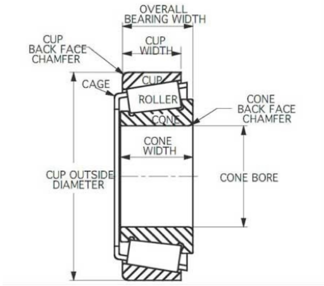
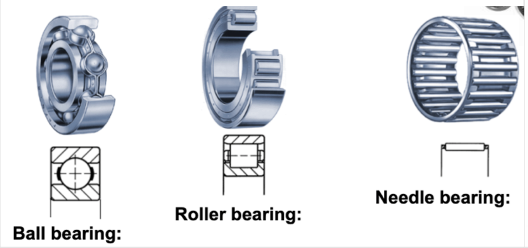
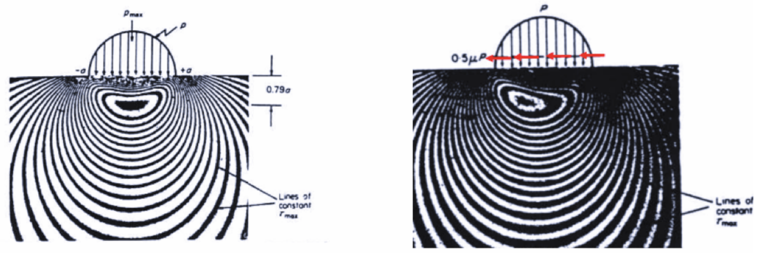
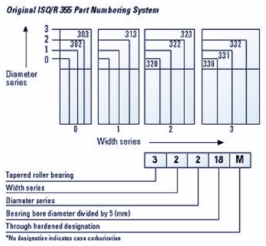
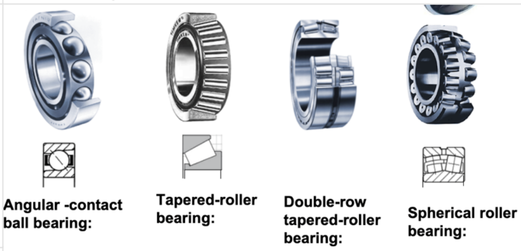
Tapered-Roller Bearing



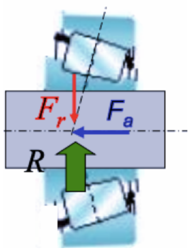


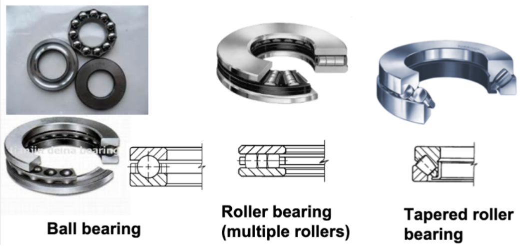


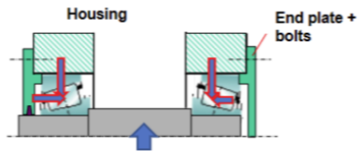


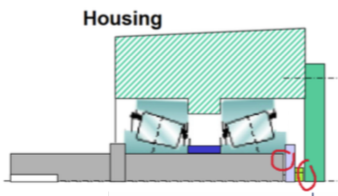


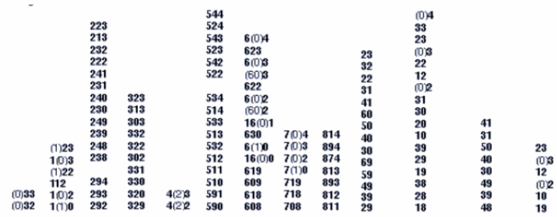
a: Half contact width, max shear stress not symmetric, gets closer to the surface





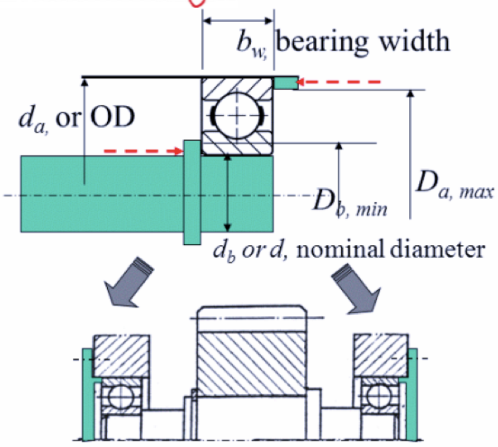






Radial and induced

Performance	Rolling bearings	Sliding bearings		
		with mixed lubrication	with full-film lubrication	Hydrostatic bearings
Load (W)-speed (N) relation	N/A	W⇓ as N⇑	W⇑ as N ⇑	N/A
Structural flexibility	Not flexible	Very flexible	Very flexible	Can be flexible
Shock buffering ability	low	low	high	high
High-speed operation	fair	poor	good	good
Starting resistance	low	high	May be high	low
Power loss	may be low	high	low	may be low
Life	fair	low	can by long	can be very long
Noise	high	low	can be very low	can be very low
Stiffness	high	fair	fair	can be very high
Rotation accuracy	high	low	can be high	very high
Radial dimension	large	small	small	small
Axial dimension	(0.2-0.5)d	(0.5-4)d	(0.5-0.4)d	fair
Lubricant	oil or grease	oil, grease, solid	oil, gas	oil, gas
Lubricant consumption	very low	low	high	very high
Maintenance	easy	change sleeves, shaft re-polishing	change sleeves, shaft re-polishing	change sleeves, shaft re-polishing
Cost	fair	may be low	high	may be very high



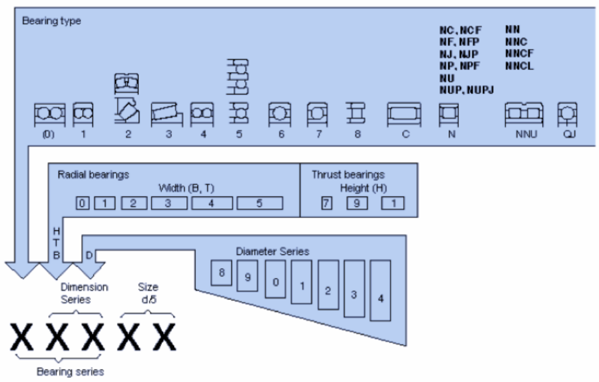


Figure 6-14 SKF bearing designations (Courtesy of SKF)

Radial	Friction	Speed	Thrust Load	Load Size	Features
Ball	Lower	Higher	Allows small	Light-Med	
Roller	Higher	Lower	No	Higher	HEAVY LOAD Separable
Needle	Higher	Lower	No	Higher	Compact

Radial-Thrust	Friction	Speed	Thrust Load	Load Size	Features
Ball	Lower	Higher	1 Direction Thrust	Light-Med	
Tapered-Roller	Higher	Lower	1 Direction Thrust	Higher	Separable
Double-Row Tapered-Roller	Higher	Lower	No	High	Compact
Spherical Roller	Higher	Lower	No	High	Misalignment tol

Thrust	Friction	Speed	Thrust Load	Load Size	Features
Ball	Lower	Higher	1 Direction Thrust	Light-Med	
Roller	Higher	Lower	1 Direction Thrust	Higher	Separable
Tapered Roller	Higher	Lower	No	High	Compact

Face to face

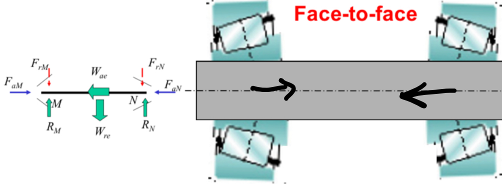
$$P_M \quad \max\begin{cases} XF_{rM} & Y_M(\frac{F_{rN}}{Y_N} \quad W_{ae}) \\ F_{rM} \end{cases}$$

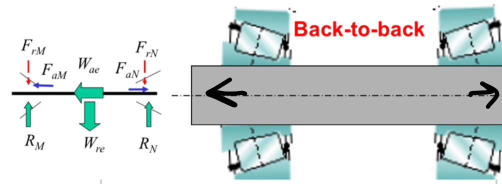
$$P_N \quad \max\begin{cases} XF_{rN} & Y_N(\frac{F_{rM}}{Y_M} \quad W_{ae}) \\ F_{rN} \end{cases}$$

Back to back

$$P_M \quad \max\begin{cases} XF_{rM} & Y_M(\frac{F_{rN}}{Y_N} \quad W_{ae}) \\ F_{rM} \end{cases}$$

$$P_N \quad \max\begin{cases} XF_{rN} & Y_N(\frac{F_{rM}}{Y_M} \quad W_{ae}) \\ F_{rN} \end{cases}$$





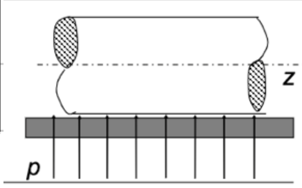
Case	Selection
High Speed	Ball
Single or Combined Radial and Thrust Load	Radial or Thrust, Radial-Thrust
Heavy Load	Roller
Very Heavy Load	Double Roller
Environmental Concern	Sealed
Misalignment	Spherical
Compact	Needle

Bearing Failure	Micor-pitting, geometric stress concentration (GSC) failure		
False Brinneling	Vibration, but no rotation, creates dents		
Bearing Life	No. of revs at which 10% fail	Rolling contact fatigue, Pitting, plastic deformation	
L10 (million revs)	$h(60)\frac{n}{10^6}$	n: rpm, h: life hours	
Same Capacity	$L_1 F_1^m = L_2 F_2^m$	m=3 [bb], m=10/3 [roller]	
Same Capacity	$\frac{L_1}{L_2} = (\frac{F_2}{F_1})^m$		
Basic Load Rating C (bearing rating)	$L = (\frac{C}{P})^m$	$C = PL^{\frac{1}{m}}$	
Equivalent Load (Ball Bearing)	$P = A_P(XP_r + YP_a)$	$P = A_P P_r$	
	Pa/Pr > e	Pa/Pr <= e	Ap = 1
Equivalent Load (Tapered)	$P = A_p(XF_r + YA_t)$	X=0.4, At: total Axial force,	
Induced Axial Force on Inner Ring	$F_a = \frac{\lambda F_r}{Y}$	lambda Timken ISO: 0.5, Timken: 0.47	
Bearing Designation SKF	6[x][xx]	x: structure factor (+1 wider, stronger), xx: diameter (/5 mm)	
Timken Tapered Roller	3[width][diameter][bore divded by 5 mm][through hardened designation]		

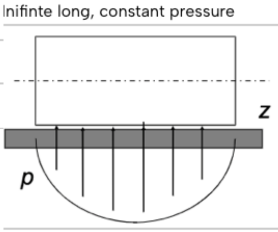
Hydrodynamic/Static Jounral Bearings	Radial Forces
Hydrodynamic/Static Slider Bearings	Thrust
Why Sliding bearing for ICE?	Save space, weight, assembly is easier
Lubrication	Thin film produced from dynamics or external pressure
Friction Coefficient and Hersey Number	higher viscosity, velocity, average pressure
Dynamic viscosity	Laminar flow, no boundary slip, newtonian fluid
Reynolds Assumptions	Laminar flow, no slip boundary condition, newtonian fluid, incompressible fluid, inertia ignored, surface rigid and smooth, fluid film thickness << geometry, constant pressure
Conditions for hydrodynamic lub	Converging gap, relative velocity, non zero viscosity
Influence of bearing geometry	Side Leakage reduced, more fluid doing antraining motion, load capactiy increased, film thickness increased
Load Film Relation	Higher load, thinner film
Slider bearings	Have wedges

Disadvantages of Hydrodynamic bearings	v=0 means no pressure build up, start, stop has higher friction and wear, eccentricity has low rotational accuracy
Hydrostatic bearings	Performace independent of bearing motion, no effect of start stop, bearing and shaft concentric

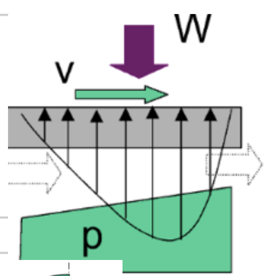
hmin not at Pmax

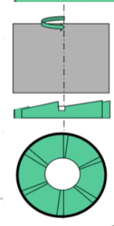


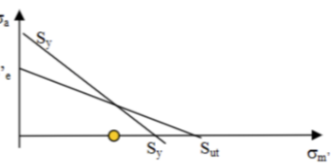
Inifinte long, constant pressure

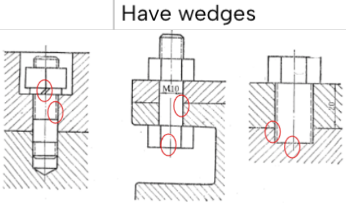


Finite, has boundary condition









...er direction wrong
...hould be smaller than that of the hole
... be removed
...art, Lt, too short. If it a tensile connection, there should be no fit
... hole; if shear connection, the thread OD should be thinner than the
...nience of assembly.
...d be no thread in the top member
...ctures for the hole and thread are not correct.

5-3 Factor of safety against fatigue failure

$$\sigma_{ba} = \frac{CP_a n_{fi}}{A_t} \qquad \sigma_{bm} = \frac{P_i + CP_m n_f}{A_t}$$

Goodman's line $\frac{K_f \sigma_a}{S_e} + \frac{\sigma_m}{S_{ut}} = 1$

$$\frac{K_f CP_a n_{fi}}{S_e A_t} + \frac{P_i + CP_m n_{fi}}{S_{ut} A_t} = 1 \qquad n_{fi} = \frac{(1 - \frac{P_i}{S_{ut} A_t})}{\left(\frac{K_f CP_a}{S_e A_t} + \frac{CP_{m'}}{S_{ut} A_t}\right)}$$

Recall

$S_e = k_f k_s k_r k_t k_m S_e' \approx k_r S_e'$ **For bolt materials**

$S_e' = 0.45 S_{ut}$ **For axial loading**

$$S_e = k_f k_s k_r k_t k_m S_e' \approx 0.45 k_r S_{ut}$$

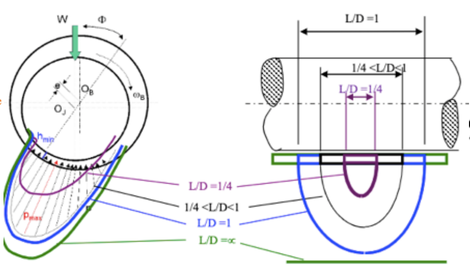
9

$$n_{fi} = \frac{(1 - \frac{P_i}{S_{ut} A_t})}{\left(\frac{K_f CP_a}{S_e A_t} + \frac{CP_{m'}}{S_{ut} A_t}\right)} \qquad n_{fi} = \frac{S_e(S_{ut} A_t - P_i)}{C(K_f S_{ut} P_a + S_e P_{m'})}$$

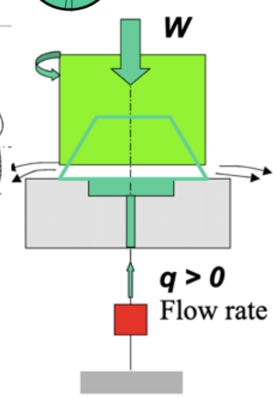
$$S_e \approx 0.45 k_r S_{ut}$$

Fatigue stress concentration factor, K_f

Metric grade	Rolled threads	Cut threads
3.6-5.8	2.2	2.8
6.6-10.9	3.0	3.8



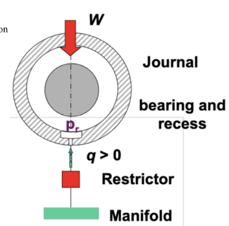
Length/diameter: *L/D*

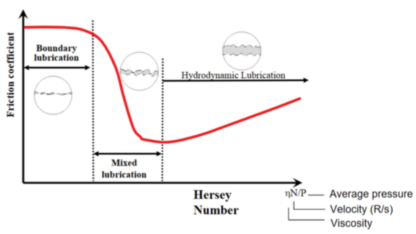


Slider bearing

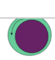
Table 6-3 (continued)

Nominal diameter	OD	Width	Basic load ratings Dynamic	Static limit	Allowable load	Speed ratings Grease	Oil	Mass	Designation
mm	mm	mm	C (N)	C_0(N)	(N)	rmp	rpm	Kg	
40	**68**	**15**	**16800**	**11600**	**490**	**9500**	**12000**	**0.19**	**6008**
	80	18	30700	19000	800	8 500	10000	0.37	6208
	90	23	41000	24000	1020	7500	9000	0.63	6308
	110	27	63700	36500	1530	6700	8000	1.25	6408
45	**75**	**16**	**20800**	**14600**	**640**	**9000**	**11000**	**0.25**	**6009**
	85	19	33200	21600	915	7500	9000	0.41	6209
	100	25	52700	31500	1340	6700	8000	0.83	6309
	120	29	76100	45000	1900	6000	7000	0.55	6409
50	**80**	**16**	**21600**	**16000**	**710**	**8500**	**10000**	**0.26**	**6010**
	90	20	35100	23200	980	7000	8500	0.46	6210
	110	27	61800	38000	1600	6300	7500	1.05	6310
	130	31	87100	52000	2200	5300	6300	1.90	6410





1 **Characteristics of journa-bearing interaction**
Long path of fluid flow
Long path of debris transport, frictional heating,
Large area of interaction, if any; scuffing or seizure possible
Contact occurs when at start and stop

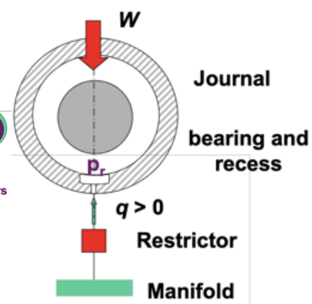


2 **Basic requirements**
Sufficient strength
Heat conductive
No adhesion
Compliant to deformations
Good embeddability
Fatigue resistant
Corrosion resistant
Low cost

3 Basic materials for low pairs

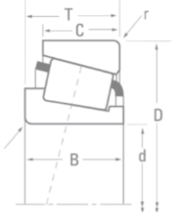
One harder one softer, and the softer one:

- Tin-based materials
- Cupper-based materials
- Aluminum alloy
- Cast iron
- Carbon-based materials
- Silver overlay
- Polymers and composites



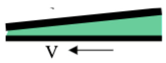
Timken Standard IsoClass Bearings

Base Part Number	ISO 355 Reference	d	D	T	B	C	R	r	Load rating C1(N)	Factors e	Y	Weight (kg)
30203	2DB	17	40	13.25	12.0	11.0	1.0	1.0	20 000	0.35	1.74	0.08
30204	2DB	20	47	15.25	14.0	12.0	1.0	1.0	28 300	0.35	1.74	0.13
30205	3CC	25	52	16.25	15.0	13.0	1.0	1.0	31 400	0.37	1.60	0.15
30206	3DB	30	62	17.25	16.0	14.0	1.0	1.0	41 800	0.37	1.60	0.23
30207	3DB	35	72	18.25	17.0	15.0	1.5	1.5	51 100	0.37	1.60	0.32
30208	3DB	40	80	19.75	18.0	16.0	1.5	1.5	60 100	0.37	1.60	0.41
30209	3DB	45	85	20.75	19.0	16.0	1.5	1.5	69 400	0.40	1.48	0.46
30210	3DB	50	90	21.75	20.0	17.0	1.5	1.5	72 100	0.42	1.43	0.52
30211	3DB	55	100	22.75	21.0	18.0	2.0	1.5	90 400	0.40	1.48	0.68
30212	3EB	60	110	23.75	22.0	19.0	2.0	1.5	96 700	0.40	1.48	0.87
30213	3EB	65	120	24.75	23.0	20.0	2.0	1.5	115 200	0.40	1.48	1.10
30214	3EB	70	125	26.25	24.0	21.0	2.0	1.5	126 000	0.42	1.43	1.28
30215	4DB	75	130	27.25	25.0	22.0	2.0	1.5	134 000	0.44	1.38	1.33
30216	3EB	80	140	28.25	26.0	22.0	2.5	2.0	152 000	0.42	1.43	1.59
30217	3EB	85	150	30.50	28.0	24.0	2.5	2.0	176 000	0.42	1.43	2.02
30218	3FB	90	160	32.50	30.0	26.0	2.5	2.0	202 000	0.42	1.43	2.49
30219	3FB	95	170	34.50	32.0	27.0	3.0	2.5	221 000	0.42	1.43	2.95
30220	3FB	100	180	37.00	34.0	29.0	3.0	2.5	252 000	0.42	1.43	3.77
30221	3FB	105	190	39.00	36.0	30.0	3.0	2.5	274 000	0.42	1.43	4.48
30222	3FB	110	200	41.00	38.0	32.0	3.0	2.5	311 000	0.42	1.43	5.19
30224	4FB	120	215	43.50	40.0	34.0	3.0	2.5	331 000	0.44	1.38	6.15
30226	4FB	130	230	43.75	40.0	34.0	4.0	3.0	360 000	0.44	1.38	7.60
30228	4FB	140	250	45.75	42.0	36.0	4.0	3.0	418 000	0.44	1.38	8.65
30230	4GB	150	270	49.00	45.0	38.0	4.0	3.0	469 000	0.44	1.38	11.00



C_1: corresponding to one million revolutions (ISO)

1) Can hydrodynamic pressure be generated at the fluid-filled interface shown below, where the bottom surface moves left as indicated by the arrow, the top surface is stationary.

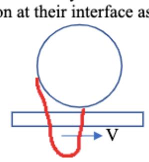


(a) Yes (b) No

2) Under the same load and speed, and using the same lubricant (oil), a wider (longer) journal bearing runs with a _________ oil film between the surfaces of the shaft and the bearing if the lubricant supply is sufficient?

a) Thicker b) Thinner

3) In the figure below, the roller is stationary while the flat surface is moving to the right, sketch the lubrication pressure distribution at their interface assuming a lubricant is used.



Missing two clearances.

6) Is the following form of the Goodman line proper for the analysis of the bolt factor of safety against fatigue?

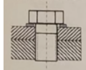
No

$$\frac{K_f\sigma_a}{S_e} + \frac{\sigma_m}{S_{ut}} = 1/n$$

7) In lubrication, should the maximum pressure and the minimum film thickness appear at the same location?

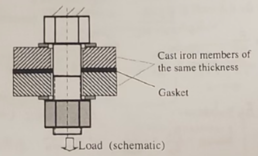
No

8). Is the following bolted connection designed correctly or not?



No

9-10) The bolted connection below has a steel bolt (B), two steel washers (W) of the same thickness, two cast-iron members (M) of the same thickness, and a gasket (G). The bolt and frustum-cone stiffnesses are K_B, K_{Wi}, K_{Mi}, and K_{Gi}, respectively, with i for the i^{th} cone of each material.



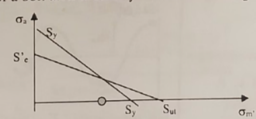
9) Number of frustum cones = 6

10) Member stiffness = k_G or $1/k_m = 1/k_{G1} + 1/k_{G1}$.

3) For a bolted connection under shear loading, which of the following is proper for interfacial friction calculation?

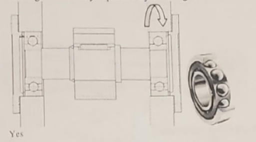
(a) Preload (b) Bolt force (c) Member force

4) Mark, without calculation, the work stress point of a bolt after assembly but before loading and explain why.



5) A gasket between members can increase/decrease (choose one) the member stiffness. Can you quickly estimate the member stiffness if the stiffness of a gasket is Kg?

1) Two deep groove ball bearings are used to support the shaft system shown below. Can these bearings be directly replaced by two angular contact ball bearings?



Yes

2) Can hydrodynamic pressure be generated at the fluid-filled interface shown below, where the top surface moves left as indicated by the arrow, the bottom surface is stationary.

No

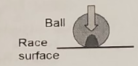
3) Under the same load and speed, and using the same lubricant (oil), should a narrower (shorter) journal bearing run with a thinner or thicker oil film between the surfaces of the shaft and the bearing if the lubricant supply is sufficient?

Thinner

4) For a regular bolted connection under shear loading, in which the shear load is taken by friction, the stress in the bolt is determined by the

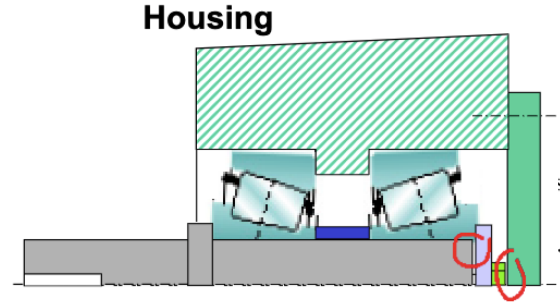
(a) Preload (b) Shear force (c) Member force

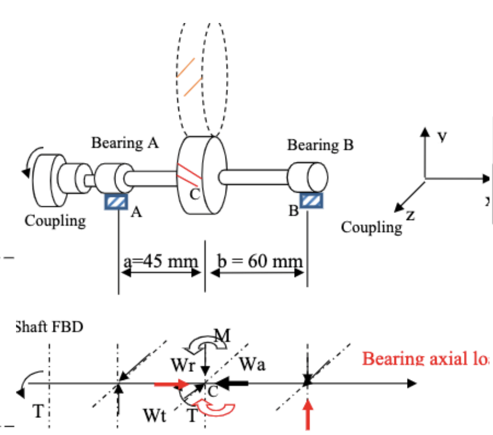
5) For the roller rolling on the bearing race surface (shown below, simplified as a roller on a flat surface) with lubrication, is the hydrodynamic pressure distribution the same as the following dry contact pressure distribution (shown in red)?

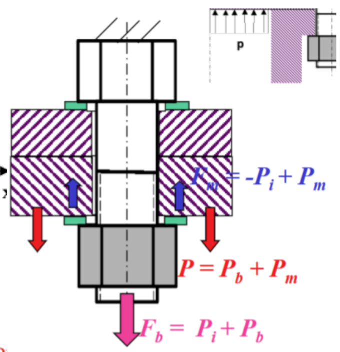


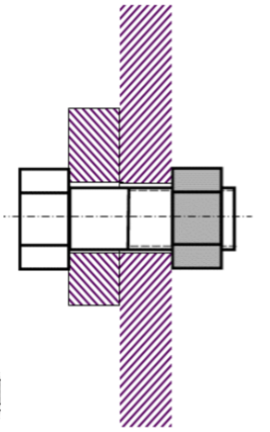
No

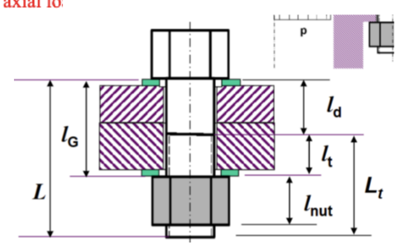
Housing

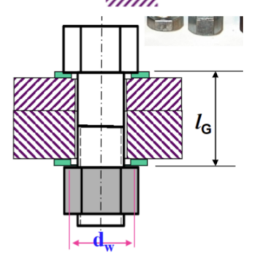


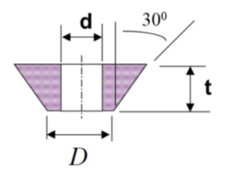


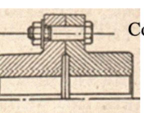








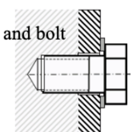


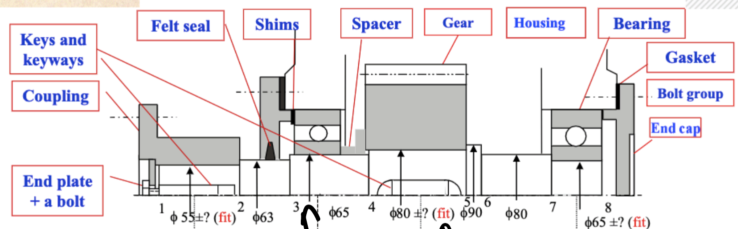


Coupling halves and bolt

Spacer: optional for no thrust-loading cases, but it is required for our project

End cap and bolt





TIGHT

TRANSITION

$$k_b = \frac{A_t A_d E}{(l_t + 0.4 d_r) A_d + (l_d + 0.4 d) A_t}$$

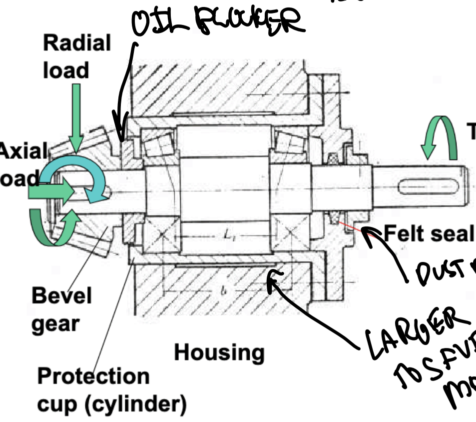
k_d l_{de}

k_G l_{ee}

FOR GASKET: $k_g \ll k_{mc}$, $k_m \simeq k_g$

$$\frac{1}{k_m} = \sum_{i=1}^{n} \frac{1}{k_{mi}}$$; n CONES

OIL BLOCKER



PROTECTION CUP: ASSEMBLY OUTSIDE

DUST BLOCKER

LARGER TO SAVE MONEY

MANUFACTURING: COLD HEADING (GRAIN FLOW) FOLLOW PART WORK HARDENING

THREAD ROLLING

4-3 Load in the bold

A. Forces Applied load, P

Preload, or the clamping force: P_i

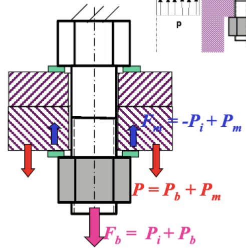
Applied load supported by the bolt: P_b

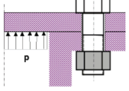
Applied load supported by the member: P_m

Applied tensile load: $P = P_b + P_m$

Resultant bolt load: $F_b = P_i + P_b$

Resultant member load: $F_m = -P_i + P_m$





B. Load in the bolt, F_b

LOAD!

$$F_b = (P_b + P_i) = \frac{k_b}{k_m + k_b} P + P_i$$

C. Joint constant

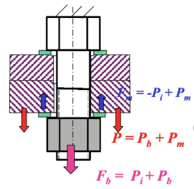
Joint constant:

$$C = \frac{k_b}{k_m + k_b}$$

D. Stress in the bolt, σ_b

$$\sigma_b = \frac{F_b}{A_t} = \frac{CP}{A_t} + \frac{P_i}{A_t}$$

$$F_b = (P_b + P_i) = \frac{k_b}{k_m + k_b} P + P_i$$



Preload selection

$P_i = 0.75\, A_t S_p$ Reused connection

$P_i = 0.9\, A_t S_p$ Permanent connection

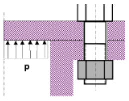
10

$P_b = \frac{k_b}{k_m + k_b} P$ $F_m = -P_i + P_m$ $C = k_b / (k_m + k_b)$

$$P_m = (P - P_b) = \frac{k_m}{k_m + k_b} P = (1 - C) P$$

4-4 Factors of safety

$$\sigma_b = \frac{F_b}{A_t} = \frac{CP}{A_t} + \frac{P_i}{A_t}$$



Factors of safety

A Load factor and anti over proof loading

$$n = \frac{(A_t S_p - P_i)}{CP}$$

LOAD FACTOR

B Anti separation

$$P_{n_{sp}} = P_0, \quad F_m = 0$$

$$n_{sp} = \frac{P_i}{P(1-C)}$$

$$C = \frac{k_b}{k_m + k_b}$$

e. Factors of safety

Anti separation

$$n_{sp} = \frac{P_i}{P(1-C)} =$$

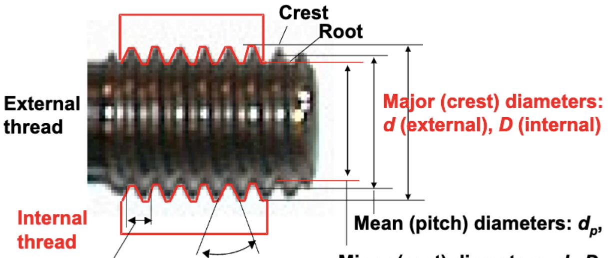
Anti over proof-loading

$$n = \frac{S_p A_t - P_i}{CP}$$

$P_i = 0.75\, A_t S_p$ **Reused connection**

$P_i = 0.9\, A_t S_p$ **Permanent connection**

$$S_p = \frac{proof-load}{Tensile-strength-area: A_T} \approx 0.9 S_y$$



Pitch: distance between two adjacent profiles

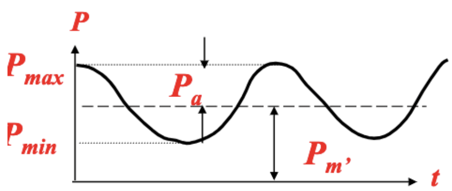
Thread angle: Usually, $2\alpha = 60^0$

Tensile stress area:

$$A_t = \frac{\pi}{4}\left(\frac{d_r + d_p}{2}\right)^2$$

Axial loading: $S'_e = 0.45 S_{ut}$

$$P_a = \frac{P_{max} - P_{min}}{2}$$



$$P_{m'} = \frac{P_{max} + P_{min}}{2}$$

Amplitude and midrange of the applied load

$$P_a = \frac{P_{max} - P_{min}}{2} \qquad P_{m'} = \frac{P_{max} + P_{min}}{2}$$

$$F_{ba} = CP_a \qquad F_{bm} = P_i + CP_{m'}$$

ANTI FATIGUE

$$\sigma_{ba} = \frac{CP_a n_{ft}}{A_t} \qquad \sigma_{bm} = \frac{P_i + CP_{m'} n_{ft}}{A_t}$$

$$n_{ft} = \frac{\left(1 - \frac{P_i}{S_{ut} A_t}\right)}{\left(\frac{K_f C P_a}{S_e A_t} + \frac{C P_{m'}}{S_{ut} A_t}\right)}$$

$$n_{ft} = \frac{S_e (S_{ut} A_t - P_i)}{C(K_f S_{ut} P_a + S_e P_{m'})}$$

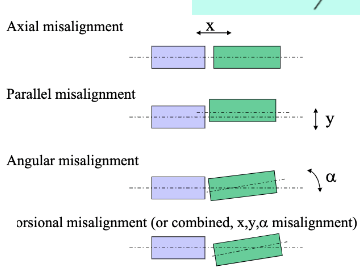
$$S_e = k_f k_s k_r k_t k_m S'_e \approx k_r S'_e$$

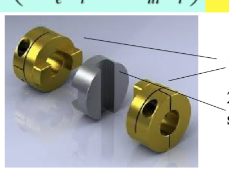
$$S'_e = 0.45 S_{ut}$$

$$S_e = k_f k_s k_r k_t k_m S_e \approx 0.45 k_r S_{ut}$$

Rigid couplings:

or well aligned shafts



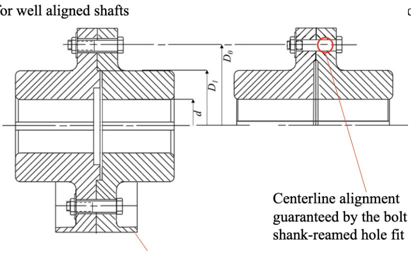


1 and 3 are identical

2 may be a soft material, such as a soft metal.

It may allow shaft axial and parallel misalignments.

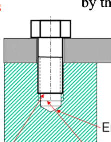
Low speed applications, due to wear and the centrifugal force on element 2.



Centerline alignment guaranteed by the bolt shank-reamed hole fit

Three flange couplings

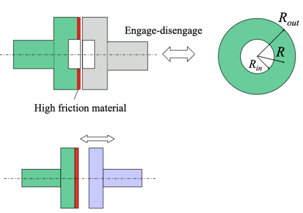
Centerline alignment guaranteed by the shoulder

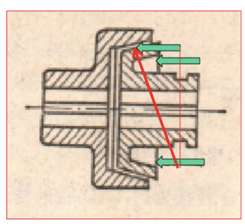


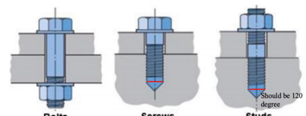
Clutches and Brakes

- Connecting and separating two rotating shafts
- Disk type: using friction
- Tooth type: using tooth engagement
- Over-running clutch: using the centrifugal force

Friction clutches







Bolts: goes entirely through both members. Wrench space on both sides. Fatigue affects bolt, not material

Screws: wrench space on top. Fatigue affects lower material. Pull whole thing out to get to members

Studs: plant rod first, then put nut on top. Remove nut without affecting bottom material.

Angle at tip should be 120 deg

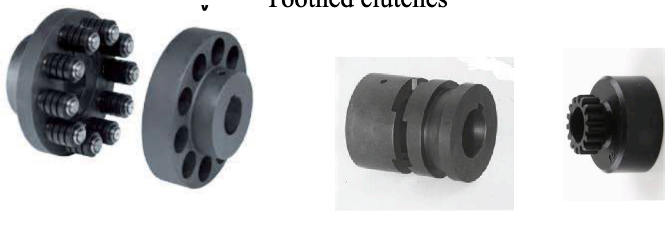
WEDGE IS GOOD FOR AMPLIFY FORCE

	External thread	Internal thread
Major (Nominal) diameter (Crest diameter of a bolt)	d	D
Minor diameter	d_r $d_r = d - 1.226869p$ (Metric) $d_r = d - 1.299038p$ (US)	D_r (Root diameter of a bolt)
Mean diameter	d_p $d_p = (d_r + d_p)/2$	D_p
Pitch, Lead, Number of thread,	p L_d n	$L_d = n\,p$
Tensile stress area	$A_t = \frac{\pi}{4}\left(\frac{d_r + d_p}{2}\right)^2$	

US:

Metr

Toothed clutches



No. of teeth: 3~60, smaller number for larger torque.