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Mechanical Analysis and Design
ME 2104

Lecture 2

Mechanical Analysis
Bearings and Screws

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www.staff.city.ac.uk/~ra600/intro.htm

Plan for the analysis of mechanical elements

Objective:

Procedures for design and selection of mechanical elements

- Week 1 – Shafts and keyways
- Week 2 – Bearings and screws
- Week 3 – Belt and chain drives
- Week 4 – Gears and gear trains
- Week 5 – Design Project Review

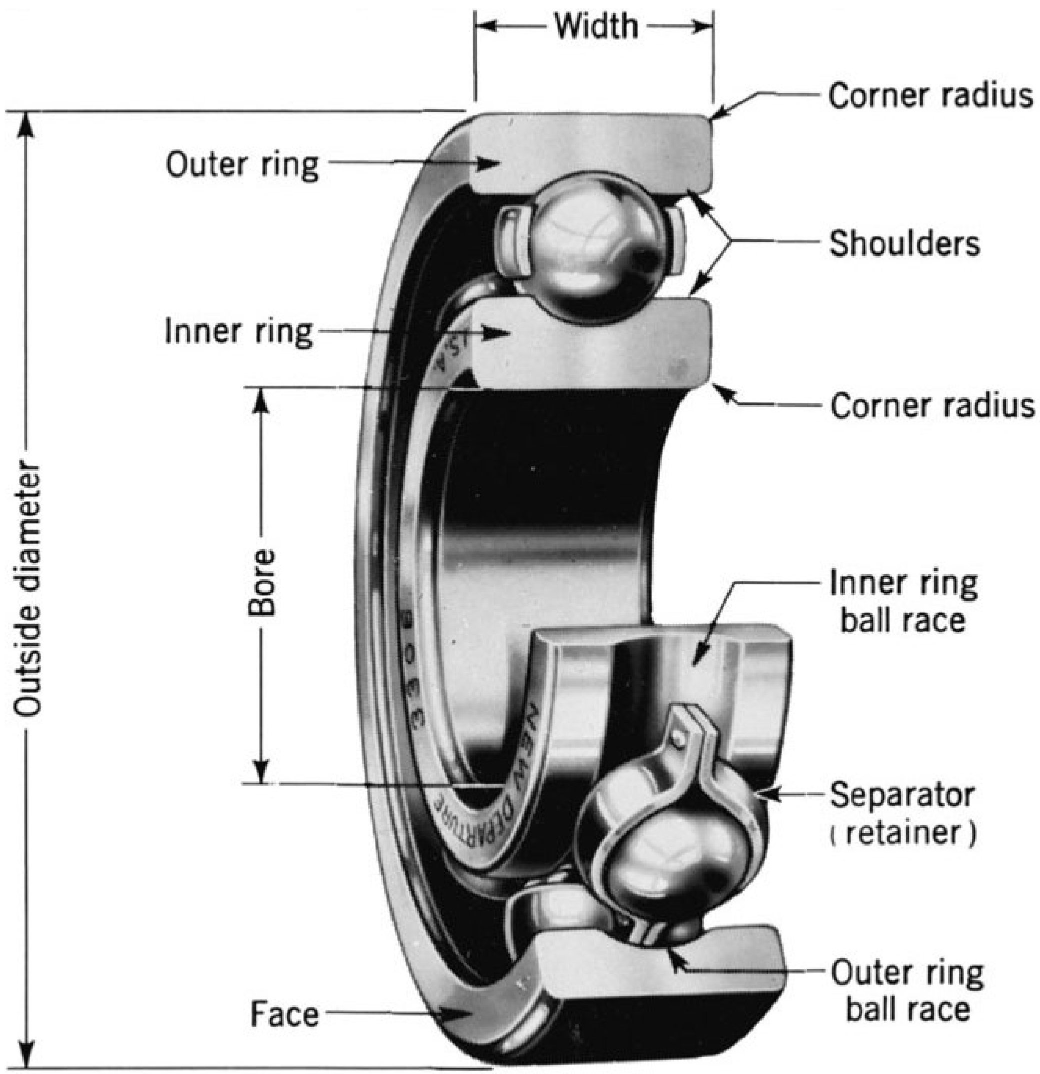
Plan for this week

Examples:

- Bearings
- Screws (with examples)

Rolling element bearings

Rolling element bearings



Designed to take:

- Pure radial loads
- Pure thrust loads
- Combination of the two kinds of loads

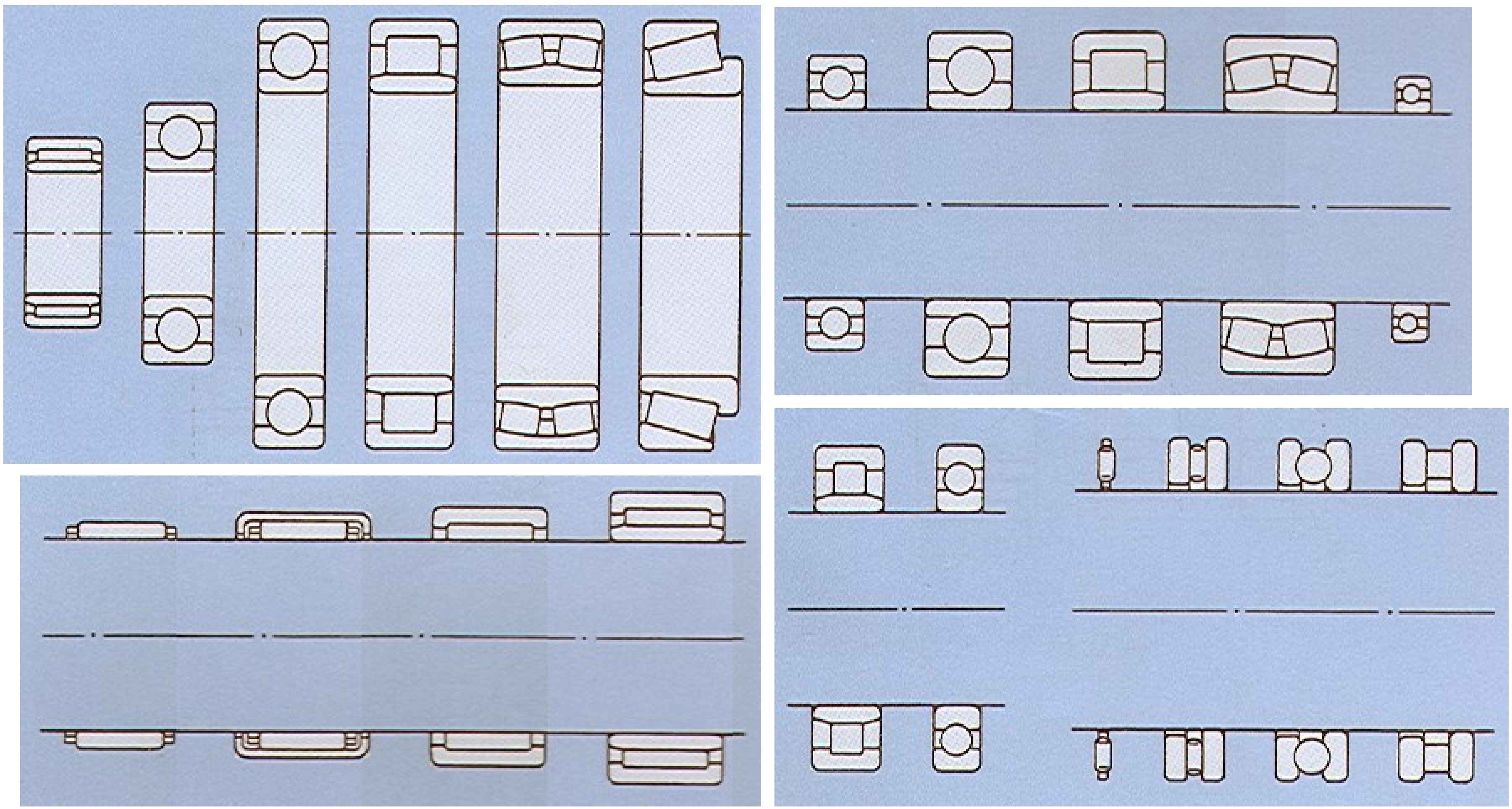
Main parts:

- Outer ring
- Inner ring
- Rolling elements (balls)
- Separator

Selection of bearings:

- Type and amount of load (axial - thrust, radial)
- Size, Speed
- Lubrication
- Life rating

Comparison of bearing types



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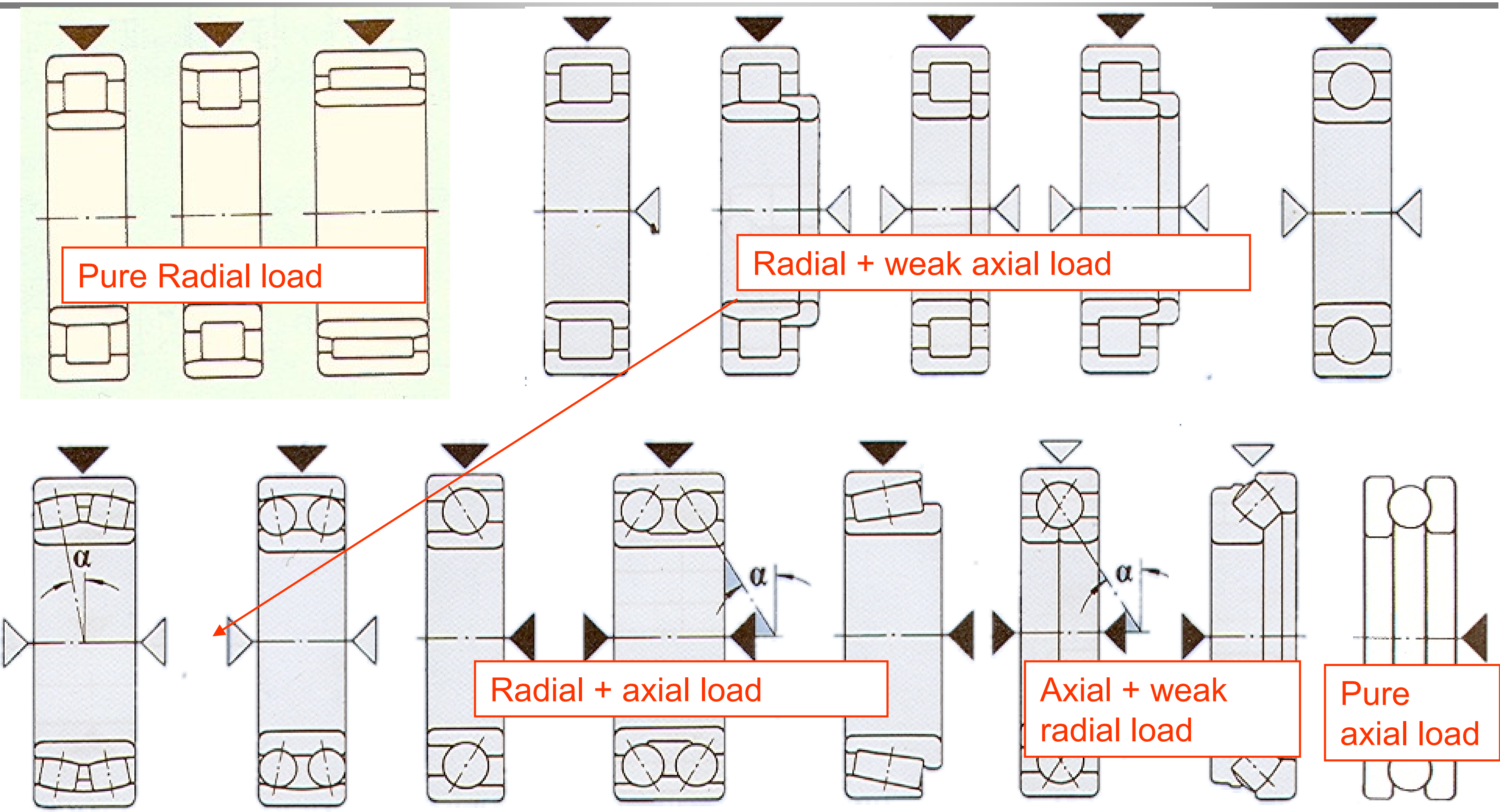
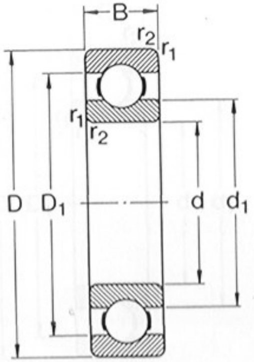
Forces that bearings can sustain

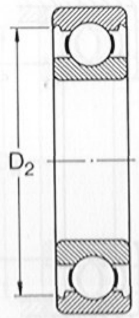


Deep groove ball bearings
single row
d 35–55 mm

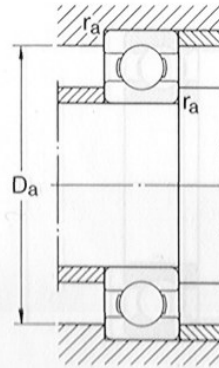
How to select a bearing from the catalogue



With full outer ring shoulders



With recessed outer ring shoulders



Principal dimensions			Basic load ratings		Fatigue load limit	Speed ratings		Mass	Designation
			dynamic	static		Lubrication grease	oil		
d	D	B	C	C_0	P_u				
mm			N		N	r/min		kg	–
35	47	7	4 750	3 200	166	13 000	16 000	0,030	**61807**
	55	10	9 560	6 200	290	11 000	14 000	0,080	**61907**
	62	9	12 400	8 150	375	10 000	13 000	0,11	**16007**
	62	14	15 900	10 200	440	10 000	13 000	0,16	**6007**
	72	17	25 500	15 300	655	9 000	11 000	0,29	**6207**
	80	21	33 200	19 000	815	8 500	10 000	0,46	**6307**
	100	25	55 300	31 000	1 290	7 000	8 500	0,95	**6407**
40	52	7	4 940	3 450	186	11 000	14 000	0,034	**61808**
	62	12	13 800	9 300	425	10 000	13 000	0,12	**61908**
	68	9	13 300	9 150	440	9 500	12 000	0,13	**16008**
	68	15	16 800	11 600	490	9 500	12 000	0,19	**6008**
	80	18	30 700	19 000	800	8 500	10 000	0,37	**6208**
	90	23	41 000	24 000	1 020	7 500	9 000	0,63	**6308**
	110	27	63 700	36 500	1 530	6 700	8 000	1,25	**6408**
45	58	7	6 050	4 300	228	9 500	12 000	0,040	**61809**
	68	12	14 000	9 800	465	9 000	11 000	0,14	**61909**
	75	10	15 600	10 800	520	9 000	11 000	0,17	**16009**
	75	16	20 800	14 600	640	9 000	11 000	0,25	**6009**
	85	19	33 200	21 600	915	7 500	9 000	0,41	**6209**
	100	25	52 700	31 500	1 340	6 700	8 000	0,83	**6309**
	120	29	76 100	45 000	1 900	6 000	7 000	1,55	**6409**

Dimensions					Abutment and fillet dimensions		
d	d_1 ≈	D_1 ≈	D_2 ≈	$r_{1,2}$ min	d_a min	D_a max	r_a max
mm					mm		
35	38,7	43,5	–	0,3	37	45	0,3
	41,6	48,6	–	0,6	39	51	0,6
	44	53,3	–	0,3	37	60	0,3
	43,7	53,6	55,7	1	40	57	1
	46,9	60,6	62,7	1,1	41,5	65,5	1
	49,5	66,1	69,2	1,5	43	72	1,5
	57,4	80,6	–	1,5	43	92	1,5
40	43,7	48,5	–	0,3	42	50	0,3
	47	55,2	–	0,6	44	58	0,6
	49,4	57	–	0,3	42	66	0,3
	49,2	59,1	61,1	1	45	63	1
	52,6	67,9	69,8	1,1	46,5	73,5	1
	56,1	74,7	77,7	1,5	48	82	1,5
	62,8	88	–	2	49	101	2
45	48,7	54,5	–	0,3	47	56	0,3
	52,3	60,8	–	0,6	49	64	0,6
	55	65,4	–	0,6	49	71	0,6
	54,7	65,6	67,8	1	50	70	1
	57,6	72,9	75,2	1,1	51,5	78,5	1
	62,1	83,7	86,7	1,5	53	92	1,5
	68,9	96,9	–	2	54	111	2

Bearing Life - Definitions

- Stresses: inner ring - rolling element + rolling element - outer ring.
- Metal fatigue is the only cause of failure for clean, properly lubricated, sealed and cooled bearings. (Dynamic loading)
- Endurance of a bearing is limiting factor – *bearing life* $\boldsymbol{L}$:
  - » Number of revolutions of the inner ring until the first evidence of fatigue.
  - » Number of hours of use at standard angular speed until the first evidence of fatigue
- *Rating life (minimum life)* of a bearing, $\boldsymbol{L}_{10}$
  - » *number of revolution or hours of operation that 90% of a group of identical bearings will achieve or exceed before the failure criterion develops.*
- Both previous life estimations are based on the *reliability factor*.
- The 'new' theory includes *fatigue load limit* $\boldsymbol{P}_u$ when estimating *the bearing life*

Bearing Life - Calculation

- *The* ***size*** *of a bearing* is initially selected on the relation of its ***load carrying capacity*** and ***the*** carried ***load*** with the ***life and reliability r***equirement.
- *Load carrying capacity* is specified for each bearing in a catalogue with:
  - » C – basic dynamic load rating – *for variable loads and high speeds*
  - » Co – basic static load rating - for static loads and low speeds
- *The load* calculated from free body diagrams or by other means.
- *Rating life* can be calculated from the life equation. The form of the life equation depends on the accuracy required. ***Basic rating life is***:

$$L_{10} = \left(\frac{C}{P}\right)^a \qquad [10^6\, rev]$$

$$L_{10h} = \frac{10^6}{60n} L_{10} \qquad [hours]$$

$$L_{10s} = \frac{\pi D}{1000} L_{10} \qquad [10^6\, km]$$

a = 3 - for ball bearings
a = 3.33- for roller bearings
P [N] – equivalent dynamic load rating
n [rpm] – rotational speed
D [m] - wheel diameter

Equivalent dynamic bearing load

$$\boxed{P = x F_r + y F_a}$$

P [N] - equivalent dynamic bearing load
F_r [N] – actual radial bearing load
F_a [N] – actual axial bearing load
x – radial load factor
y – axial load factor

Bearing type	Condition	x	y
Deep groove ball bearing	$F_a/F_r<=0.5$	1	0
	$F_a/F_r>0.5$	0.56	1-2
Self aligning ball bearings	$F_a/F_r<=e^*$	1	Y^*
	$F_a/F_r>e^*$	0.65	y^*
Angular contact ball bearings	$F_a/F_r<=1.14$	1	0
	$F_a/F_r>1.14$	0.35	0.57
Double row angular contact ball bearings	$F_a/F_r<=0.86$	1	0.73
	$F_a/F_r>0.86$	0.62	1.17
Four-point contact ball bearings	$F_a/F_r<=0.95$	1	0.66
	$F_a/F_r>0.95$	0.6	1.07
Cylindrical roller bearing (with flanges)	$F_a/F_r<=0.2$	1	0
	$F_a/F_r>0.2$	0.92	0.6
Needle roller bearings	-	1	0
Trust roller bearings	-	0	1
Taper roller bearings	$F_a/F_r<=e^*$	1	0
	$F_a/F_r>e^*$	0.4	Y^*
Taper roller bearings	1.00	0.75	0.60

Adjusting Bearing Life

- To adjust bearing life from rated to desired speed and life

$$P L_R^{1/a} = const.$$

$$L_R = 60 n L_{Rh}$$

$$P_R (60 L_R n_R)^{1/a} = P_D (60 L_D n_D)^{1/a}$$

- To adjust bearing life for different reliability

$$R_D = \exp\left[-\left(\frac{x_B - x_o}{\theta - x_o}\right)^b\right]$$

$$x = L / L_{10}$$

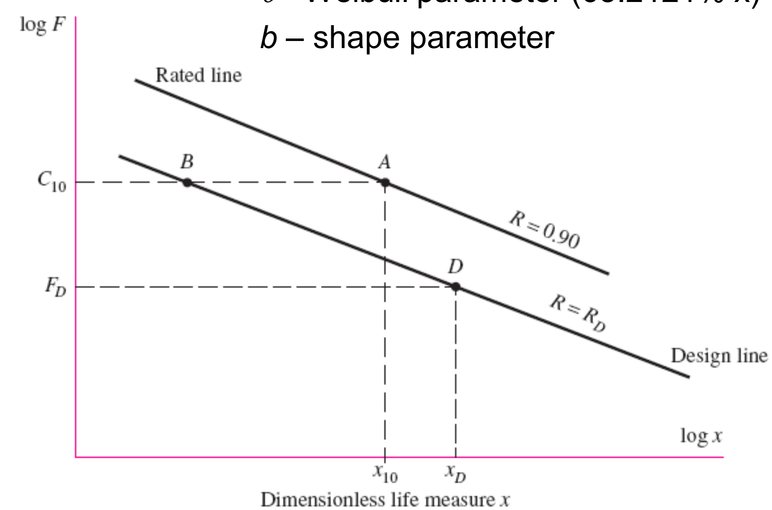
$$C_{10} = a_{oc} P_D \left[\frac{x_D}{x_0 + (\theta - x_0)(1 - R_D)^{1/b}}\right]^{1/a}$$

x – life measure variate

x_o – guaranteed variate

θ - Weibull parameter (63.2121% x)

b – shape parameter



Adjusted Bearing Life

- If the bearing is not operating in the ideal conditions then the basic rating life should be adjusted:

$$L_{adj} = a_T \, a_R \, a_{OC} \, L_{10} \qquad [10^6 \, rev]$$

Temperature [°C]	150	200	250	300
a_T	1.00	0.90	0.75	0.60

Reliability [%]	90	95	96	97	98	99
a_R	1.00	0.62	0.53	0.44	0.33	0.21

- a_{OC} – application factor (quality of lubrication and sealing).
  a_{OC} = 0.20 – 2.20
  Values depend on relative viscosity of lubricant.

Rearing Type	Direction of Load			Ratio of Load/Bulk			Misalignment Capacity		
	radial	axial	both	high	med	low	high	med	low
Thrust Ball		y			y				y
Deep Groove Ball	y		y		y			y	
Cylindrical Roller	y		certain types		y				y
Needle Roller	y			y					y
Tapered Roller	y	y	y		y				y
Self-aligning Ball	y		y			y	y		
Self-aligning Spherical Roller	y		y		y		y		
Angular Contact Ball		y	y			y			y

Machine Usage Type	Life Required of Bearings (Hours)
household appliances — intermittent use	300 - 3000
hand tools, construction equipment — short period use	3000 - 8000
lifts, cranes — high reliability for short periods	8000 - 12000
8h/day gears, motors — full day partial use	10000 - 25000
8h/day machine tools, fans — full day full use	20000 - 30000
continuous use	40000 - 50000

Example

Consider SKF, which rates its bearings for 1 million revolutions. If you desire a life of 5000 h at 1725 rev/min with a load of 400 lbf with a reliability of 90 percent, for which catalog rating would you search in an SKF catalog?

The rating life is $L_{10} = L_R = \mathcal{L}_R n_R 60 = 10^6$ revolutions. From Eq. (11–3),

$$C_{10} = F_D \left(\frac{\mathcal{L}_D n_D 60}{\mathcal{L}_R n_R 60} \right)^{1/a} = 400 \left[\frac{5000(1725)60}{10^6} \right]^{1/3} = 3211 \text{ lbf} = 14.3 \text{ kN}$$

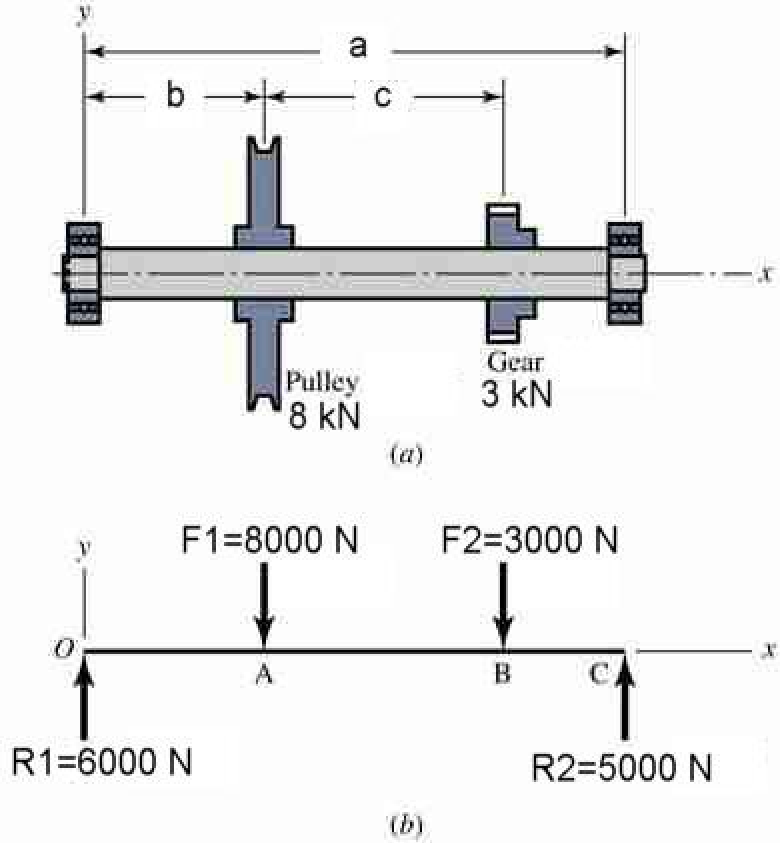
The design load on a ball bearing is 413 lbf and an application factor of 1.2 is appropriate. The speed of the shaft is to be 300 rev/min, the life to be 30 kh with a reliability of 0.99. What is the C_{10} catalog entry to be sought (or exceeded) when searching for a deep-groove bearing in a manufacturer's catalog on the basis of 10^6 revolutions for rating life? The Weibull parameters are $x_0 = 0.02$, $(\theta - x_0) = 4.439$, and $b = 1.483$.

$$x_D = \frac{L_D}{L_R} = \frac{60\mathcal{L}_D n_D}{L_{10}} = \frac{60(30\ 000)300}{10^6} = 540$$

Thus, the design life is 540 times the L_{10} life. For a ball bearing, $a = 3$. Then, from Eq. (11–7),

$$C_{10} = (1.2)(413) \left[\frac{540}{0.02 + 4.439(1 - 0.99)^{1/1.483}} \right]^{1/3} = 6696 \text{ lbf}$$

Example



Select the bearings and determine their rating life for the driving mechanism shown in the Figure. The shaft is 450 mm long and supported by deep-groove bearing in point *O* and plane roller bearing in point *C*. Assume minimum shaft diameter to be 20 mm. Mounted upon the shaft are a V-belt pulley, which contributes a radial load of F_1=8kN to the shaft, and a gear which contributes a radial load of F_2=3kN. The two loads are in the same plane and have the same direction. Minimum required bearing life is 2000 h with 90% reliability. Shaft rotates constantly at n=1000 rpm.

F1=8 kN	a=450 mm	c=200 mm
F2=3 kN	b=150 mm	d=20 mm

$L_{10h}=(L_{10h})_0=(L_{10h})_C=2000$ h n=1000 rpm

SOLUTION:

$$L_{10h}=\frac{10^6}{60n}\left(\frac{C}{P}\right)^a \Rightarrow C=P*\sqrt[a]{\frac{60n}{10^6}L_{10h}}$$

$$P_O=R_1=6000N \qquad P_C=R_2=5000N$$

$$C_0=6000*\sqrt[3]{\frac{60*1000}{10^6}2000}=29,595N$$

Selected from the catalogue for deep-groove ball bearings:
6404 20x72x19 mm C=30,700 N

$$C_0=5000*\sqrt[3.33]{\frac{60*1000}{10^6}2000}=21,025N$$

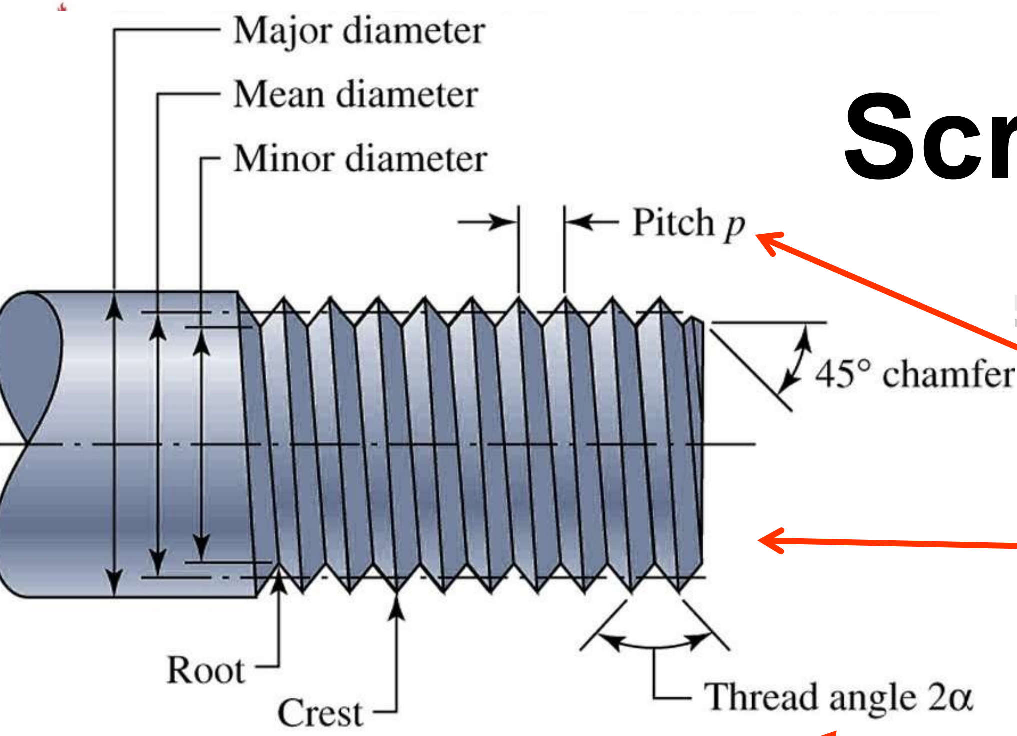
Selected from the catalogue for cylindrical roller bearings:
NU 204 20x47x14 mm C=25,100 N

Screws

and

Bolts

Screws and bolts



Pitch
distance between adjacent thread forms measured parallel to the thread axis.

Orientation
Threads are usually right-handed (RH) unless otherwise required

Thread height (H) $H = \frac{\sqrt{3}}{2} p$
For M and UN threads

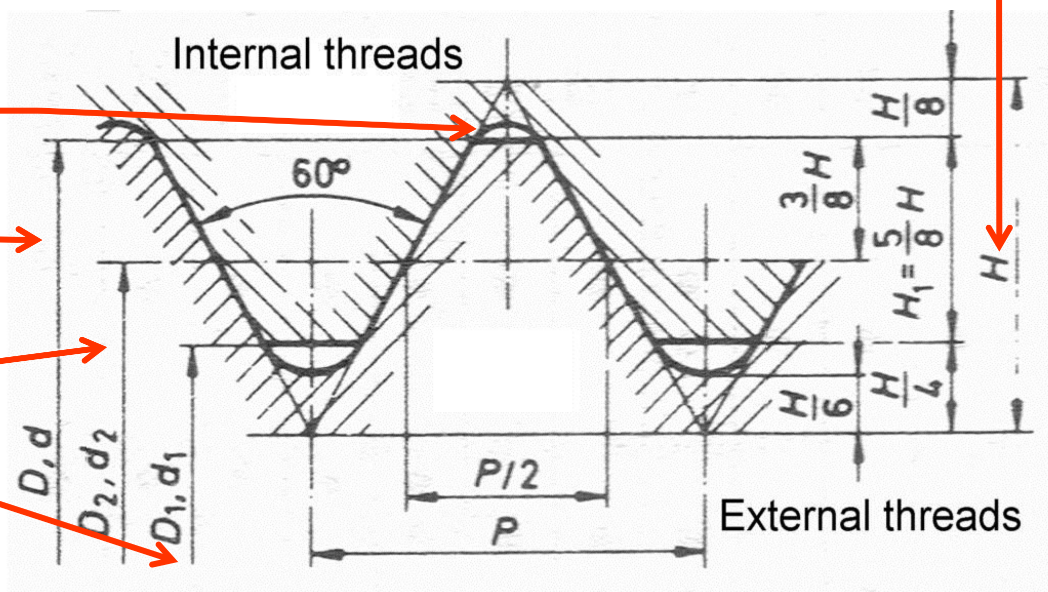
Thread angle
60o for M and UN threads

Crest of the thread
May be flat or round

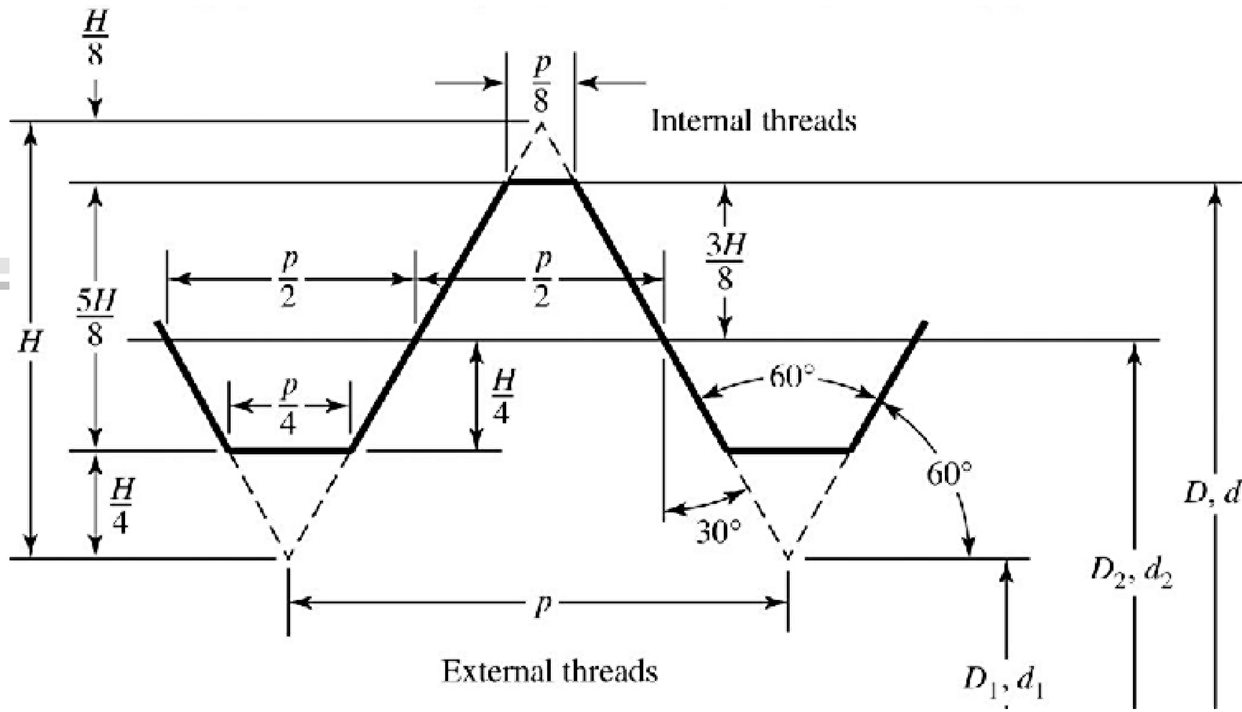
Major diameter (d, d_d)
Largest diameter of a screw thread

Mean–Pitch diameter (d_2, d_p)
Mean diameter; teeth section is p/2 long

Minor diameter (d_1, d_t)
Smalest diameter of the thread



Screw Threads



Metric Threads:

Thread angle = 60°

Symmetric profiles

Identified as M and MJ

Coarse and fine pitch

Specification of the thread:

M12 x 1.75

Pitch: coarse or fine

Nominal major diameter

The metric thread designation

Unified threads:

(usually pipe threads)

Thread angle = 60°

Symmetric profiles

Series UN and UNR

Coarse (C) and fine (F) pitch

Specification of the thread:

Thread series: UNC, UNF, UNRC, UNRF

Threads per inch (coarse or fine)

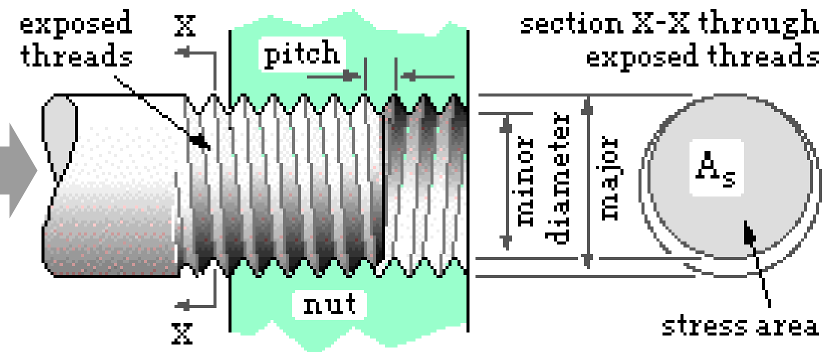
Nominal major diameter

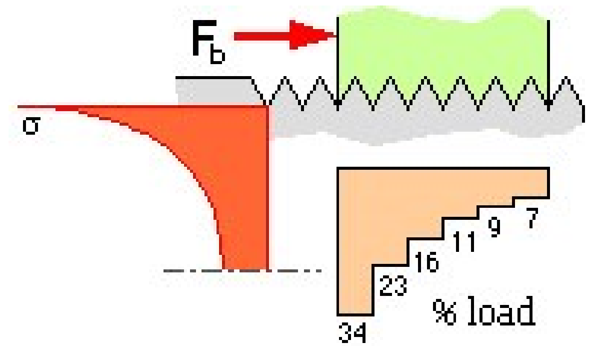
¼ in-20 UNRC

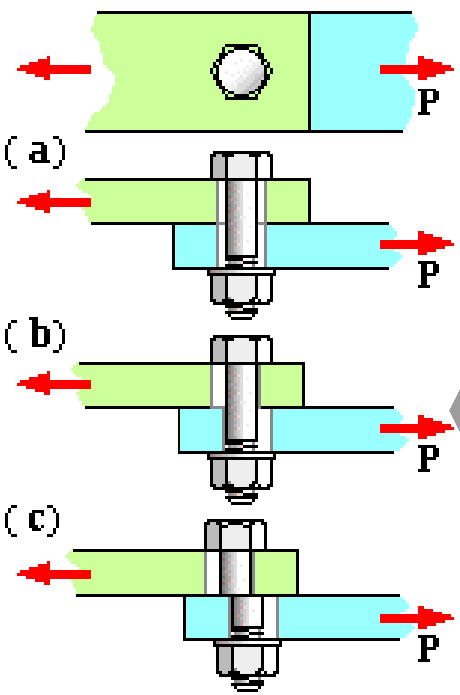
Load that a bolt can sustain

Tensile stress

$$\sigma = {F_b}/{A_t}$$







Shear stress

$$\tau = {P}/{A_d}$$

Metric threads

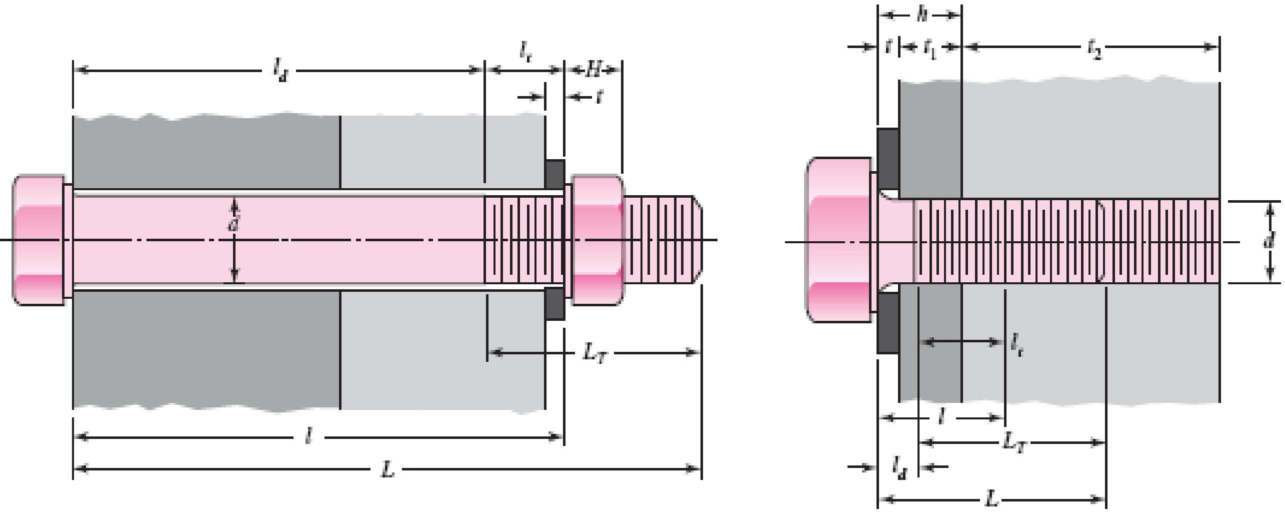
(all dimensions in mm)

Nominal Major Diameter d	Coarse-Pitch Series Pitch p	Coarse-Pitch Series Tensile-Stress Area A_t	Coarse-Pitch Series Minor-Diameter Area A_r	Fine-Pitch Series Pitch p	Fine-Pitch Series Tensile-Stress Area A_t	Fine-Pitch Series Minor-Diameter Area A_r
1.6	0.35	1.27	1.07			
2	0.40	2.07	1.79			
2.5	0.45	3.39	2.98			
3	0.5	5.03	4.47			
3.5	0.6	6.78	6.00			
4	0.7	8.78	7.75			
5	0.8	14.2	12.7			
6	1	20.1	17.9			
8	1.25	36.6	32.8	1	39.2	36.0
10	1.5	58.0	52.3	1.25	61.2	56.3
12	1.75	84.3	76.3	1.25	92.1	86.0
14	2	115	104	1.5	125	116
16	2	157	144	1.5	167	157
20	2.5	245	225	1.5	272	259
24	3	353	324	2	384	365
30	3.5	561	519	2	621	596
36	4	817	759	2	915	884
42	4.5	1120	1050	2	1260	1230
48	5	1470	1380	2	1670	1630
56	5.5	2030	1910	2	2300	2250
64	6	2680	2520	2	3030	2980
72	6	3460	3280	2	3860	3800
80	6	4340	4140	1.5	4850	4800
90	6	5590	5360	2	6100	6020
100	6	6990	6740	2	7560	7470
110				2	9180	9080

Property classes for screws

Metric Mechanical-Property Classes for Steel Bolts, Screws, and Studs*

Property Class	Size Range, Inclusive	Minimum Proof Strength,† MPa	Minimum Tensile Strength,† MPa	Minimum Yield Strength,† MPa	Material	Head Marking
4.6	M5–M36	225	400	240	Low or medium carbon	4.6
4.8	M1.6–M16	310	420	340	Low or medium carbon	4.8
5.8	M5–M24	380	520	420	Low or medium carbon	5.8
8.8	M16–M36	600	830	660	Medium carbon, Q&T	8.8
9.8	M1.6–M16	650	900	720	Medium carbon, Q&T	9.8
10.9	M5–M36	830	1040	940	Low-carbon martensite, Q&T	10.9
12.9	M1.6–M36	970	1220	1100	Alloy, Q&T	12.9



(a) (b)

Joint-Fastener stiffness

Washer thickness: t from Table A–32 or A–33

Nut thickness [Fig. (a) only]: H from Table A–31

Grip length:

For Fig. (a): l = thickness of all material squeezed between face of bolt and face of nut

For Fig. (b): $l = \begin{cases} h + t_2/2, & t_2 < d \\ h + d/2, & t_2 \geq d \end{cases}$

Fastener length (round up using Table A–17*):

For Fig. (a): $L > l + H$

For Fig. (b): $L > h + 1.5d$

Threaded length L_T: Inch series:

$$L_T = \begin{cases} 2d + \frac{1}{4}\ \text{in}, & L \leq 6\ \text{in} \\ 2d + \frac{1}{2}\ \text{in}, & L > 6\ \text{in} \end{cases}$$

Metric series:

$$L_T = \begin{cases} 2d + 6\ \text{mm}, & L \leq 125\ \text{mm}, d \leq 48\ \text{mm} \\ 2d + 12\ \text{mm}, & 125 < L \leq 200\ \text{mm} \\ 2d + 25\ \text{mm}, & L > 200\ \text{mm} \end{cases}$$

Length of unthreaded portion in grip: $l_d = L - L_T$

Length of threaded portion in grip: $l_t = l - l_d$

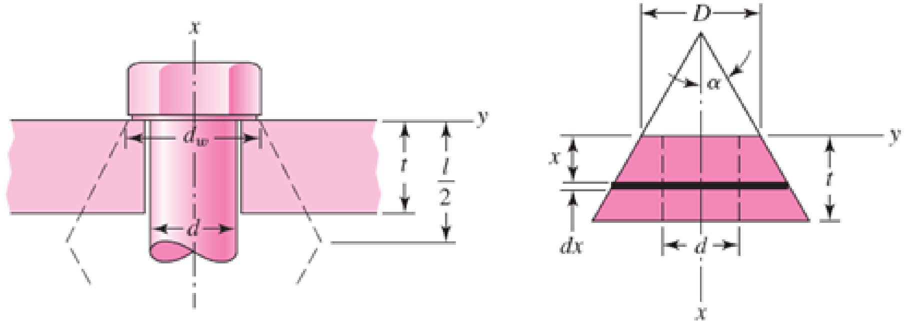
Area of unthreaded portion: $A_d = \pi d^2/4$

Area of threaded portion: A_t from Table 8–1 or 8–2

Fastener stiffness: $k_b = \dfrac{A_d A_t E}{A_d l_t + A_t l_d}$

Joint-Members stiffness

Compression of a member with the equivalent elastic properties represented by a frustum of a hollow cone. Here, l represents the grip length.



(a) (b)

$$k = \frac{0.5774\pi E d}{\ln \dfrac{(1.155t + D - d)(D + d)}{(1.155t + D + d)(D - d)}}$$

$$\frac{1}{k_m} = \frac{1}{k_1} + \frac{1}{k_2} + \frac{1}{k_3} + \cdots$$

$$\frac{k_m}{Ed} = A\exp(Bd/l)$$

Stiffness of the joint $C = \dfrac{k_b}{k_b + k_m}$

Load on the bolt

$$F_b = P_b + F_i = CP + F_i \qquad F_m < 0$$

$$F_m = P_m - F_i = (1 - C)P - F_i \qquad F_m < 0$$

Load on a member

Stiffness Parameters of Various Member Materials

J. Wileman, M. Choudury, and I. Green, "Computation of Member Stiffness in Bolted Connections,"

Material Used	Poisson Ratio	Elastic GPa	Modulus Mpsi	A	B
Steel	0.291	207	30.0	0.787 15	0.628 73
Aluminum	0.334	71	10.3	0.796 70	0.638 16
Copper	0.326	119	17.3	0.795 68	0.635 53
Gray cast iron	0.211	100	14.5	0.778 71	0.616 16
General expression				0.789 52	0.629 14

Statically loaded joint with Preload

Recommended Preload

$$F_i = \begin{cases} 0.75F_p & \text{for nonpermanent connections, reused fasteners} \\ 0.90F_p & \text{for permanent connections} \end{cases}$$

Proof load

$$F_p = A_t S_p$$

Load factor per a bolt (ratio of proof load and operating load)

$$n_L = \frac{S_p A_t - F_i}{CP}$$

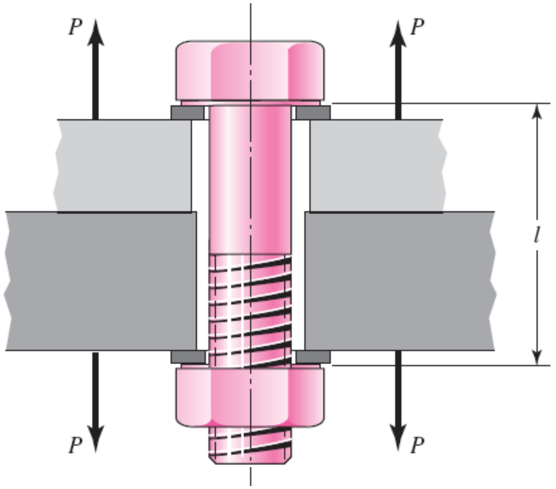
$$P = \frac{P_{total}}{N}$$

Number of bolts

$$N = \frac{C n_L P_{total}}{S_p A_t - F_i}$$

Yielding factor of safety

$$n_p = \frac{S_p A_t}{CP + F_i}$$



Factor of safety against joint separation

$$n_0 = \frac{P_0}{P} \qquad n_0 = \frac{F_i}{P(1-C)}$$

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Table A-8

Properties of Round Tubing

w_a = unit weight of aluminum tubing, lbf/ft
w_s = unit weight of steel tubing, lbf/ft
m = unit mass, kg/m
A = area, in^2 (cm^2)
I = second moment of area, in^4 (cm^4)
J = second polar moment of area, in^4 (cm^4)
k = radius of gyration, in (cm)
Z = section modulus, in^3 (cm^3)
d, t = size (OD) and thickness, in (mm)

From: Shigley's Mechanical Engineering Design, 9th edition

Size, in	w_a	w_s	A	I	k	Z	J
$1 \times \frac{1}{8}$	0.416	1.128	0.344	0.034	0.313	0.067	0.067
$1 \times \frac{1}{4}$	0.713	2.003	0.589	0.046	0.280	0.092	0.092
$1\frac{1}{2} \times \frac{1}{8}$	0.653	1.769	0.540	0.129	0.488	0.172	0.257
$1\frac{1}{2} \times \frac{1}{4}$	1.188	3.338	0.982	0.199	0.451	0.266	0.399
$2 \times \frac{1}{8}$	0.891	2.670	0.736	0.325	0.664	0.325	0.650
$2 \times \frac{1}{4}$	1.663	4.673	1.374	0.537	0.625	0.537	1.074
$2\frac{1}{2} \times \frac{1}{8}$	1.129	3.050	0.933	0.660	0.841	0.528	1.319
$2\frac{1}{2} \times \frac{1}{4}$	2.138	6.008	1.767	1.132	0.800	0.906	2.276
$3 \times \frac{1}{4}$	2.614	7.343	2.160	2.059	0.976	1.373	4.117
$3 \times \frac{3}{8}$	3.742	10.51	3.093	2.718	0.938	1.812	5.436
$4 \times \frac{3}{16}$	2.717	7.654	2.246	4.090	1.350	2.045	8.180
$4 \times \frac{3}{8}$	5.167	14.52	4.271	7.090	1.289	3.544	14.180

Size, mm	m	A	I	k	Z	J
12×2	0.490	0.628	0.082	0.361	0.136	0.163
16×2	0.687	0.879	0.220	0.500	0.275	0.440
16×3	0.956	1.225	0.273	0.472	0.341	0.545
20×4	1.569	2.010	0.684	0.583	0.684	1.367
25×4	2.060	2.638	1.508	0.756	1.206	3.015
25×5	2.452	3.140	1.669	0.729	1.336	3.338
30×4	2.550	3.266	2.827	0.930	1.885	5.652
30×5	3.065	3.925	3.192	0.901	2.128	6.381
42×4	3.727	4.773	8.717	1.351	4.151	17.430
42×5	4.536	5.809	10.130	1.320	4.825	20.255
50×4	4.512	5.778	15.409	1.632	6.164	30.810
50×5	5.517	7.065	18.118	1.601	7.247	36.226

Table A–17

Preferred Sizes and Renard (R-Series) Numbers

(When a choice can be made, use one of these sizes; however, not all parts or items are available in all the sizes shown in the table.)

Fraction of Inches

$\frac{1}{64}$, $\frac{1}{32}$, $\frac{1}{16}$, $\frac{3}{32}$, $\frac{1}{8}$, $\frac{5}{32}$, $\frac{3}{16}$, $\frac{1}{4}$, $\frac{5}{16}$, $\frac{3}{8}$, $\frac{7}{16}$, $\frac{1}{2}$, $\frac{9}{16}$, $\frac{5}{8}$, $\frac{11}{16}$, $\frac{3}{4}$, $\frac{7}{8}$, 1, $1\frac{1}{4}$, $1\frac{1}{2}$, $1\frac{3}{4}$, 2, $2\frac{1}{4}$, $2\frac{1}{2}$, $2\frac{3}{4}$, 3, $3\frac{1}{4}$, $3\frac{1}{2}$, $3\frac{3}{4}$, 4, $4\frac{1}{4}$, $4\frac{1}{2}$, $4\frac{3}{4}$, 5, $5\frac{1}{4}$, $5\frac{1}{2}$, $5\frac{3}{4}$, 6, $6\frac{1}{2}$, 7, $7\frac{1}{2}$, 8, $8\frac{1}{2}$, 9, $9\frac{1}{2}$, 10, $10\frac{1}{2}$, 11, $11\frac{1}{2}$, 12, $12\frac{1}{2}$, 13, $13\frac{1}{2}$, 14, $14\frac{1}{2}$, 15, $15\frac{1}{2}$, 16, $16\frac{1}{2}$, 17, $17\frac{1}{2}$, 18, $18\frac{1}{2}$, 19, $19\frac{1}{2}$, 20

Decimal Inches

0.010, 0.012, 0.016, 0.020, 0.025, 0.032, 0.040, 0.05, 0.06, 0.08, 0.10, 0.12, 0.16, 0.20, 0.24, 0.30, 0.40, 0.50, 0.60, 0.80, 1.00, 1.20, 1.40, 1.60, 1.80, 2.0, 2.4, 2.6, 2.8, 3.0, 3.2, 3.4, 3.6, 3.8, 4.0, 4.2, 4.4, 4.6, 4.8, 5.0, 5.2, 5.4, 5.6, 5.8, 6.0, 7.0, 7.5, 8.5, 9.0, 9.5, 10.0, 10.5, 11.0, 11.5, 12.0, 12.5, 13.0, 13.5, 14.0, 14.5, 15.0, 15.5, 16.0, 16.5, 17.0, 17.5, 18.0, 18.5, 19.0, 19.5, 20

Millimeters

0.05, 0.06, 0.08, 0.10, 0.12, 0.16, 0.20, 0.25, 0.30, 0.40, 0.50, 0.60, 0.70, 0.80, 0.90, 1.0, 1.1, 1.2, 1.4, 1.5, 1.6, 1.8, 2.0, 2.2, 2.5, 2.8, 3.0, 3.5, 4.0, 4.5, 5.0, 5.5, 6.0, 6.5, 7.0, 8.0, 9.0, 10, 11, 12, 14, 16, 18, 20, 22, 25, 28, 30, 32, 35, 40, 45, 50, 60, 80, 100, 120, 140, 160, 180, 200, 250, 300

Renard Numbers*

1st choice, R5: 1, 1.6, 2.5, 4, 6.3, 10

2d choice, R10: 1.25, 2, 3.15, 5, 8

3d choice, R20: 1.12, 1.4, 1.8, 2.24, 2.8, 3.55, 4.5, 5.6, 7.1, 9

4th choice, R40: 1.06, 1.18, 1.32, 1.5, 1.7, 1.9, 2.12, 2.36, 2.65, 3, 3.35, 3.75, 4.25, 4.75, 5.3, 6, 6.7, 7.5, 8.5, 9.5

*May be multiplied or divided by powers of 10.

From: Shigley's Mechanical Engineering Design, 9th edition

Table A–31

Dimensions of Hexagonal Nuts

Nominal Size, in	Width W	Height H Regular Hexagonal	Thick or Slotted	JAM
$\frac{1}{4}$	$\frac{7}{16}$	$\frac{7}{32}$	$\frac{9}{32}$	$\frac{5}{32}$
$\frac{5}{16}$	$\frac{1}{2}$	$\frac{17}{64}$	$\frac{21}{64}$	$\frac{3}{16}$
$\frac{3}{8}$	$\frac{9}{16}$	$\frac{21}{64}$	$\frac{13}{32}$	$\frac{7}{32}$
$\frac{7}{16}$	$\frac{11}{16}$	$\frac{3}{8}$	$\frac{29}{64}$	$\frac{1}{4}$
$\frac{1}{2}$	$\frac{3}{4}$	$\frac{7}{16}$	$\frac{9}{16}$	$\frac{5}{16}$
$\frac{9}{16}$	$\frac{7}{8}$	$\frac{31}{64}$	$\frac{39}{64}$	$\frac{5}{16}$
$\frac{5}{8}$	$\frac{15}{16}$	$\frac{35}{64}$	$\frac{23}{32}$	$\frac{3}{8}$
$\frac{3}{4}$	$1\frac{1}{8}$	$\frac{41}{64}$	$\frac{13}{16}$	$\frac{27}{64}$
$\frac{7}{8}$	$1\frac{5}{16}$	$\frac{3}{4}$	$\frac{29}{32}$	$\frac{31}{64}$
1	$1\frac{1}{2}$	$\frac{55}{64}$	1	$\frac{35}{64}$
$1\frac{1}{8}$	$1\frac{11}{16}$	$\frac{31}{32}$	$1\frac{5}{32}$	$\frac{39}{64}$
$1\frac{1}{4}$	$1\frac{7}{8}$	$1\frac{1}{16}$	$1\frac{1}{4}$	$\frac{23}{32}$
$1\frac{3}{8}$	$2\frac{1}{16}$	$1\frac{11}{64}$	$1\frac{3}{8}$	$\frac{25}{32}$
$1\frac{1}{2}$	$2\frac{1}{4}$	$1\frac{9}{32}$	$1\frac{1}{2}$	$\frac{27}{32}$
Nominal Size, mm				
M5	8	4.7	5.1	2.7
M6	10	5.2	5.7	3.2
M8	13	6.8	7.5	4.0
M10	16	8.4	9.3	5.0
M12	18	10.8	12.0	6.0
M14	21	12.8	14.1	7.0
M16	24	14.8	16.4	8.0
M20	30	18.0	20.3	10.0
M24	36	21.5	23.9	12.0
M30	46	25.6	28.6	15.0
M36	55	31.0	34.7	18.0

Table A–33

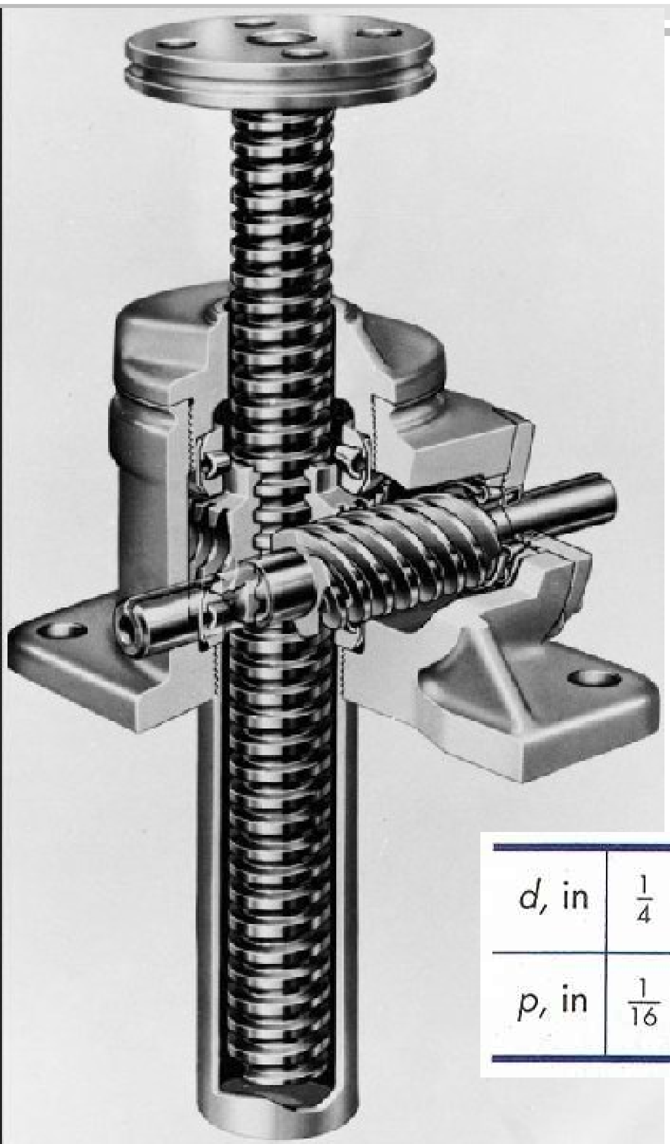
Dimensions of Metric Plain Washers (All Dimensions in Millimeters)

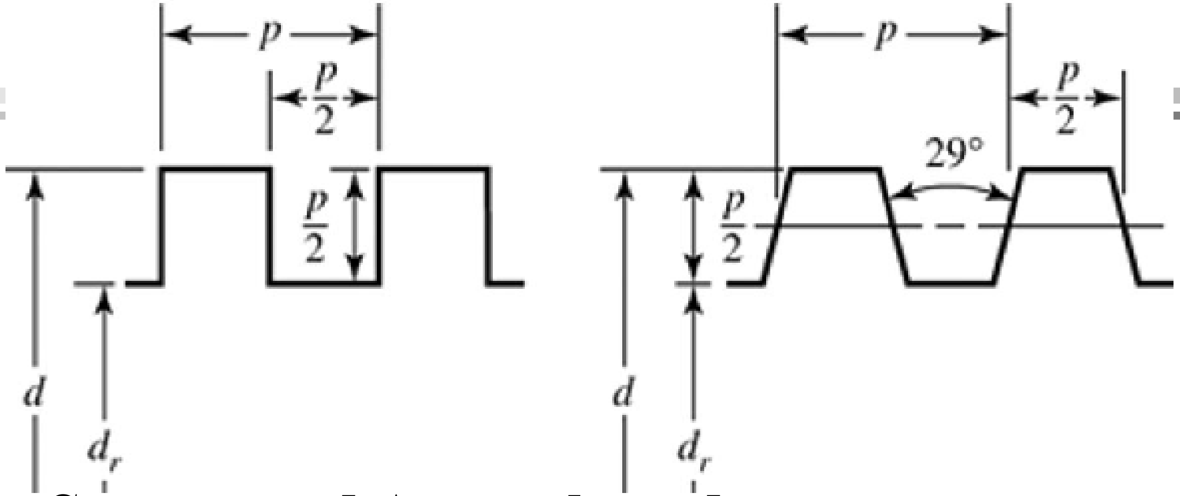
Washer Size*	Minimum ID	Maximum OD	Maximum Thickness	Washer Size*	Minimum ID	Maximum OD	Maximum Thickness
1.6 N	1.95	4.00	0.70	10 N	10.85	20.00	2.30
1.6 R	1.95	5.00	0.70	10 R	10.85	28.00	2.80
1.6 W	1.95	6.00	0.90	10 W	10.85	39.00	3.50
2 N	2.50	5.00	0.90	12 N	13.30	25.40	2.80
2 R	2.50	6.00	0.90	12 R	13.30	34.00	3.50
2 W	2.50	8.00	0.90	12 W	13.30	44.00	3.50
2.5 N	3.00	6.00	0.90	14 N	15.25	28.00	2.80
2.5 R	3.00	8.00	0.90	14 R	15.25	39.00	3.50
2.5 W	3.00	10.00	1.20	14 W	15.25	50.00	4.00
3 N	3.50	7.00	0.90	16 N	17.25	32.00	3.50
3 R	3.50	10.00	1.20	16 R	17.25	44.00	4.00
3 W	3.50	12.00	1.40	16 W	17.25	56.00	4.60
3.5 N	4.00	9.00	1.20	20 N	21.80	39.00	4.00
3.5 R	4.00	10.00	1.40	20 R	21.80	50.00	4.60
3.5 W	4.00	15.00	1.75	20 W	21.80	66.00	5.10
4 N	4.70	10.00	1.20	24 N	25.60	44.00	4.60
4 R	4.70	12.00	1.40	24 R	25.60	56.00	5.10
4 W	4.70	16.00	2.30	24 W	25.60	72.00	5.60
5 N	5.50	11.00	1.40	30 N	32.40	56.00	5.10
5 R	5.50	15.00	1.75	30 R	32.40	72.00	5.60
5 W	5.50	20.00	2.30	30 W	32.40	90.00	6.40
6 N	6.65	13.00	1.75	36 N	38.30	66.00	5.60
6 R	6.65	18.80	1.75	36 R	38.30	90.00	6.40
6 W	6.65	25.40	2.30	36 W	38.30	110.00	8.50
8 N	8.90	18.80	2.30				
8 R	8.90	25.40	2.30				
8 W	8.90	32.00	2.80				

N = narrow; R = regular; W = wide.

*Same as screw or bolt size.

Power screws





Square and Acme threads:

Used for power transmission

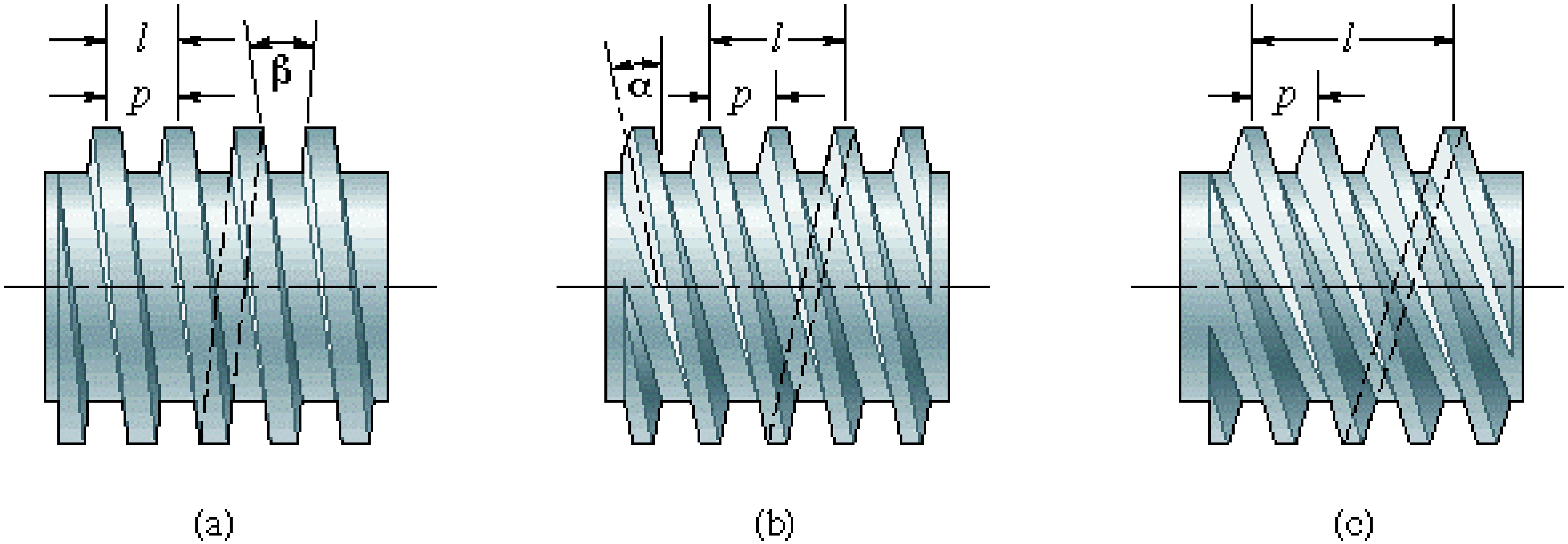
These have preferred sizes but also can vary

Modifications to these threads are easy

Preferred Pitches for power threads:

d, in	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{8}$	1	$1\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{3}{4}$	2	$2\frac{1}{2}$	3
p, in	$\frac{1}{16}$	$\frac{1}{14}$	$\frac{1}{12}$	$\frac{1}{10}$	$\frac{1}{8}$	$\frac{1}{6}$	$\frac{1}{6}$	$\frac{1}{5}$	$\frac{1}{5}$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{3}$	$\frac{1}{2}$

Multiple threaded screws

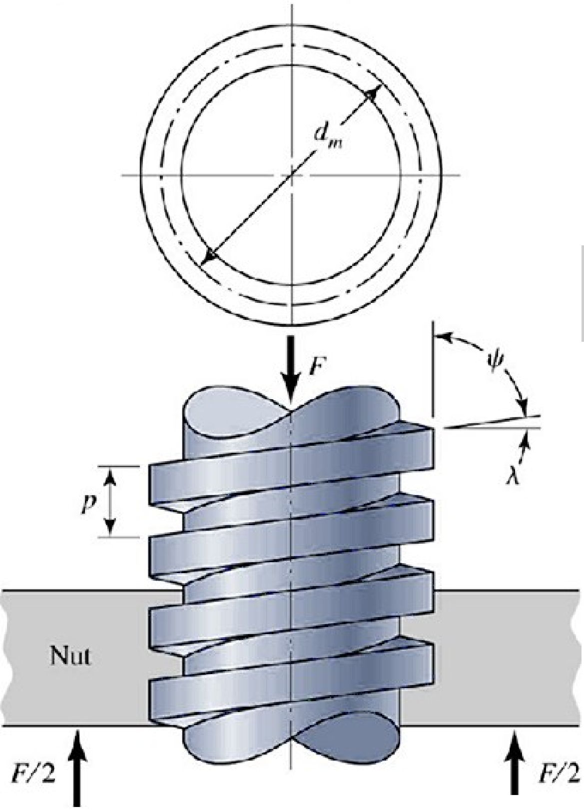


l=p

l=2p

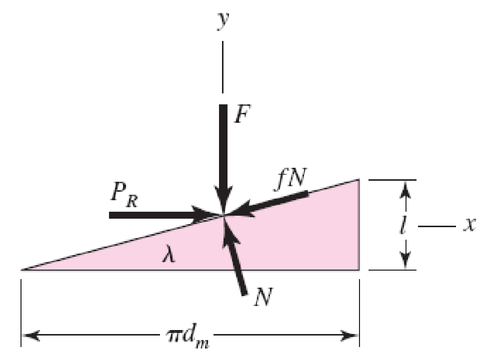
l=3p

(a) Single, (b) double, (c) triple threaded screws.



Power screws

Rising the load

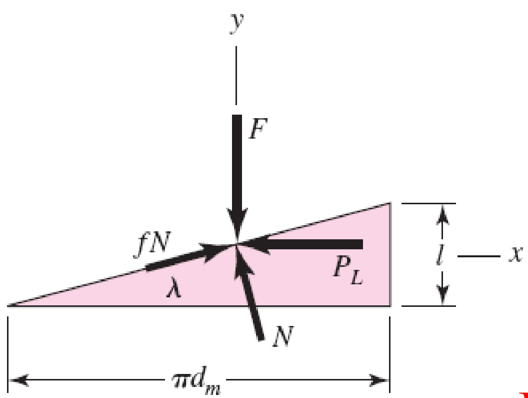


(a)

$$\sum F_x = P_R - N\sin\lambda - f\,N\cos\lambda$$

$$\sum F_y = F + f\,N\sin\lambda - N\cos\lambda$$

$$P_R = \frac{F(f\cos\lambda + \sin\lambda)}{\cos\lambda - f\sin\lambda} \qquad T_R = \frac{Fd_m}{2}\left(\frac{\pi f d_m + l}{\pi d_m - fl}\right)$$



(b)

Lowering the load

$$\sum F_x = -P_L - N\sin\lambda + f\,N\cos\lambda$$

$$\sum F_y = F - f\,N\sin\lambda - N\cos\lambda$$

$$P_L = \frac{F(f\cos\lambda - \sin\lambda)}{\cos\lambda + f\sin\lambda} \qquad T_L = \frac{Fd_m}{2}\left(\frac{\pi f d_m - l}{\pi d_m + fl}\right)$$

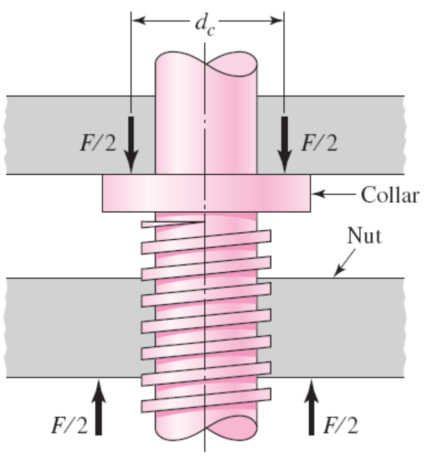
Efficiency of thread

$$e = \frac{T_0}{T_R} = \frac{Fl}{2\pi T_R}$$

Condition for self locking

$$\pi f d_m > l \qquad f > \tan\lambda$$

Stress and Strength of power screws



(b)

$$T_c = \frac{F f_c d_c}{2}$$

Torque due to friction in collar

$$\tau = \frac{16T}{\pi d_r^3} \qquad \sigma = \frac{F}{A} = \frac{4F}{\pi d_r^2}$$

Shear and Axial stress in the body of the screw

$$\sigma_B = -\frac{F}{\pi d_m n_t p/2} = -\frac{2F}{\pi d_m n_t p}$$

Bearing stress

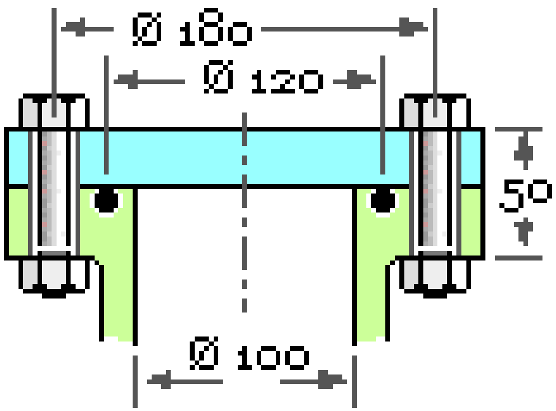
$$\sigma_b = \frac{M}{Z} = \frac{F p}{4} \frac{24}{\pi d_r n_t p^2} = \frac{6F}{\pi d_r n_t p}$$

Bending stress at the root of the thread

$c=d/2$ - maximum span
$I=\pi d^4/64$ - second moment of area
$Z=c/I$ - section modulus

Example

The cover of a pressurised cylinder is attached by a self-energising seal and 6 identical bolts M10x1.5 of class 8.8. The fluid pressure is essentially constant at 6 MPa. A safety factor of three is required. Check if the given bolt can sustain the pressure!



P=6MPa 6 class 8.8 M10x1.5

d_s=120 mm N_d=3

S_t/σ=?

- **SOLUTION:**

- Force on the cover caused by the pressure: $F_c = p \cdot A_s = p\dfrac{\pi d_s^2}{4}$ $F_c = 6\cdot 10^6 \dfrac{\pi \cdot 0.12^2}{4} = 67858N = 67.9kN$
- Force on the individual bolt $F_b = {F_c}/{6} = {67.9}/{6}$ $F_b = 11.3\,kN$
- From tables: Tensile stress area $A_t = 58mm^2$ Proof strength $S_p = 590\,MPa$
- Stress on each bolt: $\sigma = {F_b}/{A_t} = {11300}/{58}$ $\sigma = 194MPa$

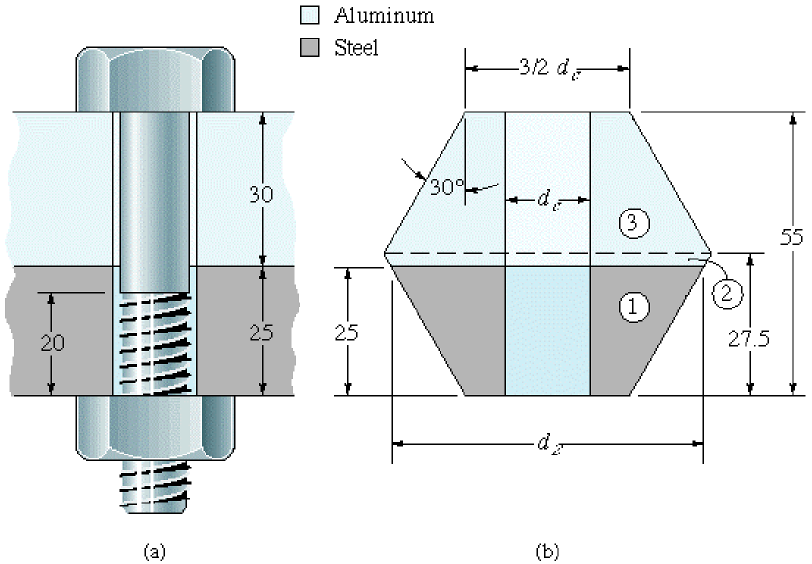
$$\frac{S_p}{\sigma} = \frac{590}{194} = 3.04 \approx N_d$$

- **Selected number of bolts can sustain the load**

Example 5 – Joint stiffness

A hexagonal bolt and nut assembly is used to join two members. The bolt and the nut are made of steel and the frustum cone angle is 30°. The tread crest diameter is 14 mm and the root diameter is 12 mm.

Use the table from lectures for material properties.



Find the bolt and joint stiffness.

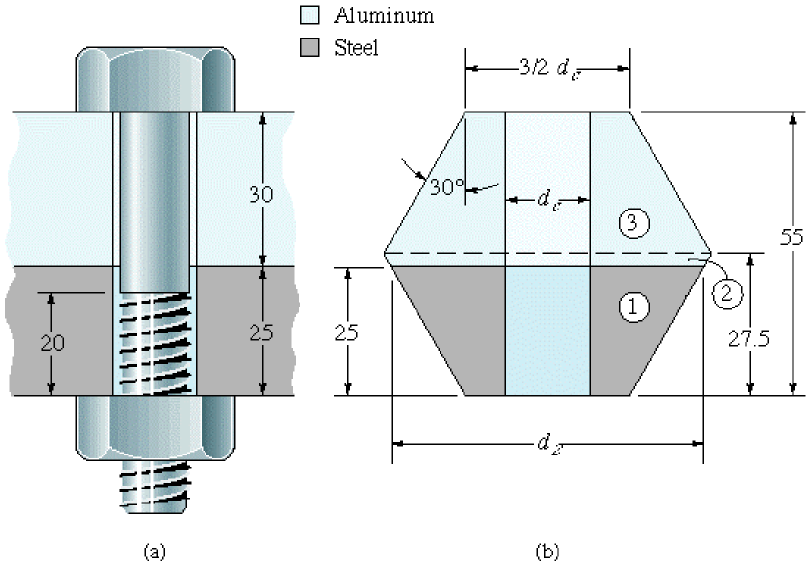
Result:

k_b=4.702e^8 N/m; k_m=12.38e^8 N/m
C=0.2753

Example 6 – Bolt preload

A hexagonal bolt and nut assembly is used to join two members. The bolt and the nut are made of steel and the frustum cone angle is 30°. The tread crest diameter is 14 mm and the root diameter is 12 mm.

Use the table from lectures for material properties.



Use calculated results k_b=4.702e^8 N/m; k_m=12.38e^8 N/m C=0.2753

Determine the maximum load for bolt-joint failure while assuming the non permanent connection and a static safety factor of 2.5. Grade 5.8 coarse thread is used.

P_i=34400 N, $P_{max,b}$=13111 N, Pmax,j=21163 N.