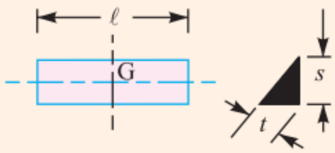
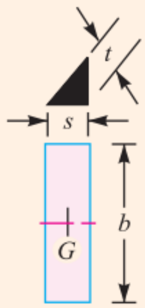
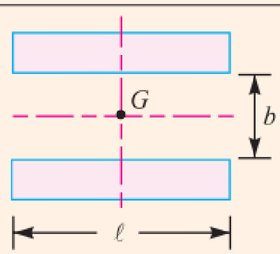
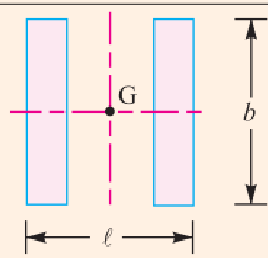
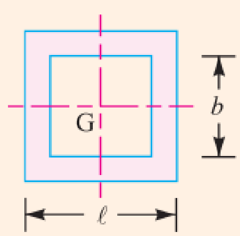
Table 10.7. Polar moment of inertia and section modulus of welds.

S.No	Type of weld	Polar moment of inertia (J)	Section modulus (Z)
1.		$\frac{t.l^3}{12}$	—
2.		$\frac{t.b^3}{12}$	$\frac{t.b^2}{6}$
3.		$\frac{t.l(3b^2+l^2)}{6}$	$t.b.l$
4.		$\frac{t.b(b^2+3l^2)}{6}$	$\frac{t.b^2}{3}$
5.		$\frac{t(b+l)^3}{6}$	$t\left(b.l+\frac{b^2}{3}\right)$

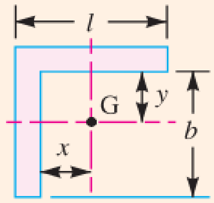
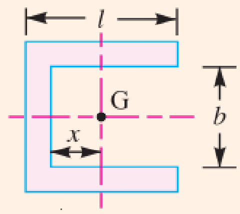
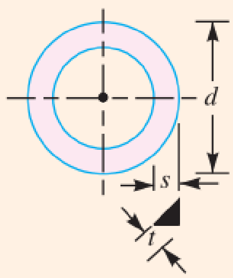
S.No	*Type of weld*	*Polar moment of inertia (J)*	*Section modulus (Z)*
6.	 $x = \dfrac{l^2}{2\,(l+b)},\ y = \dfrac{b^2}{2\,(l+b)}$	$t\left[\dfrac{(b+l)^4 - 6b^2 l^2}{12\,(l+b)}\right]$	$t\left(\dfrac{4l.b + b^2}{6}\right)$ (Top) $t\left[\dfrac{b^2\,(4lb+b)}{6\,(2l+b)}\right]$ (Bottom)
7.	 $x = \dfrac{l^2}{2l+b}$	$t\left[\dfrac{(b+2l)^3}{12} - \dfrac{l^2\,(b+l)^2}{b+2l}\right]$	$t\left(l.b + \dfrac{b^2}{6}\right)$
8.		$\dfrac{\pi t d^3}{4}$	$\dfrac{\pi t d^2}{4}$

Table 11.1. Design dimensions of screw threads, bolts and nuts according to IS : 4218 (Part III) 1976 (Reaffirmed 1996) (Refer Fig. 11.1)

Designation	*Pitch mm*	*Major or nominal diameter Nut and Bolt (d = D) mm*	*Effective or pitch diameter Nut and Bolt (d_p) mm*	*Minor or core diameter (d_c) mm*		*Depth of thread (bolt) mm*	*Stress area mm^2*
				Bolt	*Nut*		
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)
Coarse series							
M 0.4	0.1	0.400	0.335	0.277	0.292	0.061	0.074
M 0.6	0.15	0.600	0.503	0.416	0.438	0.092	0.166
M 0.8	0.2	0.800	0.670	0.555	0.584	0.123	0.295
M 1	0.25	1.000	0.838	0.693	0.729	0.153	0.460
M 1.2	0.25	1.200	1.038	0.893	0.929	0.158	0.732
M 1.4	0.3	1.400	1.205	1.032	1.075	0.184	0.983
M 1.6	0.35	1.600	1.373	1.171	1.221	0.215	1.27
M 1.8	0.35	1.800	1.573	1.371	1.421	0.215	1.70
M 2	0.4	2.000	1.740	1.509	1.567	0.245	2.07
M 2.2	0.45	2.200	1.908	1.648	1.713	0.276	2.48
M 2.5	0.45	2.500	2.208	1.948	2.013	0.276	3.39
M 3	0.5	3.000	2.675	2.387	2.459	0.307	5.03
M 3.5	0.6	3.500	3.110	2.764	2.850	0.368	6.78
M 4	0.7	4.000	3.545	3.141	3.242	0.429	8.78
M 4.5	0.75	4.500	4.013	3.580	3.688	0.460	11.3
M 5	0.8	5.000	4.480	4.019	4.134	0.491	14.2
M 6	1	6.000	5.350	4.773	4.918	0.613	20.1
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)
M 7	1	7.000	6.350	5.773	5.918	0.613	28.9
M 8	1.25	8.000	7.188	6.466	6.647	0.767	36.6
M 10	1.5	10.000	9.026	8.160	8.876	0.920	58.3
M 12	1.75	12.000	10.863	9.858	10.106	1.074	84.0
M 14	2	14.000	12.701	11.546	11.835	1.227	115
M 16	2	16.000	14.701	13.546	13.835	1.227	157
M 18	2.5	18.000	16.376	14.933	15.294	1.534	192
M 20	2.5	20.000	18.376	16.933	17.294	1.534	245
M 22	2.5	22.000	20.376	18.933	19.294	1.534	303
M 24	3	24.000	22.051	20.320	20.752	1.840	353
M 27	3	27.000	25.051	23.320	23.752	1.840	459

M 30	3.5	30.000	27.727	25.706	26.211	2.147	561
M 33	3.5	33.000	30.727	28.706	29.211	2.147	694
M 36	4	36.000	33.402	31.093	31.670	2.454	817
M 39	4	39.000	36.402	34.093	34.670	2.454	976
M 42	4.5	42.000	39.077	36.416	37.129	2.760	1104
M 45	4.5	45.000	42.077	39.416	40.129	2.760	1300
M 48	5	48.000	44.752	41.795	42.587	3.067	1465
M 52	5	52.000	48.752	45.795	46.587	3.067	1755
M 56	5.5	56.000	52.428	49.177	50.046	3.067	2022
M 60	5.5	60.000	56.428	53.177	54.046	3.374	2360
Fine series							
M 8 × 1	1	8.000	7.350	6.773	6.918	0.613	39.2
M 10 × 1.25	1.25	10.000	9.188	8.466	8.647	0.767	61.6
M 12 × 1.25	1.25	12.000	11.184	10.466	10.647	0.767	92.1
M 14 × 1.5	1.5	14.000	13.026	12.160	12.376	0.920	125
M 16 × 1.5	1.5	16.000	15.026	14.160	14.376	0.920	167
M 18 × 1.5	1.5	18.000	17.026	16.160	16.376	0.920	216
M 20 × 1.5	1.5	20.000	19.026	18.160	18.376	0.920	272
M 22 × 1.5	1.5	22.000	21.026	20.160	20.376	0.920	333
M 24 × 2	2	24.000	22.701	21.546	21.835	1.227	384
M 27 × 2	2	27.000	25.701	24.546	24.835	1.227	496
M 30 × 2	2	30.000	28.701	27.546	27.835	1.227	621
M 33 × 2	2	33.000	31.701	30.546	30.835	1.227	761
M 36 × 3	3	36.000	34.051	32.319	32.752	1.840	865
M 39 × 3	3	39.000	37.051	35.319	35.752	1.840	1028

Note : In case the table is not available, then the core diameter (d_c) may be taken as 0.84 d, where d is the major diameter.

11.12 Stresses due to External Forces

The following stresses are induced in a bolt when it is subjected to an external load.

1. *Tensile stress.* The bolts, studs and screws usually carry a load in the direction of the bolt axis which induces a tensile stress in the bolt.

Let d_c = Root or core diameter of the thread, and

σ_t = Permissible tensile stress for the bolt material.

We know that external load applied,

$$P = \frac{\pi}{4}(d_c)^2\ \sigma_t \quad \text{or} \quad d_c = \sqrt{\frac{4\,P}{\pi\,\sigma_t}}$$

Now from Table 11.1, the value of the nominal diameter of bolt corresponding to the value of d_c may be obtained or stress area $\left[\frac{\pi}{4}(d_c)^2\right]$ may be fixed.

Notes: (*a*) If the external load is taken up by a number of bolts, then

$$P = \frac{\pi}{4}(d_c)^2\ \sigma_t \times n$$

(*b*) In case the standard table is not available, then for coarse threads, $d_c = 0.84\ d$, where d is the nominal diameter of bolt.

2. *Shear stress.* Sometimes, the bolts are used to prevent the relative movement of two or more parts, as in case of flange coupling, then the shear stress is induced in the bolts. The shear stresses should be avoided as far as possible. It should be noted that when the bolts are subjected to direct shearing loads, they should be located in such a way that the shearing load comes upon the body (*i.e.* shank) of the bolt and not upon the threaded portion. In some cases, the bolts may be relieved of shear load by using shear pins. When a number of bolts are used to share the shearing load, the finished bolts should be fitted to the reamed holes.

Let d = Major diameter of the bolt, and

n = Number of bolts.

$\therefore$ Shearing load carried by the bolts,

$$P_s = \frac{\pi}{4} \times d^2 \times \tau \times n \quad \text{or} \quad d = \sqrt{\frac{4\,P_s}{\pi\,\tau\,n}}$$

3. *Combined tension and shear stress.* When the bolt is subjected to both tension and shear loads, as in case of coupling bolts or bearing, then the diameter of the shank of the bolt is obtained from the shear load and that of threaded part from the tensile load. A diameter slightly larger than that required for either shear or tension may be assumed and stresses due to combined load should be checked for the following principal stresses.

Maximum principal shear stress,

$$\tau_{max} = \frac{1}{2}\sqrt{(\sigma_t)^2 + 4\tau^2}$$

and maximum principal tensile stress,

$$\sigma_{t(max)} = \frac{\sigma_t}{2} + \frac{1}{2}\sqrt{(\sigma_t)^2 + 4\tau^2}$$

These stresses should not exceed the safe permissible values of stresses.

These stresses should not exceed the safe permissible values of stresses.

Example 11.3. *An eye bolt is to be used for lifting a load of 60 kN. Find the nominal diameter of the bolt, if the tensile stress is not to exceed 100 MPa. Assume coarse threads.*

Solution. Given : $P = 60$ kN $= 60 \times 10^3$ N ; $\sigma_t = 100$ MPa $= 100$ N/mm^2

An eye bolt for lifting a load is shown in Fig. 11.22.

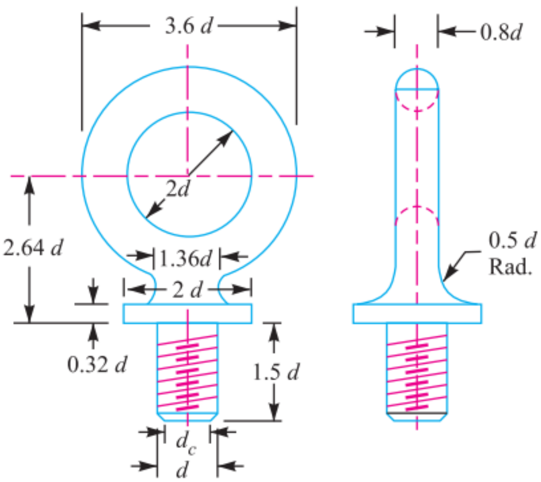


Fig. 11.22

Let d = Nominal diameter of the bolt, and

d_c = Core diameter of the bolt.

We know that load on the bolt (P),

$$60 \times 10^3 = \frac{\pi}{4}(d_c)^2 \sigma_t = \frac{\pi}{4}(d_c)^2\ 100 = 78.55\,(d_c)^2$$

$\therefore \quad (d_c)^2 = 600 \times 10^3 / 78.55 = 764 \quad$ or $\quad d_c = 27.6$ mm

From Table 11.1 (coarse series), we find that the standard core diameter (d_c) is 28.706 mm and the corresponding nominal diameter (d) is 33 mm. **Ans.**

Example 11.4. *Two shafts are connected by means of a flange coupling to transmit torque of 25 N-m. The flanges of the coupling are fastened by four bolts of the same material at a radius of 30 mm. Find the size of the bolts if the allowable shear stress for the bolt material is 30 MPa.*

Solution. Given : $T = 25$ N-m $= 25 \times 10^3$ N-mm ; $n = 4$; $R_p = 30$ mm ; $\tau = 30$ MPa $= 30$ N/mm^2

We know that the shearing load carried by flange coupling,

$$P_s = \frac{T}{R_p} = \frac{25 \times 10^3}{30} = 833.3 \text{ N} \qquad \text{...}(i)$$

Let $\quad d_c$ = Core diameter of the bolt.

$\therefore$ Resisting load on the bolts

$$= \frac{\pi}{4}(d_c)^2 \tau \times n = \frac{\pi}{4}(d_c)^2\ 30 \times 4 = 94{\cdot}26\,(d_c)^2 \qquad \text{...}(ii)$$

From equations **(*i*)** and **(*ii*)**, we get

$$(d_c)^2 = 833.3 / 94.26 = 8.84 \qquad \text{or} \qquad d_c = 2.97 \text{ mm}$$

From Table 11.1 (coarse series), we find that the standard core diameter of the bolt is 3.141 mm and the corresponding size of the bolt is M 4. **Ans.**