpm

Example of application ranges of different bearing types

BASIC LOAD RATING, C

Load for 10^6 revs of inner ring

$$L = \left(\frac{C}{F}\right)^a \quad \text{or} \quad C = F L^{1/a}$$

Millions of cycles

radial load corresponding to L

C is provided by manufacturer

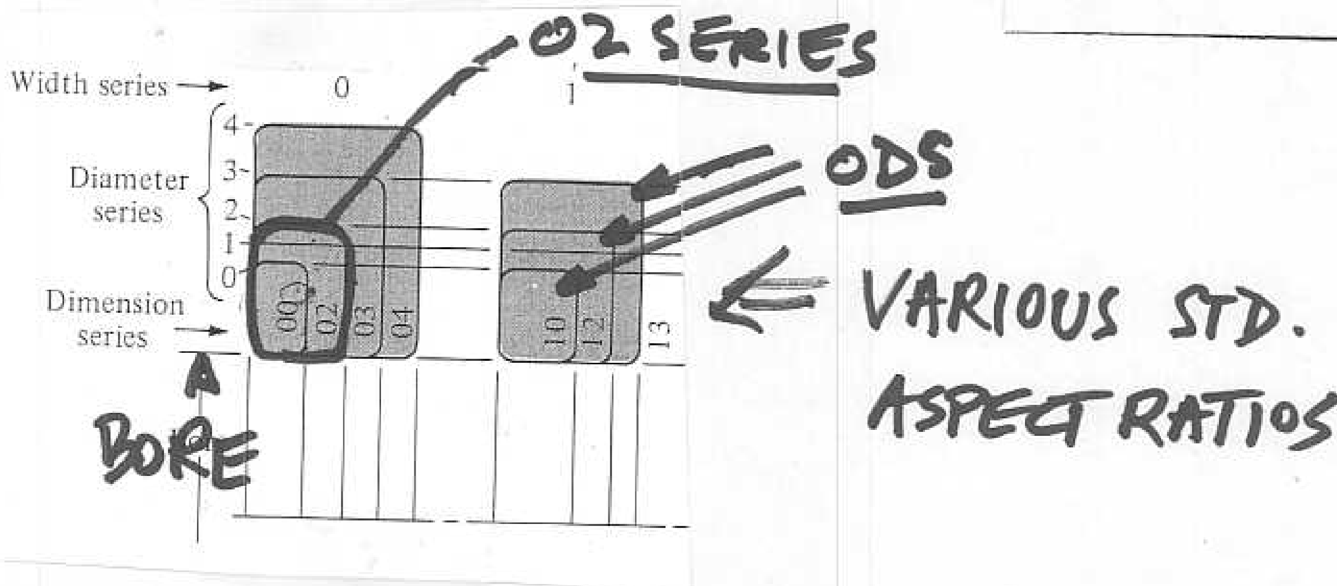
... based on testing + more + more complex computer predictions

.... bearings are assemblies ...
Not individual contacts

TABLE 11-3 02 SERIES

BORE mm	OD mm	WIDTH mm	LOAD RATINGS kN C	C_0
10	30	9	5.07	2.24
12	32	10	6.89	3.10
15	35	11	7.80	3.55
17	40	12	9.56	4.50
20	47	14	12.7	6.20
25	52	15	14.0	6.95
30	62	16	19.5	10.0
35	72	17	25.5	13.7
40	80	18	30.7	16.6
45	85	19	33.2	18.6
50	90	20	35.1	19.6
55	100	21	43.6	25.0
60	110	22	47.5	28.0
65	120	23	55.9	34.0
70	125	24	61.8	37.5
75	130	25	66.3	40.5
80	140	26	70.2	45.0
85	150	28	83.2	53.0
90	160	30	95.6	62.0
95	170	32	108	69.5

BASIC LOADRATING C kN
STATIC LOAD RATING C_0 kN



Effective radial load, ball bearings, F_e

F_e is The larger of

$$F_e = VF_r$$

OR

$$F_e = XVF_r + YF_a$$

$V \Rightarrow$ rotation factor
= 1 for <u>inner</u> ring rotation
= 1.2 for outer ring rotation

$F_r \Rightarrow$ radial load

$F_a \Rightarrow$ axial load

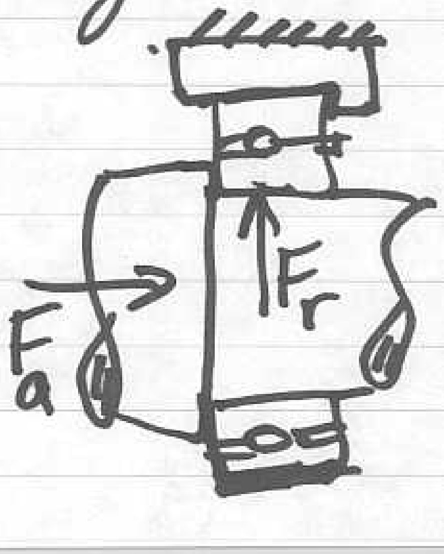


TABLE 11-2
Equivalent Radial-Load Factors for Ball Bearings

F_a/C_0	e	$F_a/F_r \le e$		$F_a/F_r > e$	
		X_1	Y_1	X_2	Y_2
0.014*	0.19	1.00	0	0.56	2.30
0.021	0.21	1.00	0	0.56	2.15
0.028	0.22	1.00	0	0.56	1.99
0.042	0.24	1.00	0	0.56	1.85
0.056	0.26	1.00	0	0.56	1.71
0.070	0.27	1.00	0	0.56	1.63
0.084	0.28	1.00	0	0.56	1.55
0.110	0.30	1.00	0	0.56	1.45
0.17	0.34	1.00	0	0.56	1.31
0.28	0.38	1.00	0	0.56	1.15
0.42	0.42	1.00	0	0.56	1.04
0.56	0.44	1.00	0	0.56	1.00

*Use 0.014 if $F_a/C_0 < 0.014$.

① Determine $\dfrac{F_a}{C_0}$

② Check for $\dfrac{F_a}{F_r} \le e$ or $\dfrac{F_a}{F_r} < e$

③ Use appropriate X, Y to get equivalent load.

TABLE 11-6
Load-Application Factors

TYPE OF APPLICATION	LOAD FACTOR
Precision gearing	1.0–1.1
Commercial gearing	1.1–1.3
Applications with poor bearing seals	1.2
Machinery with no impact	1.0–1.2
Machinery with light impact	1.2–1.5
Machinery with moderate impact	1.5–3.0

Load application factors

... increase (multiply) F_e by Load Factor

... depends on application + type of loading

TABLE 11-5
Bearing-Life Recommendations for Various Classes of Machinery

TYPE OF APPLICATION	LIFE, kh
Instruments and apparatus for infrequent use	Up to 0.5
Aircraft engines	0.5–2
Machines for short or intermittent operation where service interruption is of minor importance	4–8
Machines for intermittent service where reliable operation is of great importance	8–14
Machines for 8-h service which are not always fully utilized	14–20
Machines for 8-h service which are fully utilized	20–30
Machines for continuous 24-h service	50–60
Machines for continuous 24-h service where reliability is of extreme importance	100–200

24hr ⇒ service ⇒ 5-6 yrs is suggested design life

Example

- Need a ball bearing (02 series) for continuous 24 hr operation.
- Loading is 2kN radial and 1kN axial @ 1000 rpm
- Select bearing (No impacts)

① The life is $\simeq 50{,}000 \text{ hrs} \times 60 \frac{\text{min}}{\text{hr}} \times 1000 \frac{\text{rev}}{\text{min}}$

$\simeq 3 \times 10^9$ revolutions

$L = 3000$ million

② Try load factor $= 1$

3 Use $C = F_e L^{1/a}$

$= F_e (3000)^{1/3}$

$$\boxed{C = 14.4 F_e}$$

4 But we need F_e !

Need C_o to get X & Y

Need bearing to get C_o

Try $C = 14.4 F_r$

$= \underline{28.8 \text{ kN}}$

Then BORE = 40 mm Table 11-3

$C = 30.7 \text{ kN}$ $C_o = 16.6 \text{ kN}$

Then $\frac{F_a}{C_0} = \frac{1kN}{16.6kN} = \underline{\underline{0.06}}$

$\underline{e \simeq 0.26} < \frac{F_a}{F_r} = \frac{1kN}{2kN} = \underline{\underline{\frac{1}{2}}}$

$\therefore X_2 = 0.56$ & $Y_2 = 1.71$

(.56) (1)(2)

$F_e = XVF_r + YF_a$

$= 1.12 + 1.71 =$

$= \underline{\underline{2.83\ kN}} > 2kN = VF_r$

and $C = 40.7\ kN > \underline{30.7kN}$

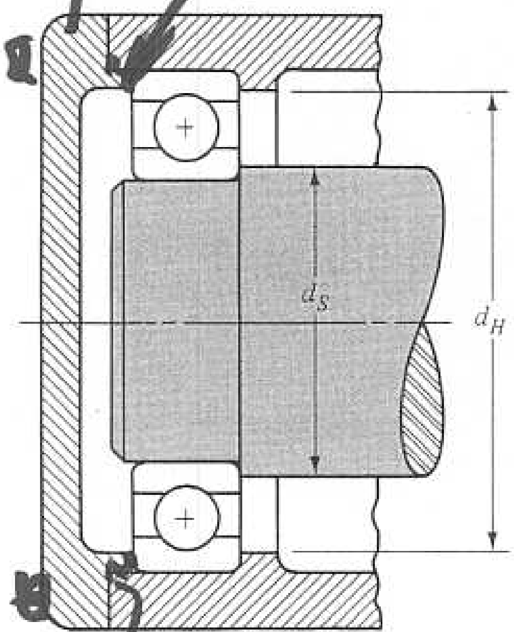
$= 2.83 \times 14.4$

Need to try larger bearing $\Rightarrow$ $\underline{\underline{55\ mm\ bore}}$

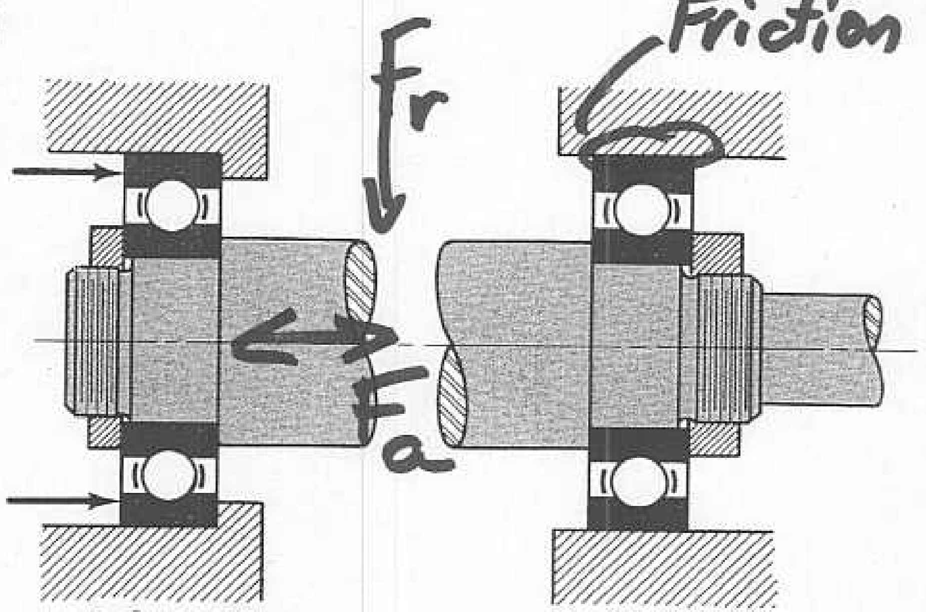
MOUNTING

Locating cap

Shim

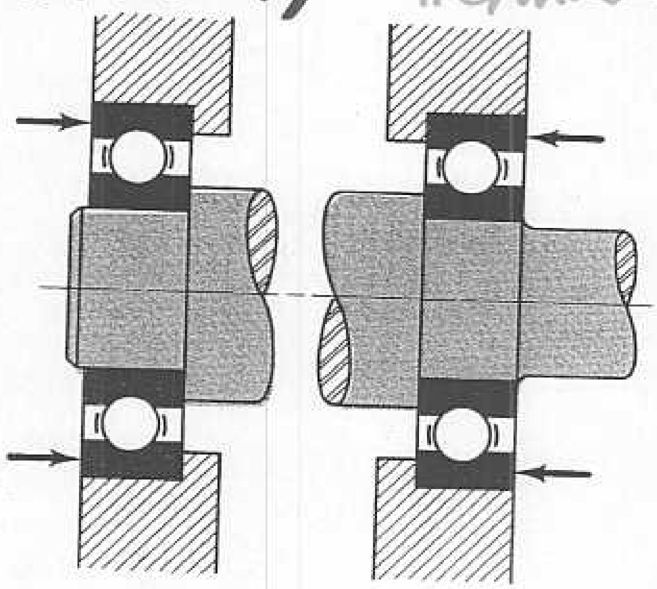


THREAD?



FIXED (Handles all thrust)

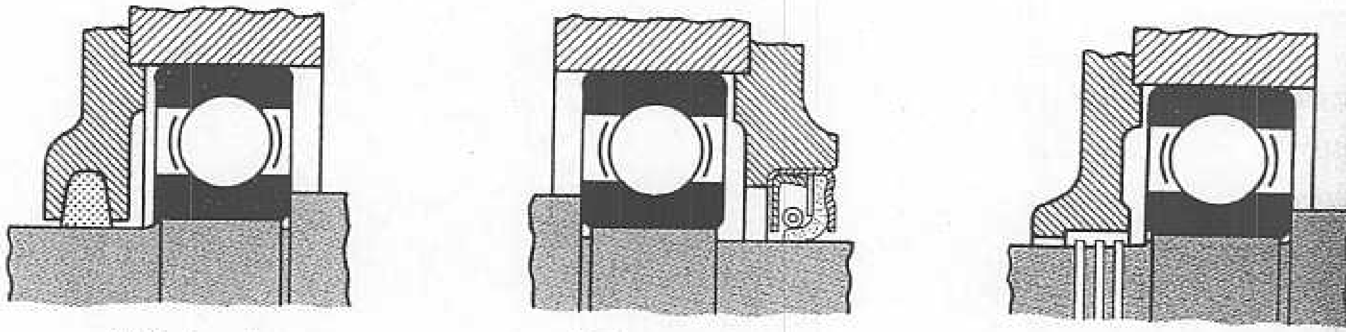
"FLOATING" e.g. for Thermal expansion



FIXED FIXED.

(DOESN'T HANDLE THERMAL EXPANSION WELL)

Lubrication, sealing/enclosure



(*a*) Felt seal (*b*) Commercial seal (*c*) Labyrinth seal

Bearings must be lubricated (grease or oil) and sealed

grease $\Rightarrow$ lower temps
below 200 °F
90 °C

oil $\Rightarrow$ needs oil/
supply, pump

CLEARANCE

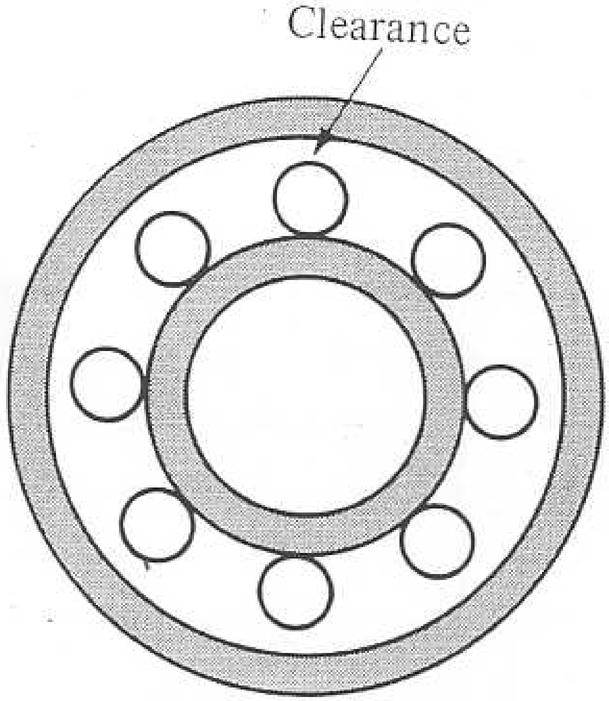
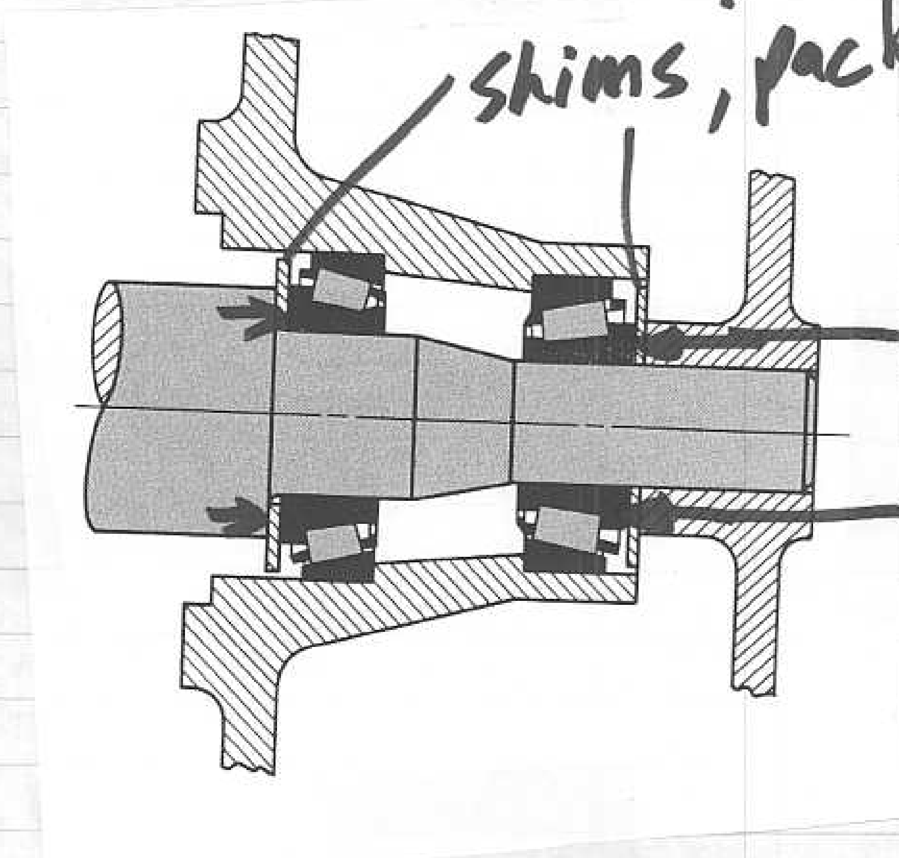


FIGURE 11-17

Clearance in an off-the-shelf bearing; exaggerated for clarity.

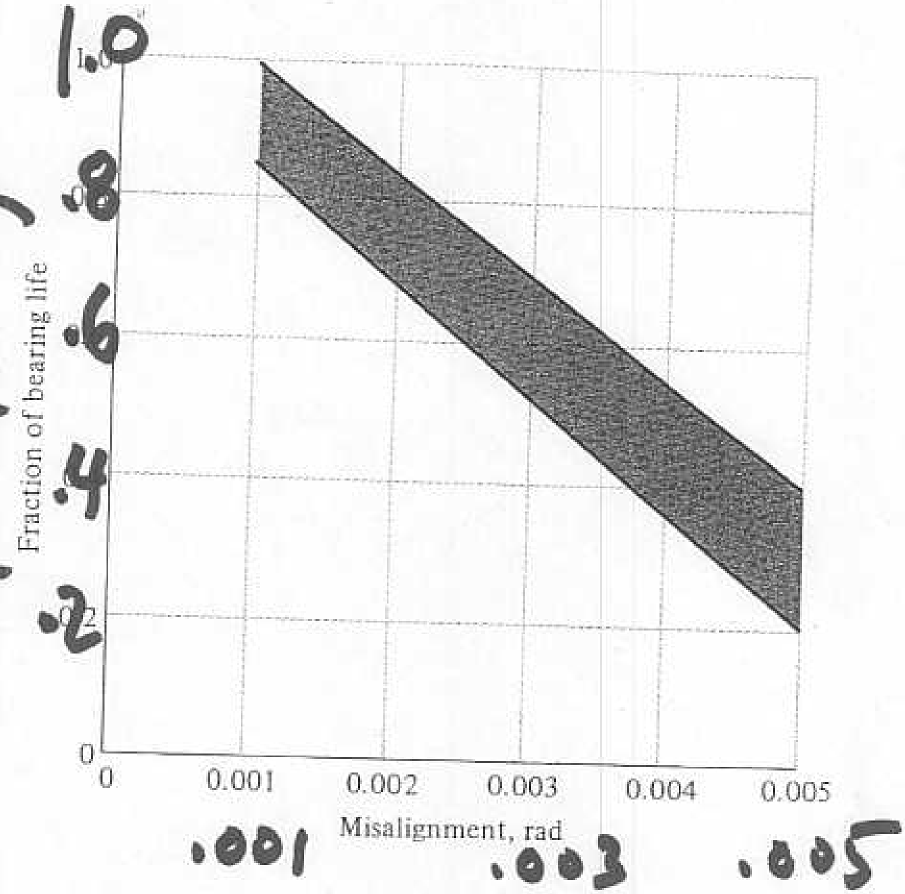
- All bearings have some clearance, "play"
- Sometimes taken up by axial preload (in part) ... roller bearings esp.

MOUNTING & PRELOAD IN TAPERED ROLLER BEARING



- Creates axial Thrust even without external axial load
- takes up play
- too much preload → friction, failure

Available fraction of L_{10} Life



Misalignment in radians

- Significant loss of life with small misalignments
- also need to be concerned with assembly tolerances and THERMAL EXPANSION

(Interesting)

Toroidal Self-
Aligning bearing

... recent SKF
invention
.... call "CARB"

... replaces ball,
roller & spherical
roller bearing

... accommodates thermal
expansion, shaft "
misalignment & housing "
misalignment

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The traditional self-aligning bearing arrangement

Bearings in rotating equipment

In typical industrial equipment, rotating shafts are generally supported by two anti-friction (rolling) bearings, one at each end of the shaft. In addition to supporting radial loads, one of the bearings must position the shaft axially with respect to its supporting structure, as well as carry any axial loads which are imposed on the shaft. This bearing is referred to as the "locating", "fixed" or "held" bearing.

The other bearing must also carry radial load, but should accommodate axial movement in order to compensate for:

- thermal elongation and contraction of the shaft or structure with temperature variations,
- manufacturing tolerances of the structure, and
- positioning tolerances at assembly of the machine.

This second bearing is referred to as the "non-locating", "floating" or "free" bearing.

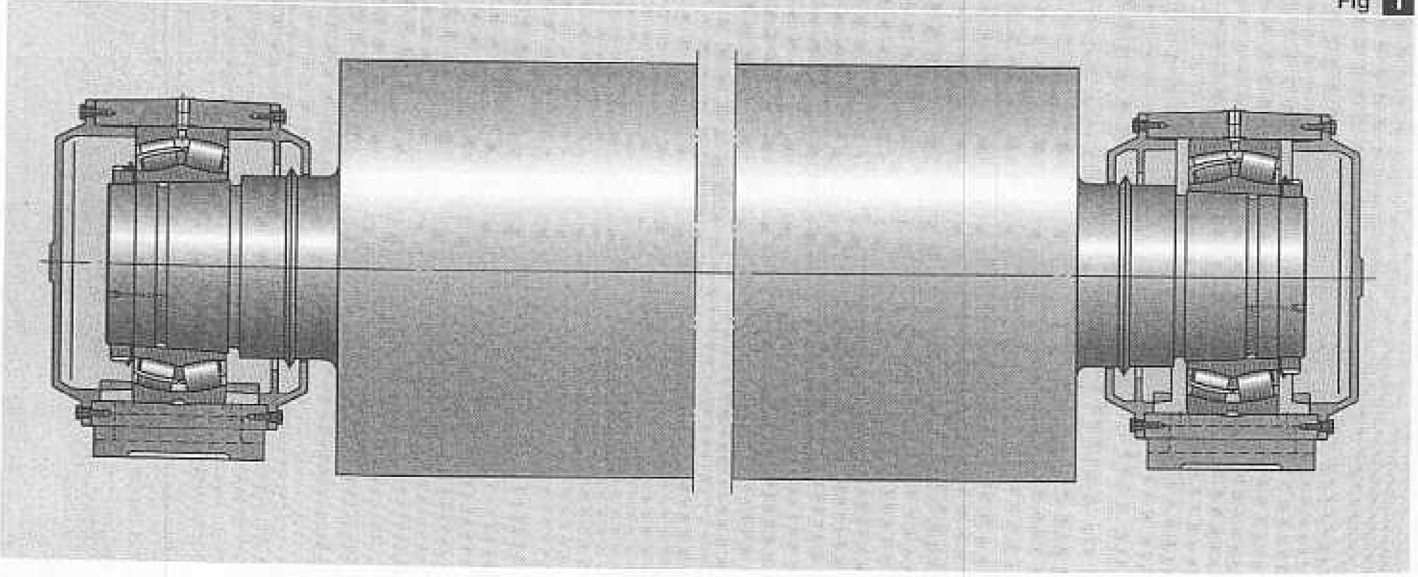
Using a self-aligning ball or a spherical roller bearing at both positions

This bearing combination has long been the basis of many industrial bearing arrangements – a simple, robust arrangement capable of withstanding high radial as well as thrust loads whilst easily internally accomodating the misalignment typically imposed through machining and assembly tolerances, thermal distortion or deflection under load. There are, however, consequences to using self-aligning ball bearings or spherical roller bearings in the non-locating position (→ fig 1).

The non-locating bearing must slide axially, usually inside the housing, to accommodate shaft expansion or contraction. To achieve this movement, one of the bearing rings must be mounted with a loose fit and axial space needs

Traditional arrangement of two self-aligning bearings with the bearing to the right "axially free"

Fig 1



Influence of friction

A more general, but less recognised, consequence of a bearing installed with a loose fit is that there is always a certain amount of friction between the loose bearing ring and the housing (or shaft) seating. In order for the shaft to expand or contract in the axial direction, it must first overcome the frictional resistance at the sliding contact. This resistance has the magnitude $F_a = F_r \times \mu$, where F_a is axial force, F_r is the radial load carried by the non-locating end bearing, and μ is the coefficient of friction between the loose bearing ring and the housing or shaft (for steel-steel and steel-cast iron interfaces, values for μ are typically around 0,12–0,16 for surfaces in good condition). Therefore, both bearings on the shaft are subjected to an additional thrust load, equivalent to several percent of the radial load (→ **fig 3**). As a result of these internal thrust forces, the load distribution within the bearings is adversely affected, with each row of rollers carrying a different load (→ **fig 4**).

Unstable load distribution

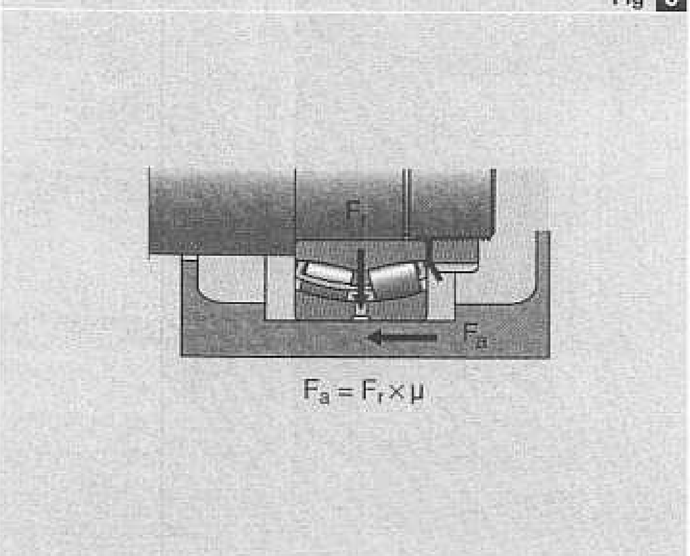
In cases with relatively high speeds, the load distribution is variable and unstable. To visualise how this mechanism occurs, picture the inner ring of one bearing being slightly askew on the shaft relative to the true axis of rotation – this is a common situation, typically a result of machining inaccuracy of the shaft, deflection of the shaft, combination of tolerances of shaft, adapter sleeve and bearing ring, and mounting inaccuracy. Then, as the inner ring rotates, it performs a very small "wobbling" motion, which imparts a small axial oscillation to the shaft. This oscillation is then transmitted to the inner ring of the second bearing in the shaft arrangement. As the inner rings move back and forth with a frequency equal to the shaft speed, the two rows of rollers are alternately loaded and unloaded (or at least change the amount of load they experience). In some cases, the axial motion is transmitted to the outer ring of the non-locating bearing, bringing about axial "scuffing" or fretting wear in the housing. The typical results of this uneven load distribution can be generalised as:

- in high load applications – elevated internal stresses, elevated temperature, impaired lubrication, accelerated bearing wear, reduced bearing fatigue life (reduction in fatigue life can be calculated)
- in high speed applications – high operating temperatures, alternating acceleration and deceleration of the roller sets with fluctuating load distribution, high forces on the bearing cages, increased rate of wear, high vibration and noise levels, rapid deterioration of grease, general reliability problems. (It is not possible to calculate any of these effects – this is the distinction between fatigue life and service life.)

These factors occur to a greater or lesser extent in all such bearing arrangements, even when the components are new and tolerances are within specification. If there is something other than normal friction which prevents movement of the non-locating bearing ring then the situation is equivalent to having a very high coefficient of friction (μ) at the non-locating bearing, and the adverse effects during operation are correspondingly severe.

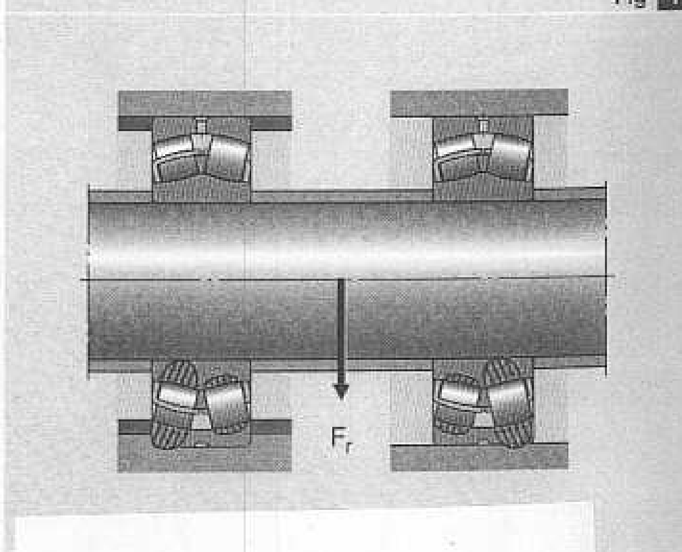
Friction between outer ring and housing induces axial load

Fig 3



Uneven load distribution due to axial friction forces

Fig 4



3

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A typical example of shaft expansion and its effects

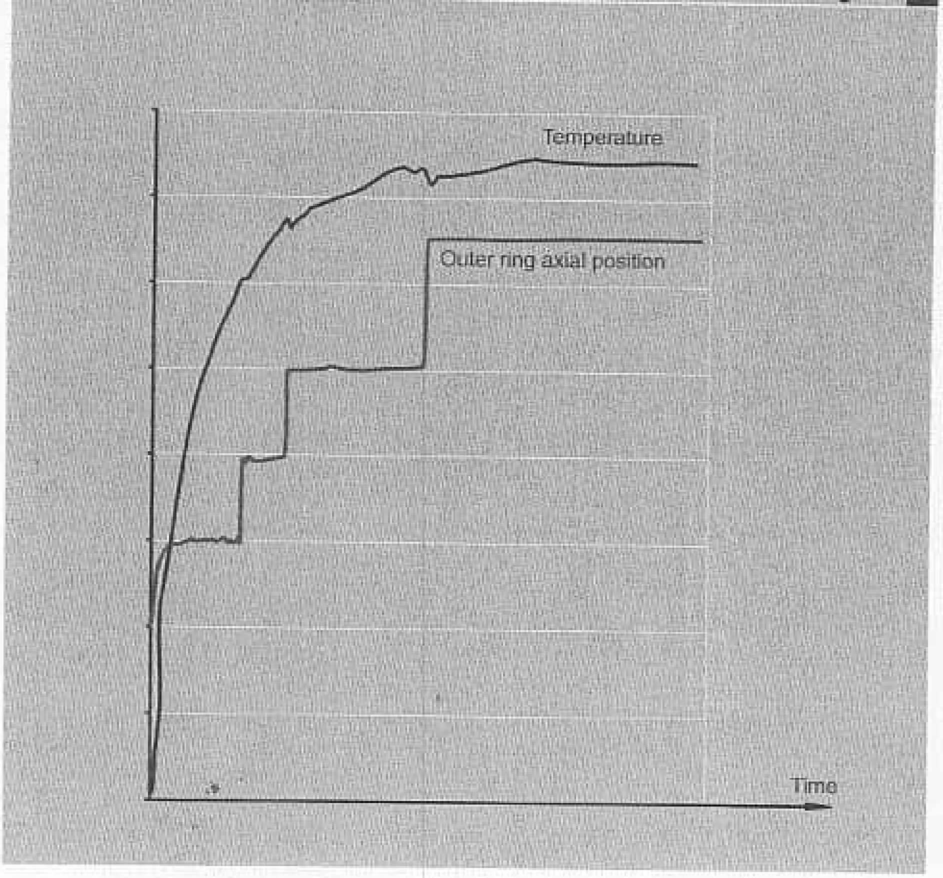
Diagram 1 shows measurements taken at the outer ring of an oil-lubricated spherical roller bearing at the non-locating position on a paper machine roll, during the machine start-up period.

It is apparent that the friction between the bearing and the housing is real and does have a significant effect on the bearing arrangement. As the operating temperature increases, the outer ring axial movement is characterised by stationary periods, with occasional sudden large movements. It is clearly noticed that the movements only occur when the axial forces have built to such a level that the stick-slip friction is overcome. With each movement of the outer ring, there is an immediate and noticeable reduction in the operating temperature, as the internal axial load is reduced.

This process is repeated until (or unless) a steady-state operating condition is reached, and then will be repeated (in reverse) with any decrease in temperature of the shaft or structure. (Shutdown, idle running, change in process parameters.) During steady state running conditions, it is likely that there will be some residual axial loading (uneven load distribution between the roller sets)

Note that for SKF "CC" and "E" spherical roller bearings, the ratio of axial load to radial load must be quite high (15 % or more) before there is a significant increase in the total rolling friction inside the bearing (total friction = load-dependent friction + viscous friction from the lubricant). Therefore, for bearings under nominally pure radial load, there must be a significant axial force resulting from friction between the bearing ring and housing in order for temperature variations such as those in the diagram to be noticeable. The fact that a change in temperature is easily measured shows that the friction factor acting will be > 0,1.

Diagram 1



#4

See

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The new self-aligning bearing system

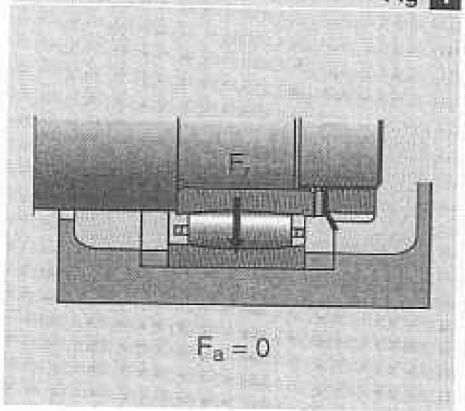
Until recently, the design compromises in each case simply had to be accepted. Now, however, a completely new design of non-locating rolling element bearing has enabled all the compromises to be eliminated from shaft/bearing systems.

The new bearing type is the toroidal roller bearing, so called because of the form of the curvature at the contact surfaces within the bearing. The toroidal roller bearing has a single row of long rollers with a slightly curved profile. The internal design enables the bearing to accommodate axial movement internally, like a cylindrical or needle roller bearing, without any frictional resistance. This eliminates the need for a loose fit for any of the bearing rings. There is also no possibility for generating additional axial (thrust) forces between the two bearings on the shaft (→ **fig 1**).

In addition to eliminating all the axial interaction between the bearings, the roller and raceway profile in the toroidal roller bearing is designed to automatically adjust the roller position inside the bearing so that the load is distributed evenly along the roller contact length, irrespective of any misalignment. This means that there is no possibility for high edge stresses, so the bearing always operates at the optimum stress level, and therefore achieves its theoretical fatigue life under all conditions (→ **fig 2**).

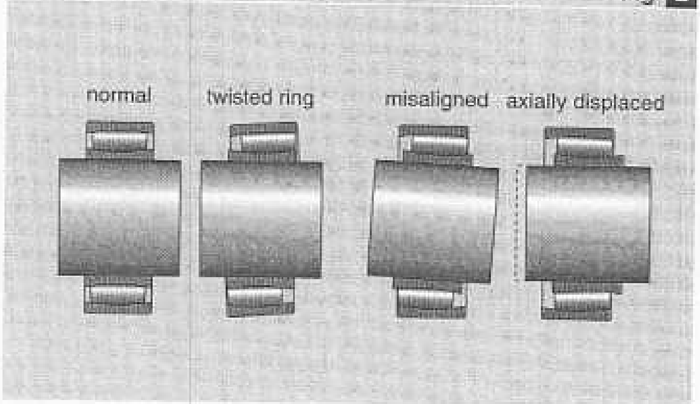
The combination of the self-aligning properties, and the frictionless axial adjustment, ensures that the load is distributed as evenly and consistently as possible along all the rows of rolling elements in both bearings on the shaft. The actual distribution will depend on the externally applied radial and axial loads. An optimised load distribution will mean that stresses are low, temperature is minimised, maximum fatigue life is always achieved, and the chance of vibrations and cage damage are reduced. In addition, because tight fits can be used for all bearing rings in the system, the risk of worn out housings due to spinning rings can be eliminated (→ **fig 3**).

Fig 1



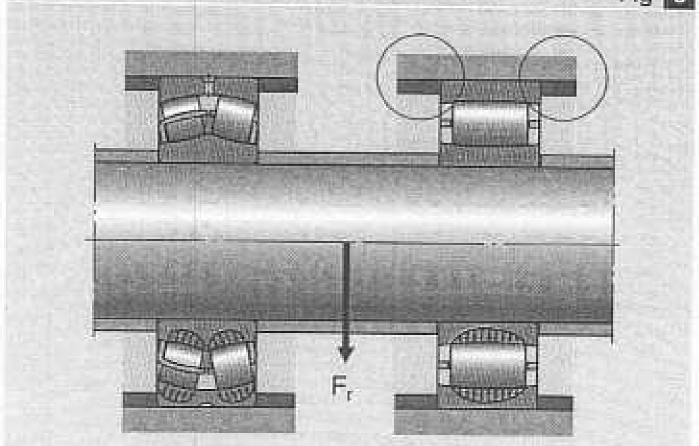
No axial force induced with toroidal roller bearing

Fig 2



Toroidal roller bearing contact surface ensures even load distribution along roller length

Fig 3



Please note that when using CARB, the outer ring of the non-locating bearing must also be fixed!

No internally induced axial load ensures even load distribution in both bearings

Cost reduction through downsizing

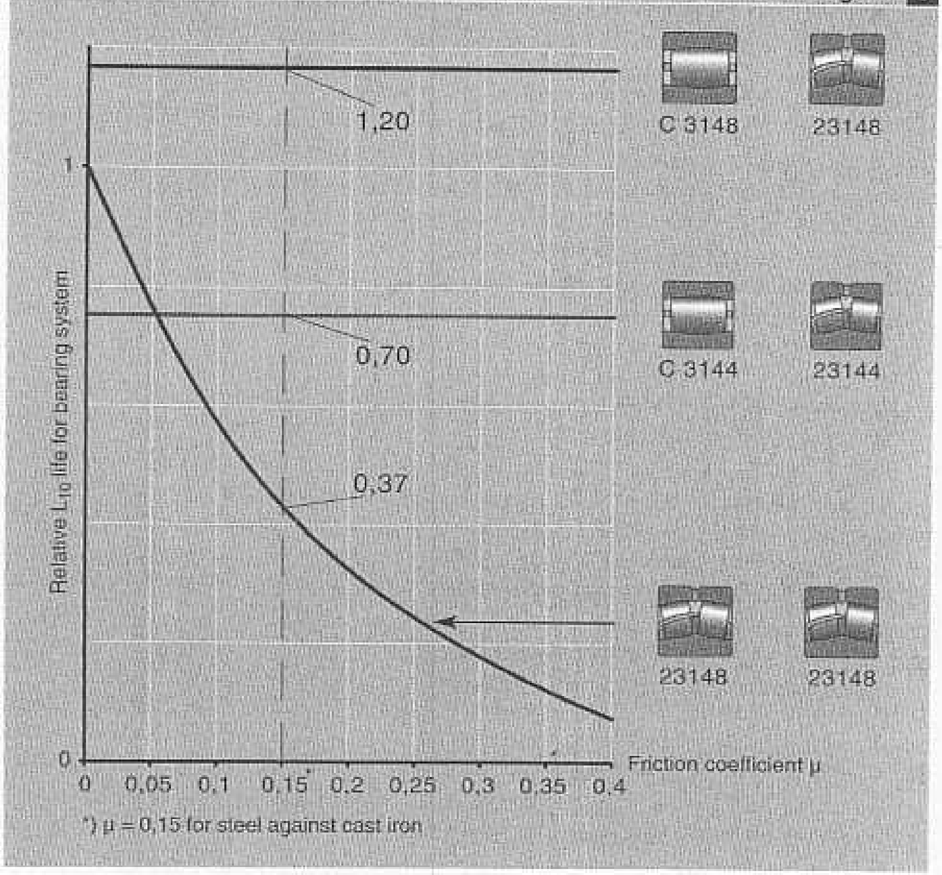
In addition to the obvious performance enhancements, operating cost reductions and productivity improvements that the CARB/spherical solution can give, there are additional benefits that can be realised from this new and unique self-aligning bearing system. The bearing system with CARB as the non-locating bearing has no internally generated thrust forces ($F_a = 0$ for both bearings) whereas traditional bearing arrangements do have thrust forces ($F_a = F_r \times \mu$ for both bearings). From this it is a simple matter to calculate the difference in fatigue life obtained in each system. In cases where the life of a conventional arrangement restricts machine performance, substituting a toroidal roller bearing at the non-locating position could significantly extend service life.

Where a conventional arrangement provides adequate service life, then using the new bearing system design, but with smaller bearings in both positions, can also often achieve the required life (→ **Diagram 6**). Therefore, there can be significant opportunities to not only use smaller, less expensive bearing assemblies without the risks of failure associated with axial preloading and general lack of axial freedom, but also to reap the flow-on cost benefits by the consequential size and weight reduction of other structural components.

For example, if one size smaller bearing, housing, seal and adapter sleeve assembly can be utilised at each end of a shaft, then the potential savings can include:

- bearing assembly purchase price
- bearing assembly weight
- shaft diameter and length reduction (material cost and weight savings)
- support structure size and weight reduction[1] (material cost and weight savings)
- less stringent machining and assembly tolerances[1] (production time savings)
- transportation cost reduction as a result of decreased overall machine weight

Diagram 6



The new self-aligning bearing system can give better life even with smaller bearings

[1] As there is no risk of cross-location of the bearings due to the frictionless internal axial movement in CARB, there is less importance placed upon the form and rigidity of the supporting structure for the bearings, meaning that lighter, more flexible and less precise components can be tolerated without a corresponding reduction in bearing performance.

Low load, high speed

In high speed applications where there are light loads and the possibility of misalignment, self-aligning ball bearings have been the standard solution. These applications can also benefit greatly from using the toroidal roller bearing at the non-locating position, for all the same reasons as described previously for the spherical roller bearing solution. Self-aligning ball bearings are much more susceptible to damage from axial preloading than spherical roller bearings, so eliminating the possibility of friction-induced axial forces has even greater significance in avoiding premature failure (→ **Diagram 7**).

SKF's new self-aligning bearing system consists of a CARB® toroidal roller bearing as the non-locating bearing in combination with a double row spherical roller or self-aligning ball bearing as the locating bearing.

The bearing arrangement accomodates both misalignment and axial adjustment internally and without frictional resistance, with no possibility of generating internal axial forces in the bearing system. Due to the ideal interaction between the two bearings, the applied load is always distributed consistently and in the assumed (theoretical) manner between all load-carrying elements.

The design characteristics of both bearings in the new system are fully exploited; they function as the machine designer intends and assumes they should. This is often not the case with other bearing systems, which have some compromise in the arrangement which produces non-ideal operating conditions.

The new compromise-free SKF system delivers increased reliability and performance, enabling designers to confidently optimise the selection of bearings and the machine construction as a whole.

Both manufacturers and end users of machines achieve significant cost reductions through a leaner design and improved productivity.

Depending on the machine and application, the benefits seen with SKF's new self-aligning bearing system are:

- safer, more optimised designs
- increased bearing service life
- extended maintenance intervals
- lower running temperature
- lower vibration and noise levels
- greater throughput of the machine
- same throughput with a lighter, or simpler machine
- improved product quality/less scrap