Screw Joint

Designation of Screw Threads

According to Indian standards, IS : 4218 (Part IV) 1976 (Reaffirmed 1996), the complete designation of the screw thread shall include

1. ***Size designation*****.** The size of the screw thread is designated by the letter `*M*' followed by the diameter and pitch, the two being separated by the sign ×. When there is no indication of the pitch, it shall mean that a coarse pitch is implied.

2. ***Tolerance designation*****.** This shall include

(*a*) A figure designating tolerance grade as indicated below:

'7' for fine grade, '8' for normal (medium) grade, and '9' for coarse grade.

(*b*) A letter designating the tolerance position as indicated below :

'*H*' for unit thread, '*d*' for bolt thread with allowance, and '*h*' for bolt thread without allowance. For example, A bolt thread of 6 mm size of coarse pitch and with allowance on the threads and normal (medium) tolerance grade is designated as *M*6-8*d*.

Standard Dimensions of Screw Threads

The design dimensions of I.S.O. screw threads for screws, bolts and nuts of coarse and fine series are shown in Table 1.

Table 1. Design dimensions of screw threads, bolts and nuts according to IS : 4218 (Part III) 1976 (Reaffirmed 1996)

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 mm²*
				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

Stresses in Screwed Fastening due to Static Loading

The following stresses in screwed fastening due to static loading are important from the subject point of view :

1. Internal stresses due to screwing up forces,

2. Stresses due to external forces, and

3. Stress due to combination of stresses at (1) and (2).

We shall now discuss these stresses, in detail, in the following articles

Initial Stresses due to Screwing up Forces

The following stresses are induced in a bolt, screw or stud when it is screwed up tightly.

1. ***Tensile stress due to stretching of bolt.***

The initial tension in a bolt, based on experiments, may be found by the relation

$$P_i = 2840\, d \text{ N}$$

Where P_i = Initial tension in a bolt, and

d = Nominal diameter of bolt, in mm.

The maximum safe axial load which may be applied to it, is given by

P = Permissible stress × Cross-sectional area at bottom of the thread(*i.e.* stress area)

The stress area may be obtained from Table 1 or it may be found by using the relation

$$\text{Stress area} = \frac{\pi}{4}\left(\frac{d_p + d_c}{2}\right)^2$$

d_p = Pitch diameter, and

d_c = Core or minor diameter.

2. ***Torsional shear stress caused by the frictional resistance of the threads during its tightening*.**

$$\frac{T}{J} = \frac{\tau}{r}$$

$$\tau = \frac{T}{J} \times r = \frac{T}{\frac{\pi}{32}(d_c)^4} \times \frac{d_c}{2} = \frac{16\,T}{\pi\,(d_c)^3}$$

τ = Torsional shear stress,

T = Torque applied, and

d_c = Minor or core diameter of the thread.

3. ***Shear stress across the threads*.** The average thread shearing stress for the screw (τ_s) is obtained by using the relation :

$$\tau_s = \frac{P}{\pi d_c \times b \times n}$$

b = Width of the thread section at the root.

The average thread shearing stress for the nut is

$$\tau_n = \frac{P}{\pi d \times b \times n}$$

d = Major diameter.

4. ***Compression or crushing stress on threads*.** The compression or crushing stress between the threads (σ_c) may be obtained by using the relation:

$$\sigma_c = \frac{P}{\pi\,[d^2 - (d_c)^2]\,n}$$

d = Major diameter,

d_c = Minor diameter, and

n = Number of threads in engagement

5. ***Bending stress if the surfaces under the head or nut are not perfectly parallel to the bolt axis.***

$$\sigma_b = \frac{x \, . \, E}{2l}$$

x = Difference in height between the extreme corners of the nut or head,

l = Length of the shank of the bolt, and

E = Young's modulus for the material of the bolt.

Problem 1

Determine the safe tensile load for a bolt of M 30, assuming a safe tensile stress of 42 MPa.

Solution

d = 30 mm ; σ_t = 42 MPa = 42 N/mm^2

From Table 1 (coarse series), we find that the stress area *i.e*. cross-sectional area at the bottom of the thread corresponding to M 30 is 561 mm2.

∴ Safe tensile load = Stress area × σ_t = 561 × 42 = 23 562 N = 23.562 kN

Problem 2

Two machine parts are fastened together tightly by means of a 24 mm tap bolt. If the load tending to separate these parts is neglected, find the stress that is set up in the bolt by the initial tightening.

Solution

d = 24 mm

From Table 1 (coarse series), we find that the core diameter of the thread corresponding to

M 24 is d_c = 20.32 mm. Let σ_t = Stress set up in the bolt.

We know that initial tension in the bolt,

P = 2840 d = 2840 × 24 = 68 160 N

We also know that initial tension in the bolt (P),

$$68\ 160 = \frac{\pi}{4}(d_c)^2\,\sigma_t = \frac{\pi}{4}(20.30)^2\ \sigma_t = 324\ \sigma_t$$

$$\sigma_t = 68\ 160 / 324 = 210\ \text{N/mm}^2 = 210\ \text{MPa}$$

Stresses due to External Forces

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,

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

Now from Table 1, the value of the nominal diameter of bolt corresponding to the value of *dc* may be obtained

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*.

Shearing load carried by the bolts,

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

3. *Combined tension and shear stress*

Maximum principal shear stress,

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

and maximum principal tensile stress,

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

Problem 3

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

T = 25 N-m = 25×10^3 N-mm ; n = 4; R_p = 30 mm ; τ = 30 MPa = 30 N/mm^2

We know that the shearing load carried by flange coupling,

$$P_s = \frac{T}{R_p} = \frac{25 \times 10^3}{30} = 833.3 \text{ N}$$

∴Resisting load on the bolts

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

$(d_c)^2$ = 833.3 / 94.26 = 8.84 or d_c = 2.97 mm

From Table 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