TABLE OF CONTENTS

FASTENERS

NAILS, SPIKES AND WOOD SCREWS

RIVETS AND RIVETED JOINTS

TORQUE AND TENSION IN FASTENERS

BOLTS, SCREWS, NUTS, AND WASHERS

METRIC FASTENERS

TABLE OF CONTENTS

TABLE OF CONTENTS

CAP AND SET SCREWS

SELF-THREADING SCREWS

T-SLOTS, BOLTS, AND NUTS

PINS AND STUDS

RETAINING RINGS

WING NUTS, WING SCREWS AND THUMB SCREWS

NAILS, SPIKES AND WOOD SCREWS

Standard Wire Nails and Spikes

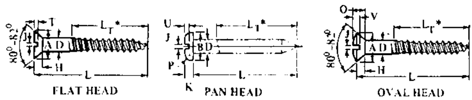
(Size, Length and Approximate Number to Pound)

Size of Nail	Length, Inches	Common Wire Nails and Brads Gage	Common Wire Nails and Brads Num/lb	Flooring Brads Gage	Flooring Brads Num/lb	Fence Nails Gage	Fence Nails Num/lb	Casing, Smooth and Barbed Box Gage	Casing, Smooth and Barbed Box Num/lb	Finishing Nails Gage	Finishing Nails Num/lb
2 *d*	1	15	876	…	…	…	…	15 ½	1010	16 ½	1351
3 *d*	1 ¼	14	568	…	…	…	…	14 ½	635	15 ½	807
4 *d*	1 ½	12 ½	316	…	…	…	…	14	473	15	584
5 *d*	1 ¾	12 ½	271	…	…	10	142	14	406	15	500
6 *d*	2	11 ½	181	11	157	10	124	12 ½	236	13 ½	309
7 *d*	2 ¼	11 ½	161	11	139	9	92	12 ½	210	13	238
8 *d*	2 ½	10 ¼	106	10	99	9	82	11 ½	145	12 ½	189
9 *d*	2 ¾	10 ¼	96	10	90	8	62	11 ½	132	12 ½	172
10 *d*	3	9	69	9	69	7	50	10 ½	94	11 ½	121
12 *d*	3 ¼	9	64	8	54	6	40	10 ½	87	11 ½	113
16 *d*	3 ½	8	49	7	43	5	30	10	71	11	90
20 *d*	4	6	31	6	31	4	23	9	52	10	62
30 *d*	4 ½	5	24	…	…	…	…	9	46	…	…
40 *d*	5	4	18	…	…	…	…	8	35	…	…
50 *d*	5 ½	3	16	…	…	…	…	…	…	…	…
60 *d*	6	2	11	…	…	…	…	…	…	…	…

Size and Length		Hinge Nails, Heavy Gage	Hinge Nails, Heavy Num/lb	Hinge Nails, Light Gage	Hinge Nails, Light Num/lb	Clinch Nails Gage	Clinch Nails Num/lb	Barbed Car Nails, Heavy Gage	Barbed Car Nails, Heavy Num/lb	Barbed Car Nails, Light Gage	Barbed Car Nails, Light Num/lb
2 *d*	1	…	…	…	…	14	710	…	…	…	…
3 *d*	1 ¼	…	…	…	…	13	429	…	…	…	…
4 *d*	1 ½	3	50	6	82	12	274	10	165	12	274
5 *d*	1 ¾	3	38	6	62	12	235	9	118	10	142
6 *d*	2	3	30	6	50	11	157	9	103	10	124
7 *d*	2 ¼	00	12	3	25	11	139	8	76	9	92
8 *d*	2 ½	00	11	3	23	10	99	8	69	9	82
9 *d*	2 ¾	00	10	3	22	10	90	7	54	8	62
10 *d*	3	00	9	3	19	9	69	7	50	8	57
12 *d*	3 ¼	…	…	…	…	9	62	6	42	7	50
16 *d*	3 ½	…	…	…	…	8	49	6	35	7	43
20 *d*	4	…	…	…	…	7	37	5	26	6	31
30 *d*	4 ½	…	…	…	…	…	…	5	24	6	28
40 *d*	5	…	…	…	…	…	…	4	18	5	21
50 *d*	5 ½	…	…	…	…	…	…	3	15	4	17
60 *d*	6	…	…	…	…	…	…	3	13	4	15

Size and Length		Boat Nails, Heavy Gage	Boat Nails, Heavy Num/lb	Boat Nails, Light Gage	Boat Nails, Light Num/lb	Slating Nails Gage	Slating Nails Num/lb	Spikes Size and Length		Spikes Gage	Spikes No. to Lb
2 *d*	1	…	…	…	…	12	411				
3 *d*	1 ¼	…	…	…	…	10 ½	225				
4 *d*	1 ½	¼	44	3/16	82	10 ½	187				
5 *d*	1 ¾	…	…	…	…	10	142	10 *d*	3	6	41
6*d*	2	¼	32	3/16	62	9	103	12 *d*	3 ¼	6	38
7 *d*	2 ¼	…	…	…	…	…	…	16 *d*	3 ½	5	30
8 *d*	2 ½	¼	26	3/16	50	…	…	20 *d*	4	4	23
9 *d*	2 ¾	…	…	…	…	…	…	30 *d*	4 ½	3	17
10 *d*	3	⅜	14	¼	22	…	…	40 *d*	5	2	13
12 *d*	3 ¼	⅜	13	¼	20	…	…	50 *d*	5 ½	1	10
16 *d*	3 ½	⅜	12	¼	18	…	…	60 *d*	6	1	8
20 *d*	4	⅜	10	¼	16	…	…	…	7	0	7
30 *d*	4 ½	…	…	…	…	…	…	…	8	00	6
40 *d*	5	…	…	…	…	…	…	…	9	00	5
50 *d*	5 ½	…	…	…	…	…	…	…	10	⅜	4
60 *d*	6	…	…	…	…	…	…	…	12	⅜	3

ANSI Flat, Pan, and Oval Head Wood Screws *ANSI B18.6.1-1981 (R1997)*



Nominal Size	Threads per inch	D^a Basic Diam. of Screw	*J* Width of Slot Max.	*J* Width of Slot Min.	*A* Head Diameter Max., Sharp Edge	*A* Head Diameter Min., Edge Rounded or Flat	*B* Head Diameter Max.	*B* Head Diameter Min.	*P* Head Radius Max.	*H* Height of Head Ref.
0	32	.060	.023	.016	.119	.099	.116	.104	.020	.035
1	28	.073	.026	.019	.146	.123	.142	.130	.025	.043
2	26	.086	.031	.023	.172	.147	.167	.155	.035	.051
3	24	.099	.035	.027	.199	.171	.193	.180	.037	.059
4	22	.112	.039	.031	.225	.195	.219	.205	.042	.067
5	20	.125	.043	.035	.252	.220	.245	.231	.044	.075
6	18	.138	.048	.039	.279	.244	.270	.256	.046	.083
7	16	.151	.048	.039	.305	.268	.296	.281	.049	.091
8	15	.164	.054	.045	.332	.292	.322	.306	.052	.100
9	14	.177	.054	.045	.358	.316	.348	.331	.056	.108
10	13	.190	.060	.050	.385	.340	.373	.357	.061	.116
12	11	.216	.067	.056	.438	.389	.425	.407	.078	.132
14	10	.242	.075	.064	.507	.452	.492	.473	.087	.153
16	9	.268	.075	.064	.544	.485	.528	.508	.094	.164
18	8	.294	.084	.072	.635	.568	.615	.594	.099	.191
20	8	.320	.084	.072	.650	.582	.631	.608	.121	.196
24	7	.372	.094	.081	.762	.685	.740	.716	.143	.230

[a] *Diameter Tolerance:* Equals + 0.004 in. and − 0.007 in. for cut threads. For rolled thread body diameter tolerances, see ANSI 18.6.1-1981 (R1991).

Nominal Size	Threads per Inch	*O* Tot. Hgt. of Head Max.	*O* Tot. Hgt. of Head Min.	*K* Height of Head Max.	*K* Height of Head Min.	*T* Depth of Slot Max.	*T* Depth of Slot Min.	*U* Depth of Slot Max.	*U* Depth of Slot Min.	*V* Depth of Slot Max.	*V* Depth of Slot Min.
0	32	.056	.041	.039	.031	.015	.010	.022	.014	.030	.025
1	28	.068	.052	.046	.038	.019	.012	.027	.018	.038	.031
2	26	.080	.063	.053	.045	.023	.015	.031	.022	.045	.037
3	24	.092	.073	.060	.051	.027	.017	.035	.027	.052	.043
4	22	.104	.084	.068	.058	.030	.020	.040	.030	.059	.049
5	20	.116	.095	.075	.065	.034	.022	.045	.034	.067	.055
6	18	.128	.105	.082	.072	.038	.024	.050	.037	.074	.060
7	16	.140	.116	.089	.079	.041	.027	.054	.041	.081	.066
8	15	.152	.126	.096	.085	.045	.029	.058	.045	.088	.072
9	14	.164	.137	.103	.092	.049	.032	.063	.049	.095	.078
10	13	.176	.148	.110	.099	.053	.034	.068	.053	.103	.084
12	11	.200	.169	.125	.112	.060	.039	.077	.061	.117	.096
14	10	.232	.197	.144	.130	.070	.046	.087	.070	.136	.112
16	9	.248	.212	.153	.139	.075	.049	.093	.074	.146	.120
18	8	.290	.249	.178	.162	.083	.054	.106	.085	.171	.141
20	8	.296	.254	.182	.166	.090	.059	.108	.087	.175	.144
24	7	.347	.300	.212	.195	.106	.070	.124	.100	.204	.168

All dimensions in inches. The edge of flat and oval head screws may be flat or rounded. Wood screws are also available with Types I, IA, and II recessed heads. Consult the standard for recessed head dimensions. *The length of the thread, L_T, on wood screws having cut threads shall be equivalent to approximately two-thirds of the nominal length of the screw. For rolled threads, L_T shall be at least four times the basic screw diameter or two-thirds of the nominal screw length, whichever is greater. Screws of nominal lengths that are too short to accommodate the minimum thread length shall have threads extending as close to the underside of the head as practicable.

Pilot Hole Drill Sizes for Wood Screws

Work Material	Wood Screw Size 2	4	6	8	10	12	14
Hardwood	3⁄64	1⁄16	5⁄64	3⁄32	7⁄64	1⁄8	9⁄64
Softwood	1⁄32	3⁄64	1⁄16	5⁄64	3⁄32	7⁄64	1⁄8

RIVETS AND RIVETED JOINTS

Classes and Types of Riveted Joints.—Riveted joints may be classified by application as: 1) pressure vessel; 2) structural; and 3) machine member.

For information and data concerning joints for pressure vessels such as boilers, reference should be made to standard sources such as the ASME Boiler Code. The following sections will cover only structural and machine-member riveted joints.

Basically there are two kinds of riveted joints, the *lap-joint* and the *butt-joint*. In the ordinary *lap-joint*, the plates overlap each other and are held together by one or more rows of rivets. In the *butt-joint*, the plates being joined are in the same plane and are joined by means of a cover plate or butt strap, which is riveted to both plates by one or more rows of rivets. The term *single riveting* means one row of rivets in a lap-joint or one row on each side of a butt-joint; *double riveting* means two rows of rivets in a lap-joint or two rows on each side of the joint in butt riveting. Joints are also triple and quadruple riveted. Lap-joints may also be made with inside or outside cover plates. Types of lap and butt joints are illustrated in the table on starting on page 1461.

General Considerations of a Riveted Joint.—Factors to be considered in the design or specification of a riveted joint are: type of joint; spacing of rivets; type and size of rivet; type and size of hole; and rivet material.

Spacing of Rivets: The spacing between rivet centers is called *pitch* and between row center lines, *back pitch* or *transverse pitch.* The distance between centers of rivets nearest each other in adjacent rows is called *diagonal pitch.* The distance from the edge of the plate to the center line of the nearest row of rivets is called *margin.*

Examination of a riveted joint made up of several rows of rivets will reveal that after progressing along the joint a given distance, the rivet pattern or arrangement is repeated. (For a butt joint, the length of a *repeating section* is usually equal to the *long pitch* or pitch of the rivets in the outer row, that is the row farthest from the edge of the joint.) For structural and machine-member joints, the proper pitch may be determined by making the tensile strength of the plate over the length of the repeating section, that is the distance between rivets in the outer row, equal to the total shear strength of the rivets in the repeating section. Minimum pitch and diagonal pitch are also governed by the clearance required for the hold-on (Dolly bar) and rivet set. Dimensions for different sizes of hold-ons and rivet sets are given in the table on page 1467.

When fastening thin plate, it is particularly important to maintain accurate spacing to avoid buckling.

Size and Type of Rivets: The rivet diameter d commonly falls between $d = 1.2\sqrt{t}$ and $d = 1.4\sqrt{t}$, where t is the thickness of the plate. Dimensions for various types of American Standard large (½-inch diameter and up) rivets and small solid rivets are shown in tables that follow. It may be noted that countersunk heads are not as strong as other types.

Size and Type of Hole: Rivet holes may be punched, punched and reamed, or drilled. Rivet holes are usually made $\frac{1}{16}$ inch larger in diameter than the nominal diameter of the rivet although in some classes of work in which the rivet is driven cold, as in automatic machine riveting, the holes are reamed to provide minimum clearance so that the rivet fills the hole completely.

When holes are punched in heavy steel plate, there may be considerable loss of strength unless the holes are reamed to remove the inferior metal immediately surrounding them. This results in the diameter of the punched hole being increased by from $\frac{1}{16}$ to $\frac{1}{8}$ inch. Annealing after punching tends to restore the strength of the plate in the vicinity of the holes.

Rivet Material: Rivets for structural and machine-member purposes are usually made of wrought iron or soft steel, but for aircraft and other applications where light weight or resistance to corrosion is important, copper, aluminum alloy, Monel, Inconel, etc., may be used as rivet material.

Failure of Riveted Joints.—Rivets may fail by:

1) Shearing through one cross-section (single shear)

2) Shearing through two cross-sections (double shear)

3) Crushing

Plates may fail by:

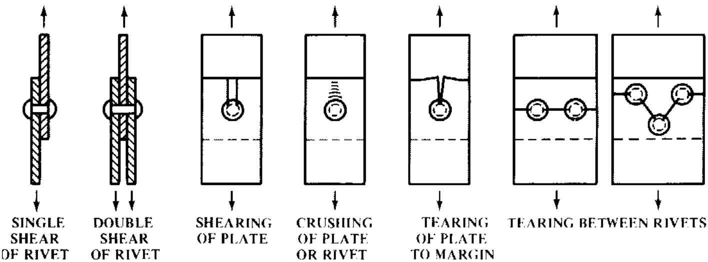
4) Shearing along two parallel lines extending from opposite sides of the rivet hole to the edge of the plate

5) Tearing along a single line from middle of rivet hole to edge of plate

6) Crushing

7) Tearing between adjacent rivets (tensile failure) in the same row or in adjacent rows

Types 4 and 5 failures are caused by rivets being placed too close to the edge of the plate. These types of failure are avoided by placing the center of the rivet at a minimum of one and one-half times the rivet diameter away from the edge.



Types of Rivet and Plate Failure

Failure due to tearing on a diagonal between rivets in adjacent rows when the pitch is four times the rivet diameter or less is avoided by making the transverse pitch one and three-quarters times the rivet diameter.

Theoretical versus Actual Riveted Joint Failure.—If it is assumed that the rivets are placed the suggested distance from the edge of the plate and each row the suggested distance from another row, then the failure of a joint is most likely to occur as a result of shear failure of the rivets, bearing failure (crushing) of the plate or rivets, or tensile failure of the plate, alone or in combination depending on the makeup of the joints.

Joint failure in actuality is more complex than this. Rivets do not undergo pure shear especially in lap-joints where rivets are subjected to single shear. The rivet, in this instance, would be subject to a combination of tensile and shearing stresses and it would fail because of combined stresses, not a single stress. Furthermore, the shearing stress is usually considered to be distributed evenly over the cross-section, which is also not the case.

Rivets that are usually driven hot contract on cooling. This contraction in the length of the rivet draws the plates together and sets up a stress in the rivet estimated to be equal in magnitude to the yield point of the rivet steel. The contraction in the diameter of the rivet results in a little clearance between the rivetand the hole in the plate. The tightness in the plates caused by the contraction in length of the rivet gives rise to a condition in which quite a sizeable frictional force would have to be overcome before the plates would slip over one another and subject the rivets to a shearing force. It is European practice to design joints for resistance to this slipping. It has been found, however, that the strength-basis designs obtained in American and English practice are not very different from European designs.

Design of Riveted Joints.—In the design of riveted joints, a simplified treatment is frequently used in which the following assumptions are made:

1) The load is carried equally by the rivets.

2) No combined stresses act on a rivet to cause failure.

3) The shearing stress in a rivet is uniform across the cross-section under question.

4) The load that would cause failure in single shear would have to be doubled to cause failure in double shear.

5) The bearing stress of rivet and plate is distributed equally over the projected area of the rivet.

6) The tensile stress is uniform in the section of metal between the rivets.

Allowable Stresses.—The design stresses for riveted joints are usually set by codes, practices, or specifications. The American Institute of Steel Construction issues specifications for the design, fabrication, and erection of structural steel for buildings in which the allowable stress permitted in tension for structural steel and rivets is specified at 20,000 pounds per square inch, the allowable bearing stress for rivets is 40,000 psi in double shear and 32,000 psi in single shear, and the allowable shearing stress for rivets is 15,000 psi. The American Society of Mechanical Engineers in its Boiler Code lists the following ultimate stresses: tensile, 55,000 psi; shearing, 44,000 psi; compressive or bearing, 95,000 psi. The design stresses usually are one-fifth of these, that is tensile, 11,000 psi; shearing, 8800 psi; compressive or bearing, 19,000 psi. In machine design work, values close to these or somewhat lower are commonly used.

Analysis of Joint Strength.—The following examples and strength analyses of riveted joints are based on the six assumptions previously outlined for a simplified treatment.

Example 1: Consider a 12-inch section of single-riveted lap-joint made up with plates of ¼-inch thickness and six rivets, ⅝ inch in diameter. Assume that rivet holes are 1⁄16 inch larger in diameter than the rivets. In this joint, the entire load is transmitted from one plate to the other by means of the rivets. Each plate and the six rivets carry the entire load. The safe tensile load L and the efficiency η may be determined in the following way: Design stresses of 8500 psi for shear, 20,000 psi for bearing, and 10,000 psi for tension are arbitrarily assigned and it is assumed that the rivets will not tear or shear through the plate to the edge of the joint.

A) The safe tensile load L based on single shear of the rivets is equal to the number of rivets n times the cross-sectional area of one rivet A_r times the allowable shearing stress S_s or

$$L = n \times A_r \times S_s$$

$$L = 6 \times \frac{\pi}{4}(0.625)^2 \times 8500$$

$$L = 15,647 \text{ pounds}$$

B) The safe tensile load L based on bearing stress is equal to the number of rivets n times the projected bearing area of the rivet A_b (diameter times thickness of plate) times the allowable bearing stress S_c or $L = n \times A_b \times S_c = 6 \times (0.625 \times 0.25) \times 20{,}000 = 18{,}750$ pounds.

C) The safe load L based on the tensile stress is equal to the net cross-sectional area of the plate between rivet holes A_p times the allowable tensile stress S_t or

$L = A_p \times S_t = 0.25[12 - 6(0.625 + 0.0625)] \times 10{,}000 = 19{,}688$ pounds.

The safe tensile load for the joint would be the least of the three loads just computed or 15,647 pounds and the efficiency η would be equal to this load divided by the tensile strength of the section of plate under consideration, if it were unperforated or

$$\eta = \frac{15,647}{12 \times 0.25 \times 10,000} \times 100 = 52.2 \text{ per cent}$$

Example 2: Under consideration is a 12-inch section of double-riveted butt-joint with main plates ½ inch thick and two cover plates each 5⁄16 inch thick. There are 3 rivets in the inner row and 2 on the outer and their diameters are 7⁄8 inch. Assume that the diameter of the rivet holes is 1⁄16 inch larger than that of the rivets. The rivets are so placed that the main plates will not tear diagonally from one rivet row to the others nor will they tear or fail in shear out to their edges. The safe tensile load L and the efficiency η may be determined in the following way: Design stresses for 8500 psi for shear, 20,000 psi for bearing, and 10,000 psi for tension are arbitrarily assigned.

A) The safe tensile load L based on double shearing of the rivets is equal to the number of rivets n times the number of shearing planes per rivet times the cross-sectional area of one rivet A_r times the allowable shearing stress S_s or $L = n \times 2$

$$\times A_r \times S_s = 5 \times 2 \times \frac{\pi}{4}(0.875)^2 \times 8500 = 51{,}112 \text{ pounds}$$

B) The safe tensile load L based on bearing stress is equal to the number of rivets n times the projected bearing area of the rivet A_b (diameter times thickness of plate) times the allowable bearing stress S_c or $L = n \times A_b \times S_c = 5 \times (0.875 \times 0.5) \times 20{,}000 = 43{,}750$ pounds.

(Cover plates are not considered since their combined thickness is ¼ inch greater than the main plate thickness.)

C) The safe tensile load L based on the tensile stress is equal to the net cross-sectional area of the plate between the two rivets in the outer row A_p times the allowable tensile stress S_t or $L = A_p \times S_t = 0.5[12 - 2(0.875 + 0.0625)] \times 10{,}000 = 50{,}625$ pounds.

In completing the analysis, the sum of the load that would cause tearing between rivets in the three-hole section plus the load carried by the two rivets in the two-hole section is also investigated. The sum is necessary because if the joint is to fail, it must fail at both sections simultaneously. The least safe load that can be carried by the two rivets of the two-hole section is based on the bearing stress (see the foregoing calculations).

1) The safe tensile load L based on the bearing strength of two rivets of the two-hole section is $L = n \times A_b \times S_c = 2 \times (0.875 \times 0.5) \times 20{,}000 = 17{,}500$ pounds.

2) The safe tensile load L based on the tensile strength of the main plate between holes in the three-hole section is $L \times A_p \times S_t = 0.5[12 - 3(0.875 + 0.0625)] \times 10{,}000 = 45{,}938$ pounds.

The total safe tensile load based on this combination is $17{,}500 + 45{,}938 = 63{,}438$ pounds, which is greater than any of the other results obtained.

The safe tensile load for the joint would be the least of the loads just computed or 43,750 pounds and the efficiency η would be equal to this load divided by the tensile strength of the section of plate under consideration, if it were unperforated or

$$\eta = \frac{43{,}750}{0.5 \times 12 \times 10{,}000} \times 100 = 72.9 \text{ per cent}$$

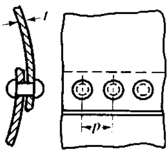
A riveted joint may fail by shearing through the rivets (single or double shear), crushing the rivets, tearing the plate between the rivets, crushing the plate or by a combination of two or more of the foregoing causes. Rivets placed too close to the edge of the plate may tear or shear the plate out to the edge but this type of failure is avoided by placing the center of the rivet 1.5 times the rivet diameter away from the edge.

The efficiency of a riveted joint is equal to the strength of the joint divided by the strength of the unriveted plate, expressed as a percentage.

In the following formulas, let

d = diameter of holes
t = thickness of plate
t_c= thickness of cove plates
p = Pitch of inner row of rivets

P = Pitch of outer row of ribets
S_s = Shear stress for rivets
S_t=Tensile stress for plates
S_c= Compressive or bearing stress for rivets or plates

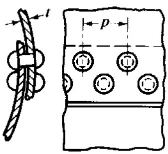


For Single-Riveted Lap-Joint

(1) Resistance to shearing one rivet = $\frac{\pi d^2}{4}S_s$

(2) Resistance to tearing plate between rivets = $(p - D)tS_t$

(3) Resistance to crushing rivet or plate = dtS_c

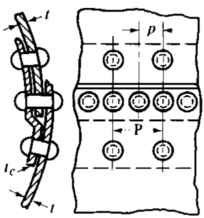


Double-Riveted Lap-Joint

(1) Resistance to shearing two rivets = $\frac{2\pi d^2}{4}S_s$

(2) Resistance to tearing between two rivets = $(p - D)tS_t$

(3) Resistance to crushing in front of two rivets = $2dtS_c$



Single-Riveted Lap-Joint with Inside Cover Plate

(1) Resistance to tearing between outer row of rivets
= $(P - D)tS_t$

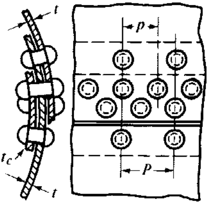
(2) Resistance to tearing between inner row of rivets, and

shearing outer row of rivets = $(P - 2D)tS_t + \frac{\pi d^2}{4}S_s$

(3) Resistance to shearing three rivets = $\frac{3\pi d^2}{4}S_s$

(4) Resistance to crushing in front of three rivets = $3tdS_c$

(5) Resistance to tearing at inner row of rivets, and crushing in front of one rivet in outer row
= $(P - 2D)tS_t + tdS_c$



Double-Riveted Lap-Joint with Inside Cover Plate

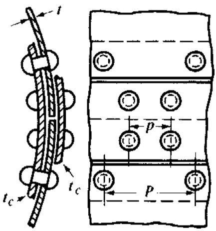
(1) Resistance to tearing at outer row of rivets = $(P - D)tS_t$

(2) Resistance to shearing four rivets = $\dfrac{4\pi d^2}{4}S_s$

(3) Resistance to tearing at inner row and shearing outer row of rivets = $(P - 1\tfrac{1}{2}D)tS_t + \dfrac{\pi d^2}{4}S_s$

(4) Resistance to crushing in front of four rivets = $4tdS_c$

(5) Resistance to tearing at inner row of rivets, and crushing in front of one rivet = $\left(P - 1\frac{1}{2}D\right)tS_t + tdS_c$



Double-Riveted Butt-Joint

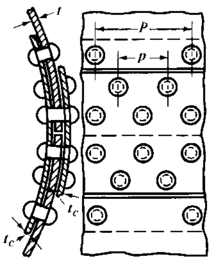
(1) Resistance to tearing at outer row of rivets = $(P - D)tS_t$

(2) Resistance to shearing two rivets in double shear and one in single shear = $\dfrac{5\pi d^2}{4}S_s$

(3) Resistance to tearing at inner row of rivets and shearing one rivet of the outer row = $(P - 2D)tS_t + \dfrac{\pi d^2}{4}S_s$

(4) Resistance to crushing in front of three rivets = $3tdS_c$

(5) Resistance to tearing at inner row of rivets, and crushing in front of one rivet in outer row = $(P - 2D)\ tS_t + tdS_c$



Triple-Riveted Butt-Joint

(1) Resistance to tearing at outer row of rivets = $(P - D)tS_t$

(2) Resistance to shearing four rivets in double shear and one in single shear = $\dfrac{9\pi d^2}{4}S_s$

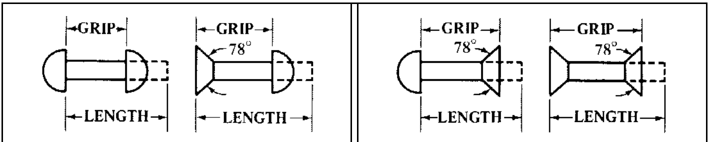
(3) Resistance to tearing at middle row of rivets and shearing one rivet = $(P - 2D)tS_t + \dfrac{\pi d^2}{4}S_s$

(4) Resistance to crushing in front of four rivets and shearing one rivet = $4dtS_c + \dfrac{\pi d^2}{4}S_s$

(5) Resistance to crushing in front of five rivets = $4dtS_c + dt_cS_c$

Dimensions are usually specified in inches and stresses in pounds per square inch. See page 1459 for a discussion of allowable stresses that may be used in calculating the strengths given by the formulas. The design stresses are usually set by codes, practices, or specifications.

Rivet Lengths for Forming Round and Countersunk Heads



Grip in Inches	To Form Round Head						
	Diameter of Rivet in Inches						
	½	⅝	¾	⅞	1	1⅛	1¼
	Length of Rivet in Inches						
½	1⅝	1⅞	1⅞	2	2⅛	…	…
⅝	1¾	2	2	2⅛	2¼	…	…
¾	1⅞	2⅛	2⅛	2¼	2⅜	…	…
⅞	2	2¼	2¼	2⅜	2½	…	…
1	2¼	2⅜	2⅜	2½	2⅝	2¾	2⅞
1⅛	2⅜	2½	2½	2⅝	2¾	2⅞	3
1¼	2½	2⅝	2⅝	2¾	2⅞	3	3⅛
1⅜	2⅝	2¾	2¾	2⅞	3	3⅛	3¼
1½	2⅞	3	3	3⅛	3¼	3⅜	3½
1⅝	3	3⅛	3⅛	3¼	3⅜	3½	3½
1¾	3⅛	3¼	3¼	3½	3⅝	3¾	3¾
1⅞	3¼	3⅜	3⅜	3⅝	3¾	3⅞	3⅞
2	3½	3½	3⅝	3¾	3⅞	4	4
2⅛	3⅝	3⅝	3¾	3⅞	4	4⅛	4⅛
2¼	3¾	3⅞	3⅞	4	4⅛	4¼	4¼
2⅜	4	4	4	4⅛	4¼	4⅜	4⅜
2½	4⅛	4⅛	4⅛	4¼	4⅜	4½	4½
2⅝	4¼	4¼	4¼	4⅜	4½	4⅝	4⅝
2¾	4⅜	4⅜	4⅜	4½	4⅝	4¾	4¾
2⅞	4⅝	4⅝	4⅝	4⅝	4¾	4⅞	5
3	…	4¾	4¾	4⅞	5	5	5⅛
3⅛	…	4⅞	4⅞	5	5⅛	5¼	5¼
3¼	…	5	5	5⅛	5¼	5⅜	5⅜
3⅜	…	5⅛	5⅛	5¼	5⅜	5½	5½
3½	…	5⅜	5⅜	5⅜	5½	5⅝	5⅝
3⅝	…	5½	5½	5½	5⅝	5¾	5¾
3¾	…	5⅝	5⅝	5⅝	5¾	5⅞	5⅞
3⅞	…	5¾	5¾	5¾	5⅞	6	6
4	…	…	5⅞	6	6	6⅛	6¼
4⅛	…	…	6	6⅛	6¼	6⅜	6⅜
4¼	…	…	6⅛	6¼	6½	6½	6½
4⅜	…	…	6⅜	6½	6½	6⅝	6⅝
4½	…	…	6½	6⅝	6⅝	6¾	6¾
4⅝	…	…	6⅝	6¾	6¾	6¾	6⅞
4¾	…	…	6¾	6⅞	6⅞	7	7
4⅞	…	…	6⅞	7	7	7⅛	7⅛
5	…	…	…	7⅛	7⅛	7¼	7¼
5⅛	…	…	…	7¼	7¼	7⅜	7⅜
5¼	…	…	…	7⅜	7⅜	7½	7½
5⅜	…	…	…	7⅝	7⅝	7¾	7⅝
5½	…	…	…	7¾	7¾	7⅞	7⅞
5⅝	…	…	…	7⅞	7⅞	8	8
5¾	…	…	…	8	8	8⅛	8⅛
5⅞	…	…	…	8⅛	8⅛	8¼	8¼

Grip in Inches	To Form Countersunk Head						
	Diameter of Rivet in Inches						
	½	⅝	¾	⅞	1	1⅛	1¼
	Length of Rivet in Inches						
½	1	1	1⅛	1¼	1¼	…	…
⅝	1⅛	1¼	1¼	1⅜	1⅜	…	…
¾	1⅜	1⅜	1⅜	1½	1½	…	…
⅞	1½	1½	1½	1⅝	1⅝	…	…
1	1⅝	1⅝	1⅝	1¾	1¾	1⅞	1⅞
1⅛	1¾	1¾	1⅞	1⅞	1⅞	2	2
1¼	2	2	2	2	2	2⅛	2⅛
1⅜	2⅛	2⅛	2⅛	2¼	2¼	2⅜	2⅜
1½	2¼	2¼	2¼	2⅜	2⅜	2½	2½
1⅝	2⅜	2⅜	2⅜	2½	2½	2⅝	2⅝
1¾	2⅝	2⅝	2⅝	2⅝	2⅝	2¾	2¾
1⅞	2¾	2¾	2¾	2¾	2¾	2⅞	2⅞
2	2⅞	2⅞	2⅞	2⅞	2⅞	3	3
2⅛	3⅛	3	3	3	3	3⅛	3⅛
2¼	3¼	3⅛	3⅛	3⅛	3¼	3¼	3¼
2⅜	3⅜	3⅜	3⅜	3⅜	3⅜	3⅜	3⅜
2½	3½	3½	3½	3½	3½	3⅝	3⅝
2⅝	3¾	3⅝	3⅝	3⅝	3⅝	3¾	3¾
2¾	3⅞	3¾	3¾	3¾	3¾	3⅞	3⅞
2⅞	4	3⅞	3⅞	3⅞	3⅞	4	4
3	…	4⅛	4⅛	4⅛	4⅛	4⅛	4⅛
3⅛	…	4¼	4¼	4¼	4¼	4¼	4¼
3¼	…	4⅜	4⅜	4⅜	4⅜	4⅜	4⅜
3⅜	…	4½	4½	4½	4½	4½	4½
3½	…	4⅝	4⅝	4⅝	4⅝	4⅝	4⅝
3⅝	…	4¾	4¾	4¾	4¾	4⅞	4⅞
3¾	…	5	5	5	5	5	5
3⅞	…	5⅛	5⅛	5⅛	5⅛	5⅛	5⅛
4	…	…	5¼	5¼	5¼	5¼	5¼
4⅛	…	…	5⅜	5⅜	5⅜	5⅜	5⅜
4¼	…	…	5½	5½	5½	5½	5½
4⅜	…	…	5⅝	5⅝	5⅝	5⅝	5⅝
4½	…	…	5¾	5¾	5¾	5¾	5¾
4⅝	…	…	6	6	6	6	6
4¾	…	…	6⅛	6⅛	6⅛	6⅛	6⅛
4⅞	…	…	6¼	6¼	6¼	6¼	6¼
5	…	…	…	6⅜	6⅜	6⅜	6⅜
5⅛	…	…	…	6½	6½	6½	6½
5¼	…	…	…	6⅝	6⅝	6⅝	6⅝
5⅜	…	…	…	6¾	6¾	6¾	6¾
5½	…	…	…	6⅞	6⅞	6⅞	6⅞
5⅝	…	…	…	7	7	7	7
5¾	…	…	…	7¼	7¼	7¼	7¼
5⅞	…	…	…	7⅜	7⅜	7⅜	7⅜

As given by the American Institute of Steel Construction. Values may vary from standard practice of individual fabricators and should be checked against the fabricator's standard.

Standard Rivets

Standards for rivets published by the American National Standards Institute and the British Standards Institution are as follows:

American National Standard Large Rivets.—The types of rivets covered by this standard (ANSI B18.1.2-1972 (R1995)) are shown on pages 1465 and 1466. It may be noted, however, that when specified, the swell neck included in this standard is applicable to all standard large rivets except the flat countersunk head and oval countersunk head types. Also shown are the hold-on (dolly bar) and rivet set impression dimensions (see page 1467). All standard large rivets have fillets under the head not exceeding an 0.062-inch radius. The length tolerances for these rivets are given as follows: through 6 inches in length, $\frac{1}{2}$- and $\frac{5}{8}$-inch diameters, ±0.03 inch; $\frac{3}{4}$- and $\frac{7}{8}$-inch diameters, ±0.06-inch; and 1- through $1\frac{3}{4}$-inch diameters, ±0.09 inch. For rivets over 6 inches in length, $\frac{1}{2}$- and $\frac{5}{8}$-inch diameters, ±0.06 inch; $\frac{3}{4}$- and $\frac{7}{8}$-inch diameters, ±0.12 inch; and 1- through $1\frac{3}{4}$-inch diameters, ±0.19 inch. Steel and wrought iron rivet materials appear in ASTM Specifications A31, A131, A152, and A502.

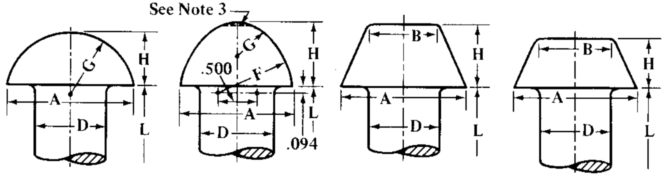
American National Standard Small Solid Rivets.—The types of rivets covered by this standard (ANSI/ASME B18.1.1-1972 (R1995)) are shown on pages 1468 through 1470. In addition, the standard gives the dimensions of 60-degree flat countersunk head rivets used to assemble ledger plates and guards for mower cutter bars, but these are not shown. As the heads of standard rivets are not machined or trimmed, the circumference may be somewhat irregular and edges may be rounded or flat. Rivets other than countersunk types are furnished with a definite fillet under the head, whose radius should not exceed 10 per cent of the maximum shank diameter or 0.03 inch, whichever is the smaller. With regard to head dimensions, tolerances shown in the dimensional tables are applicable to rivets produced by the normal cold heading process. Unless otherwise specified, rivets should have plain sheared ends that should be at right angles within 2 degrees to the axis of the rivet and be reasonably flat. When so specified by the user, rivets may have the standard header points shown on page 1468. Rivets may be made of ASTM Specification A31, Grade A steel; or may adhere to SAE Recommended Practice, Mechanical and Chemical Requirements for Nonthreaded Fasteners—SAE J430, Grade 0. When specified, rivets may be made of other materials.

ANSI/ASME B18.1.3M-1983 (R1995), Metric Small Solid Rivets, provides data for small, solid rivets with flat, round, and flat countersunk heads in metric dimensions. The main series of rivets has body diameters, in millimeters, of 1.6, 2, 2.5, 3, 4, 5, 6, 8, 10, and 12. A secondary series (nonpreferred) consists of sizes, 1, 1.2, 1.4, 3.5, 7, 9, and 11 millimeters.

British Standard Small Rivets for General Purposes.—Dimensions of small rivets for general purposes are given in British Standard 641:1951 and are shown in the table on page 1472. In addition, the standard lists the standard lengths of these rivets, gives the dimensions of washers to be used with countersunk head rivets (140°), indicates that the rivets may be made from mild steel, copper, brass, and a range of aluminum alloys and pure aluminum specified in B.S. 1473, and gives the dimensions of Coopers' flat head rivets $\frac{1}{2}$ inch in diameter and below, in an appendix. In all types of rivets, except those with countersunk heads, there is a small radius or chamfer at the junction of the head and the shank.

British Standard Dimensions of Rivets ($\frac{1}{2}$ to $1\frac{3}{4}$ inch diameter).—The dimensions of rivets covered in BS 275:1927 (obsolescent) are given on page 1471 and do not apply to boiler rivets. With regard to this standard the terms "nominal diameter" and "standard diameter" are synonymous. The term "tolerance" refers to the variation from the nominal diameter of the rivet and not to the difference between the diameter under the head and the diameter near the point.

American National Standard Large Rivets — 1 *ANSI B18.1.2-1972 (R1995)*



Button Head — High Button — Cone Head — Pan Head

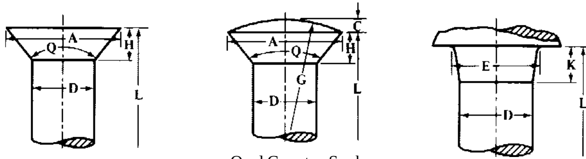
Nom. Body Dia. D[a]	Head Dia. *A*		Height *H*		Head Dia. *A*		Height *H*	
	Mfd. Note 1	Driven Note 2	Mfd. Note 1	Driven Note 2	Mfd. Note 1	Driven Note 2	Mfd. Note 1	Driven Note 2
	Button Head				High Button Head (Acorn)			
½	0.875	0.922	0.375	0.344	0.781	0.875	0.500	0.375
⅝	1.094	1.141	0.469	0.438	0.969	1.062	0.594	0.453
¾	1.312	1.375	0.562	0.516	1.156	1.250	0.688	0.531
⅞	1.531	1.594	0.656	0.609	1.344	1.438	0.781	0.609
1	1.750	1.828	0.750	0.688	1.531	1.625	0.875	0.688
1⅛	1.969	2.062	0.844	0.781	1.719	1.812	0.969	0.766
1¼	2.188	2.281	0.938	0.859	1.906	2.000	1.062	0.844
1⅜	2.406	2.516	1.031	0.953	2.094	2.188	1.156	0.938
1½	2.625	2.734	1.125	1.031	2.281	2.375	1.250	1.000
1⅝	2.844	2.969	1.219	1.125	2.469	2.562	1.344	1.094
1¾	3.062	3.203	1.312	1.203	2.656	2.750	1.438	1.172
	Cone Head				Pan Head			
½	0.875	0.922	0.438	0.406	0.800	0.844	0.350	0.328
⅝	1.094	1.141	0.547	0.516	1.000	1.047	0.438	0.406
¾	1.312	1.375	0.656	0.625	1.200	1.266	0.525	0.484
⅞	1.531	1.594	0.766	0.719	1.400	1.469	0.612	0.578
1	1.750	1.828	0.875	0.828	1.600	1.687	0.700	0.656
1⅛	1.969	2.063	0.984	0.938	1.800	1.891	0.788	0.734
1¼	2.188	2.281	1.094	1.031	2.000	2.094	0.875	0.812
1⅜	2.406	2.516	1.203	1.141	2.200	2.312	0.962	0.906
1½	2.625	2.734	1.312	1.250	2.400	2.516	1.050	0.984
1⅝	2.844	2.969	1.422	1.344	2.600	2.734	1.138	1.062
1¾	3.062	3.203	1.531	1.453	2.800	2.938	1.225	1.141

[a] Tolerance for diameter of body is plus and minus from nominal and for ½-in. size equals +0.020, –0.022; for sizes ⅝ to 1-in., incl., equals +0.030, –0.025; for sizes 1⅛ and 1¼-in. equals +0.035; –0.027; for sizes 1⅜ and 1½-in. equals +0.040, –0.030; for sizes 1⅝ and 1¾-in. equals +0.040, –0.037.

All dimensions are given in inches.
Note 1. Basic dimensions of head as manufactured.
Note 2. Dimensions of manufactured head after driving and also of driven head.
Note 3. Slight flat permissible within the specified head-height tolerance.

The following formulas give the basic dimensions for manufactured shapes: *Button Head,* $A = 1.750D$; $H = 0.750D$; $G = 0.885\,D$. *High Button Head,* $A = 1.500D + 0.031$; $H = 0.750D + 0.125$; $F = 0.750D + 0.281$; $G = 0.750D - 0.281$. *Cone Head,* $A = 1.750D$; $B = 0.938D$; $H = 0.875D$. *Pan Head,* $A = 1.600D$; $B = 1.000D$; $H = 0.700D$. Length L is measured parallel to the rivet axis, from the extreme end to the bearing surface plane for flat bearing surface head type rivets, or to the intersection of the head top surface with the head diameter for countersunk head-type rivets.

American National Standard Large Rivets — 2 *ANSI B18.1.2-1972 (R1995)*



Flat Counter-Sunk Head

Oval Counter-Sunk Head

Swell Neck

Flat and Oval Countersunk Head								
Body Diameter *D*				Head Dia. *A*		Head Depth *H*	Oval Crown height[a] *C*	Oval Crown Radius[a] *G*
Nominal[a]		Max.	Min.	Max.[b]	Min.[c]	Ref.		
1/2	0.500	0.520	0.478	0.936	0.872	0.260	0.095	1.125
5/8	0.625	0.655	0.600	1.194	1.112	0.339	0.119	1.406
3/4	0.750	0.780	0.725	1.421	1.322	0.400	0.142	1.688
7/8	0.875	0.905	0.850	1.647	1.532	0.460	0.166	1.969
1	1.000	1.030	0.975	1.873	1.745	0.520	0.190	2.250
1 1/8	1.125	1.160	1.098	2.114	1.973	0.589	0.214	2.531
1 1/4	1.250	1.285	1.223	2.340	2.199	0.650	0.238	2.812
1 3/8	1.375	1.415	1.345	2.567	2.426	0.710	0.261	3.094
1 1/2	1.500	1.540	1.470	2.793	2.652	0.771	0.285	3.375
1 5/8	1.625	1.665	1.588	3.019	2.878	0.831	0.309	3.656
1 3/4	1.750	1.790	1.713	3.262	3.121	0.901	0.332	3.938

[a] Basic dimension as manufactured. For tolerances see table footnote on page 1465.
[b] Sharp edged head.
[c] Rounded or flat edged irregularly shaped head (heads are not machined or trimmed).

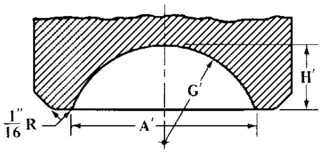
Swell Neck[a]						
Body Diameter *D*				Diameter Under Head *E*		Neck Length *K*[a]
Nominal[a]		Max.	Min.	Max. (Basic)	Min.	
1/2	0.500	0.520	0.478	0.563	0.543	0.250
5/8	0.625	0.655	0.600	0.688	0.658	0.312
3/4	0.750	0.780	0.725	0.813	0.783	0.375
7/8	0.875	0.905	0.850	0.938	0.908	0.438
1	1.000	1.030	0.975	1.063	1.033	0.500
1 1/8	1.125	1.160	1.098	1.188	1.153	0.562
1 1/4	1.250	1.285	1.223	1.313	1.278	0.625
1 3/8	1.375	1.415	1.345	1.438	1.398	0.688
1 1/2	1.500	1.540	1.470	1.563	1.523	0.750
1 5/8	1.625	1.665	1.588	1.688	1.648	0.812
1 3/4	1.750	1.790	1.713	1.813	1.773	0.875

[a] The swell neck is applicable to all standard forms of large rivets except the flat countersunk and oval countersunk head types.

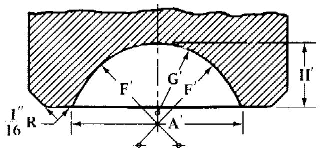
All dimensions are given in inches.

The following formulas give basic dimensions for manufactured shapes: *Flat Countersunk Head,* $A = 1.810D$; $H = 1.192(\text{Max } A - D)/2$; included angle Q of head = 78 degrees. *Oval Countersunk Head,* $A = 1.810D$; $H = 1.192(\text{Max } A - D)/2$; included angle of head = 78 degrees. *Swell Neck,* $E = D + 0.063$; $K = 0.500D$. Length L is measured parallel to the rivet axis, from the extreme end to the bearing surface plane for flat bearing surface head-type rivets, or to the intersection of the head top surface with the head diameter for countersunk head-type rivets.

American National Standard Dimensions for Hold-On (Dolly Bar) and Rivet Set Impression *ANSI B18.1.2-1972 (R1995)*

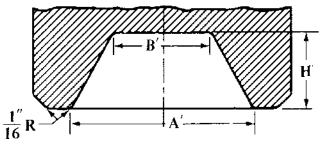


Button Head



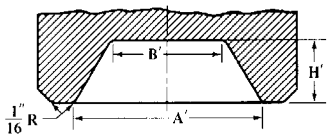
High Button Head

Rivet Body Dia.	Button Head			High Button Head			
	A′	*H′*	*G′*	*A′*	*H′*	*F′*	*G′*
½	0.906	0.312	0.484	0.859	0.344	0.562	0.375
⅝	1.125	0.406	0.594	1.047	0.422	0.672	0.453
¾	1.344	0.484	0.719	1.234	0.500	0.797	0.531
⅞	1.578	0.562	0.844	1.422	0.578	0.922	0.609
1	1.812	0.641	0.953	1.609	0.656	1.031	0.688
1⅛	2.031	0.719	1.078	1.797	0.719	1.156	0.766
1¼	2.250	0.797	1.188	1.984	0.797	1.266	0.844
1⅜	2.469	0.875	1.312	2.172	0.875	1.406	0.938
1½	2.703	0.953	1.438	2.344	0.953	1.500	1.000
1⅝	2.922	1.047	1.547	2.531	1.031	1.641	1.094
1¾	3.156	1.125	1.672	2.719	1.109	1.750	1.172



Cone Head

Rivet Body Dia.	Cone Head		
	A′	*B′*	*H′*
½	0.891	0.469	0.391
⅝	1.109	0.594	0.484
¾	1.328	0.703	0.578
⅞	1.562	0.828	0.688
1	1.781	0.938	0.781
1⅛	2.000	1.063	0.875
1¼	2.219	1.172	0.969
1⅜	2.453	1.297	1.078
1½	2.672	1.406	1.172
1⅝	2.891	1.531	1.266
1¾	3.109	1.641	1.375

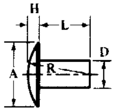


Pan Head

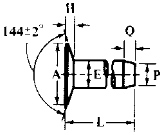
Rivet Body Dia.	Pan Head		
	A′	*B′*	*H′*
½	0.812	0.500	0.297
⅝	1.031	0.625	0.375
¾	1.234	0.750	0.453
⅞	1.438	0.875	0.531
1	1.641	1.000	0.609
1⅛	1.844	1.125	0.688
1¼	2.047	1.250	0.766
1⅜	2.250	1.375	0.844
1½	2.453	1.500	0.906
1⅝	2.656	1.625	0.984
1¾	2.875	1.750	1.063

All dimensions are given in inches.

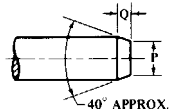
Table 1. American National Standard Small Solid Rivets
ANSI/ASME B18.1.1-1972 (R1995) and Appendix



Truss Head Rivets



Coopers Rivets



Truss Head Rivets[a]

Shank Dia.,[b] *D*		Head Dimensions: Dia., *A*		Height, *H*		Rad. *R*	Shank Dia.,[b] *D*		Head Dimensions: Dia., *A*		Height, *H*		Rad. *R*
Nominal		Max.	Min.	Max.	Min.	Approx.	Nominal		Max.	Min.	Max.	Min.	Approx.
3/32	0.094	0.226	0.206	0.038	0.026	0.239	9/32	0.281	0.661	0.631	0.103	0.085	0.706
1/8	0.125	0.297	0.277	0.048	0.036	0.314	5/16	0.312	0.732	0.702	0.113	0.095	0.784
5/32	0.156	0.368	0.348	0.059	0.045	0.392	11/32	0.344	0.806	0.776	0.124	0.104	0.862
3/16	0.188	0.442	0.422	0.069	0.055	0.470	3/8	0.375	0.878	0.848	0.135	0.115	0.942
7/32	0.219	0.515	0.495	0.080	0.066	0.555	13/32	0.406	0.949	0.919	0.145	0.123	1.028
1/4	0.250	0.590	0.560	0.091	0.075	0.628	7/16	0.438	1.020	0.990	0.157	0.135	1.098

[a] Length tolerance of rivets is + or − .016 inch. Approximate proportions of rivets: $A = 2.300 \times D$, $H = 0.330 \times D$, $R = 2.512 \times D$.

[b] Tolerances on the nominal shank diameter in inches are given for the following body diameter ranges: 3/32 to 5/32, plus 0.002, minus 0.004; 3/16 to 1/4, plus 0.003, minus 0.006; 9/32 to 11/32, plus 0.004, minus 0.008; and 3/8 to 7/16, plus 0.005, minus 0.010.

Coopers Rivets

Size No.[a]	Shank Diameter, *D*		Head Diameter, *A*		Head Height, *H*		Point Dimensions: Dia., *P*	Point Dimensions: Length, *Q*	Length, *L*	
	Max.	Min.	Max.	Min.	Max.	Min.	Nom.	Nom.	Max.	Min.
1 lb	0.111	0.105	0.291	0.271	0.045	0.031	Not Pointed		0.249	0.219
1 1/4 lb	0.122	0.116	0.324	0.302	0.050	0.036	Not Pointed		0.285	0.255
1 1/2 lb	0.132	0.126	0.324	0.302	0.050	0.036	Not Pointed		0.285	0.255
1 3/4 lb	0.136	0.130	0.324	0.302	0.052	0.034	Not Pointed		0.318	0.284
2 lb	0.142	0.136	0.355	0.333	0.056	0.038	Not Pointed		0.322	0.288
3 lb	0.158	0.152	0.386	0.364	0.058	0.040	0.123	0.062	0.387	0.353
4 lb	0.168	0.159	0.388	0.362	0.058	0.040	0.130	0.062	0.418	0.388
5 lb	0.183	0.174	0.419	0.393	0.063	0.045	0.144	0.062	0.454	0.420
6 lb	0.206	0.197	0.482	0.456	0.073	0.051	0.160	0.094	0.498	0.457
7 lb	0.223	0.214	0.513	0.487	0.076	0.054	0.175	0.094	0.561	0.523
8 lb	0.241	0.232	0.546	0.516	0.081	0.059	0.182	0.094	0.597	0.559
9 lb	0.248	0.239	0.578	0.548	0.085	0.063	0.197	0.094	0.601	0.563
10 lb	0.253	0.244	0.578	0.548	0.085	0.063	0.197	0.094	0.632	0.594
12 lb	0.263	0.251	0.580	0.546	0.086	0.060	0.214	0.094	0.633	0.575
14 lb	0.275	0.263	0.611	0.577	0.091	0.065	0.223	0.094	0.670	0.612
16 lb	0.285	0.273	0.611	0.577	0.089	0.063	0.223	0.094	0.699	0.641
18 lb	0.285	0.273	0.642	0.608	0.108	0.082	0.230	0.125	0.749	0.691
20 lb	0.316	0.304	0.705	0.671	0.128	0.102	0.250	0.125	0.769	0.711
3/8 in.	0.380	0.365	0.800	0.762	0.136	0.106	0.312	0.125	0.840	0.778

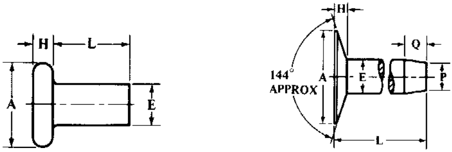
[a] Size numbers in pounds refer to the approximate weight of 1000 rivets.

When specified American National Standard Small Solid Rivets may be obtained with points. Point dimensions for belt and coopers rivets are given in the accompanying tables. Formulas for calculating point dimensions of other rivets are given with the diagram alongside

All dimensions in inches except where otherwise noted.

Table 2. American National Standard Small Solid Rivets
ANSI/ASME B18.1.1-1972 (R1995)

Tinners Rivets



Belt Rivets

Tinners Rivets									
Size No.[a]	Shank Diameter, *E*		Head Dia., *A*		Head Height, *H*		Length, *L*		
	Max.	Min.	Max.	Min.	Max.	Min.	Nom.	Max.	Min.
6 oz.	0.081	0.075	0.213	0.193	0.028	0.016	1/8	0.135	0.115
8 oz.	0.091	0.085	0.225	0.205	0.036	0.024	5/32	0.166	0.146
10 oz.	0.097	0.091	0.250	0.230	0.037	0.025	11/64	0.182	0.162
12 oz.	0.107	0.101	0.265	0.245	0.037	0.025	3/16	0.198	0.178
14 oz.	0.111	0.105	0.275	0.255	0.038	0.026	3/16	0.198	0.178
1 lb	0.113	0.107	0.285	0.265	0.040	0.028	13/64	0.213	0.193
1 1/4 lb	0.122	0.116	0.295	0.275	0.045	0.033	7/32	0.229	0.209
1 1/2 lb	0.132	0.126	0.316	0.294	0.046	0.034	15/64	0.244	0.224
1 3/4 lb	0.136	0.130	0.331	0.309	0.049	0.035	1/4	0.260	0.240
2 lb	0.146	0.140	0.341	0.319	0.050	0.036	17/64	0.276	0.256
2 1/2 lb	0.150	0.144	0.311	0.289	0.069	0.055	9/32	0.291	0.271
3 lb	0.163	0.154	0.329	0.303	0.073	0.059	5/16	0.323	0.303
3 1/2 lb	0.168	0.159	0.348	0.322	0.074	0.060	21/64	0.338	0.318
4 lb	0.179	0.170	0.368	0.342	0.076	0.062	11/32	0.354	0.334
5 lb	0.190	0.181	0.388	0.362	0.084	0.070	3/8	0.385	0.365
6 lb	0.206	0.197	0.419	0.393	0.090	0.076	25/64	0.401	0.381
7 lb	0.223	0.214	0.431	0.405	0.094	0.080	13/32	0.416	0.396
8 lb	0.227	0.218	0.475	0.445	0.101	0.085	7/16	0.448	0.428
9 lb	0.241	0.232	0.490	0.460	0.103	0.087	29/64	0.463	0.443
10 lb	0.241	0.232	0.505	0.475	0.104	0.088	15/32	0.479	0.459
12 lb	0.263	0.251	0.532	0.498	0.108	0.090	1/2	0.510	0.490
14 lb	0.288	0.276	0.577	0.543	0.113	0.095	33/64	0.525	0.505
16 lb	0.304	0.292	0.597	0.563	0.128	0.110	17/32	0.541	0.521
18 lb	0.347	0.335	0.706	0.668	0.156	0.136	19/32	0.603	0.583

[a] Size numbers refer to the approximate weight of 1000 rivets.

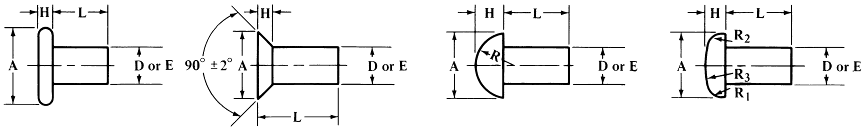
Belt Rivets[a]								
Size No.[b]	Shank Diameter, *E*		Head Dia., *A*		Head Height, *H*		Point Dimensions	
							Dia., *P*	Length, *Q*
	Max.	Min.	Max.	Min.	Max.	Min.	Nominal	Nominal
14	0.085	0.079	0.260	0.240	0.042	0.030	0.065	0.078
13	0.097	0.091	0.322	0.302	0.051	0.039	0.073	0.078
12	0.111	0.105	0.353	0.333	0.054	0.040	0.083	0.078
11	0.122	0.116	0.383	0.363	0.059	0.045	0.097	0.078
10	0.136	0.130	0.417	0.395	0.065	0.047	0.109	0.094
9	0.150	0.144	0.448	0.426	0.069	0.051	0.122	0.094
8	0.167	0.161	0.481	0.455	0.072	0.054	0.135	0.094
7	0.183	0.174	0.513	0.487	0.075	0.056	0.151	0.125
6	0.206	0.197	0.606	0.580	0.090	0.068	0.165	0.125
5	0.223	0.214	0.700	0.674	0.105	0.083	0.185	0.125
4	0.241	0.232	0.921	0.893	0.138	0.116	0.204	0.141

[a] Length tolerance on belt rivets is plus 0.031 inch, minus 0 inch.
[b] Size number refers to the Stub's iron wire gage number of the stock used in the shank of the rivet.

All dimensions in inches.

Note: American National Standard Small Solid Rivets may be obtained with or without points. Point proportions are given in the diagram in Table 1.

Table 3. American National Standard Small Solid Rivets *ANSI/ASME B18.1.1-1972 (R1995) and Appendix*



Shank Diameter				Flat Head				Flat Countersunk Head			Button Head					Pan Head						
				Head Dimensions																		
D		*E*		Dia., *A*		Height, *H*		Dia., *A* Sharp		Height[a] *H*	Dia., *A*		Height, *H*		Radius, *R*	Dia., *A*		Height, *H*		Radii *R1*	*R2*	*R3*
Nominal		Max.	Min.	Max.	Min.	Max.	Min.	Max.[b]	Min.[c]	Ref.	Max.	Min.	Max.	Min.	Approx.	Max.	Min.	Max.	Min.	Approximate		
1⁄16	0.062	0.064	0.059	0.140	0.120	0.027	0.017	0.118	0.110	0.027	0.122	0.102	0.052	0.042	0.055	0.118	0.098	0.040	0.030	0.019	0.052	0.217
3⁄32	0.094	0.096	0.090	0.200	0.180	0.038	0.026	0.176	0.163	0.040	0.182	0.162	0.077	0.065	0.084	0.173	0.153	0.060	0.048	0.030	0.080	0.326
1⁄8	0.125	0.127	0.121	0.260	0.240	0.048	0.036	0.235	0.217	0.053	0.235	0.215	0.100	0.088	0.111	0.225	0.205	0.078	0.066	0.039	0.106	0.429
5⁄32	0.156	0.158	0.152	0.323	0.301	0.059	0.045	0.293	0.272	0.066	0.290	0.268	0.124	0.110	0.138	0.279	0.257	0.096	0.082	0.049	0.133	0.535
3⁄16	0.188	0.191	0.182	0.387	0.361	0.069	0.055	0.351	0.326	0.079	0.348	0.322	0.147	0.133	0.166	0.334	0.308	0.114	0.100	0.059	0.159	0.641
7⁄32	0.219	0.222	0.213	0.453	0.427	0.080	0.065	0.413	0.384	0.094	0.405	0.379	0.172	0.158	0.195	0.391	0.365	0.133	0.119	0.069	0.186	0.754
1⁄4	0.250	0.253	0.244	0.515	0.485	0.091	0.075	0.469	0.437	0.106	0.460	0.430	0.196	0.180	0.221	0.444	0.414	0.151	0.135	0.079	0.213	0.858
9⁄32	0.281	0.285	0.273	0.579	0.545	0.103	0.085	0.528	0.491	0.119	0.518	0.484	0.220	0.202	0.249	0.499	0.465	0.170	0.152	0.088	0.239	0.963
5⁄16	0.312	0.316	0.304	0.641	0.607	0.113	0.095	0.588	0.547	0.133	0.572	0.538	0.243	0.225	0.276	0.552	0.518	0.187	0.169	0.098	0.266	1.070
11⁄32	0.344	0.348	0.336	0.705	0.667	0.124	0.104	0.646	0.602	0.146	0.630	0.592	0.267	0.247	0.304	0.608	0.570	0.206	0.186	0.108	0.292	1.176
3⁄8	0.375	0.380	0.365	0.769	0.731	0.135	0.115	0.704	0.656	0.159	0.684	0.646	0.291	0.271	0.332	0.663	0.625	0.225	0.205	0.118	0.319	1.286
13⁄32	0.406	0.411	0.396	0.834	0.790	0.146	0.124	0.763	0.710	0.172	0.743	0.699	0.316	0.294	0.358	0.719	0.675	0.243	0.221	0.127	0.345	1.392
7⁄16	0.438	0.443	0.428	0.896	0.852	0.157	0.135	0.823	0.765	0.186	0.798	0.754	0.339	0.317	0.387	0.772	0.728	0.261	0.239	0.137	0.372	1.500

[a] Given for reference purposes only. Variations in this dimension are controlled by the head and shank diameters and the included angle of the head.

[b] Tabulated maximum values calculated on basic diameter of rivet and 92° included angle extended to a sharp edge.

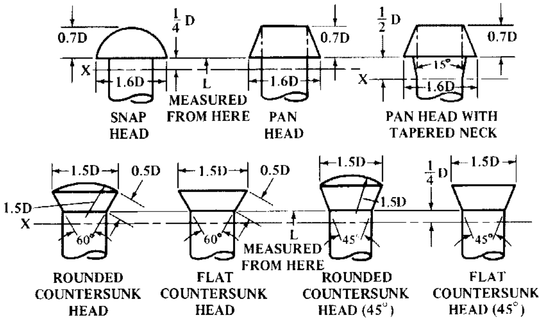
[c] Minimum of rounded or flat-edged irregular-shaped head. Rivet heads are not machined or trimmed and the circumference may be irregular and edges rounded or flat.

All dimensions in inches. Length tolerance of all rivets is plus or minus 0.016 inch. Approximate proportions of rivets: flat head, $A = 2.00 \times D$, $H = 0.33\ D$; flat countersunk head, $A = 1.850 \times D$, $H = 0.425 \times D$; button head, $A = 1.750 \times D$, $H = 0.750 \times D$, $R = 0.885 \times D$; pan head, $A = 1.720 \times D$, $H = 0.570 \times D$, $R1 = 0.314 \times D$, $R2 = 0.850 \times D$, $R3 = 3.430 \times D$.

Note: ANSI Small Solid Rivets may be obtained with or without points. Point proportions are given in the diagram in Table 1.

Head Dimensions and Diameters of British Standard Rivets

BS 275:1927 (obsolescent) This standard does not apply to Boiler Rivets



Nominal Rivet Diameter, *D*	Shank Diameter[a]				
	At Position *X*[b]		At Position *Y*[b]		At Position *Z*[b]
	Minimum	Maximum	Minimum	Maximum	Minimum
1/2	1/2	17/32	31/64	1/2	31/64
9/16[c]	9/16	19/32	35/64	9/16	35/64
5/8	5/8	21/32	39/64	5/8	39/64
11/16[c]	11/16	23/32	43/64	11/16	43/64
3/4	3/4	25/32	47/64	3/4	47/64
13/16[c]	13/16	27/32	51/64	13/16	51/64
7/8	7/8	29/32	55/64	7/8	55/64
15/16[c]	15/16	31/32	59/64	15/16	59/64
1	1	1 1/32	63/64	1	63/64
1 1/16[c]	1 1/16	1 3/32	1 3/64	1 1/16	1 3/64
1 1/8	1 1/8	1 5/32	1 7/64	1 1/8	1 7/64
1 3/16[c]	1 3/16	1 7/32	1 11/64	1 3/16	1 11/64
1 1/4	1 1/4	1 9/32	1 15/64	1 1/4	1 15/64
1 5/16[c]	1 5/16	1 11/32	1 19/64	1 5/16	1 19/64
1 3/8	1 3/8	1 13/32	1 23/64	1 3/8	1 23/64
1 7/16[c]	1 7/16	1 15/32	1 27/64	1 7/16	1 27/64
1 1/2	1 1/2	1 17/32	1 31/64	1 1/2	1 31/64
1 9/16[c]	1 9/16	1 19/32	1 35/64	1 9/16	1 35/64
1 5/8	1 5/8	1 21/32	1 39/64	1 5/8	1 39/64
1 11/16[c]	1 11/16	1 23/32	1 43/64	1 11/16	1 43/64
1 3/4	1 3/4	1 25/32	1 47/64	1 3/4	1 47/64

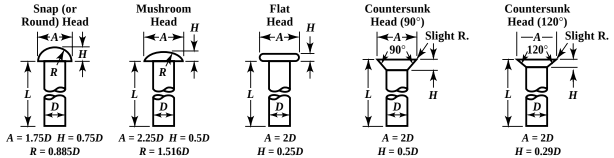
[a] Tolerances of the rivet diameter are as follows: at position *X*, plus 1/32 inch, minus zero; at position *Y*, plus zero, minus 1/64 inch; at position *Z*, minus 1/64 inch but in no case shall the difference between the diameters at positions *X* and *Y* exceed 1/32 inch, nor shall the diameter of the shank between positions *X* and *Y* be less than the minimum diameter specified at position *Y*.

[b] The location of positions *Y* and *Z* are as follows: Position *Y* is located 1/2*D* from the end of the rivet for rivet lengths 5 diameters long and under. For longer rivets, position *Y* is located 4 1/2 *D* from the head of the rivet. Position *Z* (found only on rivets longer than 5*D*) is located 1/2*D* from the end of the rivet.

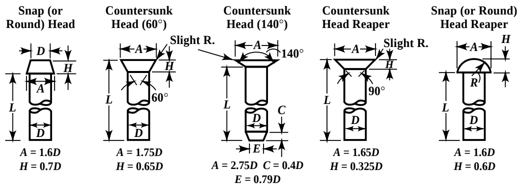
[c] At the recommendation of the British Standards Institution, these sizes are to be dispensed with wherever possible.

All dimensions that are tabulated are given in inches.

British Standard Small Rivets for General Purposes *BS 641:1951 (obsolescent)*



Nom. Diam. D	Snap (or Round Head)			Mushroom Head			Flat Head		Countersunk Head (90°)		Countersunk Head (120°)	
	Head Dimensions											
	Diam. A	Ht. H	Rad. R	Diam. A	Ht. H	Rad. R	Diam. A	Ht. H	Diam. A	Ht. H	Diam. A	Ht. H
1⁄16	0.109	0.047	0.055	0.141	0.031	0.095	0.125	0.016	0.125	0.031	…	…
3⁄32	0.164	0.070	0.083	0.211	0.047	0.142	0.188	0.023	0.188	0.047	…	…
1⁄8	0.219	0.094	0.111	0.281	0.063	0.189	0.250	0.031	0.250	0.063	0.250	0.036
5⁄32	0.273	0.117	0.138	0.352	0.078	0.237	0.313	0.039	0.313	0.078	…	…
3⁄16	0.328	0.141	0.166	0.422	0.094	0.284	0.375	0.047	0.375	0.094	0.375	0.054
1⁄4	0.438	0.188	0.221	0.563	0.125	0.379	0.500	0.063	0.500	0.125	0.500	0.073
5⁄16	0.547	0.234	0.277	0.703	0.156	0.474	0.625	0.078	0.625	0.156	0.625	0.091
3⁄8	0.656	0.281	0.332	0.844	0.188	0.568	0.750	0.094	0.750	0.188	0.750	0.109
7⁄16	0.766	0.328	0.387	0.984	0.219	0.663	0.875	0.109	0.875	0.219	…	…



Nominal Diameter D		Pan Head		Countersunk Head (60°)		Countersunk Head (140°)			Countersunk Head Reaper		Snap (or Round) Head Reaper	
		Head Dimensions										
Inch	Gage No.[a]	Diam. A	Ht. H	Diam. A	Ht. H	Diam. A	Ht. C	Diam. E	Diam. A	Ht. H	Diam. A	Ht. H
0.104	12	…	…	…	…	0.286	0.042	0.082	…	…	…	…
0.116	11	…	…	…	…	0.319	0.046	0.092	…	…	…	…
0.128	10	…	…	…	…	0.352	0.051	0.101	…	…	…	…
0.144	9	…	…	…	…	0.396	0.058	0.114	…	…	…	…
0.160	8	…	…	…	…	0.440	0.064	0.126	…	…	…	…
0.176	7	…	…	…	…	0.484	0.070	0.139	…	…	…	…
3⁄16	…	0.300	0.131	0.328	0.122	…	…	…	…	…	…	…
0.192	6	…	…	…	…	0.528	0.077	0.152	0.317	0.062	0.307	0.115
0.202	…	…	…	…	…	…	…	…	0.333	0.066	0.323	0.121
0.212	5	…	…	…	…	0.583	0.085	0.167	0.350	0.069	0.339	0.127
0.232	4	…	…	…	…	0.638	0.093	0.183	0.383	0.075	0.371	0.139
1⁄4	…	0.400	0.175	0.438	0.162	0.688	0.100	0.198	…	…	…	…
0.252	3	…	…	…	…	…	…	…	0.416	0.082	0.403	0.151
5⁄16	…	0.500	0.219	0.547	0.203	0.859	0.125	0.247	…	…	…	…
3⁄8	…	0.600	0.263	0.656	0.244	10.031	0.150	0.296	…	…	…	…
7⁄16	…	0.700	0.306	0.766	0.284	…	…	…	…	…	…	…

[a] Gage numbers are British Standard Wire Gage (S.W.G.) numbers.

All dimensions in inches unless specified otherwise0.

British Standard Rivets for General Engineering.—Dimensions in metric units of rivets for general engineering purposes are given in this British Standard, BS 4620;1970, which is based on ISO Recommendation ISO/R 1051. The snap head rivet dimensions of 14 millimeters and above are taken from the German Standard DIN 124, Round Head Rivets for Steel Structures. The shapes of heads have been restricted to those in common use in the United Kingdom. Table 2 shows the rivet dimensions. Table 1 shows a tentative range of preferred nominal lengths as given in an appendix to the Standard. It is stated that these lengths will be reviewed in the light of usage. The rivets are made by cold or hot forging methods from mild steel, copper, brass, pure aluminum, aluminum alloys, or other suitable metal. It is stated that the radius under the head of a rivet shall run smoothly into the face of the head and shank without step or discontinuity.

In this Standard, the following definitions apply: 1) *Nominal diameter:* The diameter of the shank; 2) *Nominal length of rivets other than countersunk or raised countersunk rivets:* The length from the underside of the head to the end of the shank; 3) *Nominal length of countersunk and raised countersunk rivets:* The distance from the periphery of the head to the end of the rivet measured parallel to the axis of the rivet; and 4) *Manufactured head:* The head on the rivet as received from the manufacturer.

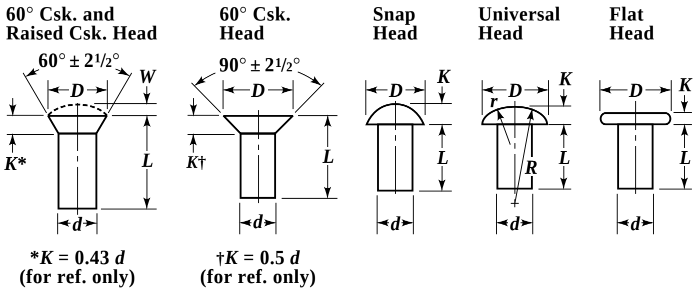
Table 1. Tentative Range of Lengths for Rivets *Appendix to BS 4620:1970 (1998)*

Nom. Shank Dia.	Nominal Length																				
	3	4	5	6	8	10	12	14	16	(18)	20	(22)	25	(28)	30	(32)	35	(38)	40	45	..
1	X	X	X	X	X	X	X	X	X	..	X	..	..	..	..	..	..	..	..	..	..
1.2	X	X	X	X	X	X	X	X	X	..	X	..	..	..	..	..	..	..	..	..	..
1.6	X	X	X	X	X	X	X	X	X	X	..	X	X	..	..	..	..	..	..	..	..
2	X	X	X	X	X	X	X	X	X	..	X	X	X	..	..	..	..	..	..	..	..
2.5	X	X	X	X	X	X	X	X	X	X	..	X	X	..	..	..	..	..	..	..	..
3	..	X	X	X	X	X	X	X	X	X	X	X	X	..	..	..	..	..	..	..	..
(3.5)	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..
4	..	..	..	X	X	X	X	X	X	X	..	X	X	..	..	..	..	..	..	..	..
5	..	..	..	X	X	X	X	X	X	X	X	X	X	X	X	..	X	X	..	..	..
6	..	..	..	X	X	X	X	X	X	X	X	X	X	X	X	..	X	X	..	X	..
Nom. Shank Dia.	Nominal Length																				
	10	12	14	16	(18)	20	(22)	25	(28)	30	(32)	35	(38)	40	45	50	55	60	65	70	75
(7)	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..
8	X	X	X	X	X	X	X	X	X	X	..	X	X	..	X	X	..	..	..	..	..
10	..	..	X	X	X	X	X	X	X	X	..	X	X	X	X	X	..	..	..	..	..
12	..	..	..	..	..	..	..	X	..	X	..	X	..	X	..	X	X	..	..	..	X
(14)	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..
16	..	..	..	..	..	..	..	..	..	..	..	..	..	X	X	X	X	X	X	..	X
Nom. Shank Dia.	Nominal Length																				
	45	50	55	60	65	70	75	80	85	90	(95)	100	(105)	110	(115)	120	(125)	130	140	150	160
(18)	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..
20	X	..	..	X	..	X	..	X	..	..	..	..	..	..	..	..	..	..	..	..	..
(22)	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..
24	..	..	..	..	X	..	X	..	X	X	..	X	..	..	..	..	..	..	..	..	..
(27)	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..
30	..	..	..	..	..	..	..	..	X	X	..	X	..	X	..	X	..	..	..	..	..
(33)	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..
36	..	..	..	..	..	..	..	..	..	..	..	X	..	X	..	X	..	X	..	..	..
(39)	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	X	..	X	X	X	X

All dimensions are in millimeters.

Note: Sizes and lengths shown in parenthesis are nonpreferred and should be avoided if possible.

Table 2. British Standard Rivets for General Engineering Purposes
BS 4620:1970 (1998)



Hot Forged Rivets

Nom. Shank Diam. d	Tol. on Dia. d	60° Csk. and Raised Csk. Head: Nom. Dia. D	60° Csk. and Raised Csk. Head: Height of Raise W	Snap Head: Nom. Dia. D	Snap Head: Nom. Depth K	Universal Head: Nom. Dia. D	Universal Head: Nom. Depth K	Universal Head: Rad. R	Universal Head: Rad. r
(14)		21	2.8	22	9	28	5.6	42	8.4
16	±0.43	24	3.2	25	10	32	6.4	48	9.6
(18)		27	3.6	28	11.5	36	7.2	54	11
20		30	4.0	32	13	40	8.0	60	12
(22)	±0.52	33	4.4	36	14	44	8.8	66	13
24		36	4.8	40	16	48	9.6	72	14
(27)		40	5.4	43	17	54	10.8	81	16
30		45	6.0	48	19	60	12.0	90	18
(33)	±0.62	50	6.6	53	21	66	13.2	99	20
36		55	7.2	58	23	72	14.4	108	22
(39)		59	7.8	62	25	78	15.6	117	23

Cold Forged Rivets

Nom. Shank Dia. d	Tol. on Dia. d	90° Csk. Head: Nom. Dia. D	Snap Head: Nom. Dia. D	Snap Head: Nom. Dia. K	Universal Head: Nom. Dia. D	Universal Head: Nom. Depth K	Universal Head: Rad. R	Universal Head: Rad. r	Flat Head: Nom. Dia. D	Flat Head: Nom. Depth K
1		2	1.8	0.6	2	0.4	3.0	0.6	2	0.25
1.2		2.4	2.1	0.7	2.4	0.5	3.6	0.7	2.4	0.3
1.6	±0.07	3.2	2.8	1.0	3.2	0.6	4.8	1.0	3.2	0.4
2		4	3.5	1.2	4	0.8	6.0	1.2	4	0.5
2.5		5	4.4	1.5	5	1.0	7.5	1.5	5	0.6
3		6	5.3	1.8	6	1.2	9.0	1.8	6	0.8
(3.5)		7	6.1	2.1	7	1.4	10.5	2.1	7	0.9
4	±0.09	8	7	2.4	8	1.6	12	2.4	8	1.0
5		10	8.8	3.0	10	2.0	15	3.0	10	1.3
6		12	10.5	3.6	12	2.4	18	3.6	12	1.5
(7)		14	12.3	4.2	14	2.8	21	4.2	14	1.8
8	±0.11	16	14	4.8	16	3.2	24	4.8	16	2
10		20	18	6.0	20	4.0	30	6	20	2.5
12		24	21	7.2	24	4.8	36	7.2	…	…
(14)	±0.14	…	25	8.4	28	5.6	42	8.4	…	…
16		…	28	9.6	32	6.4	48	9.6	…	…

All dimensions are in millimeters. Sizes shown in parentheses are nonpreferred.

TORQUE AND TENSION IN FASTENERS

Tightening Bolts: Bolts are often tightened by applying torque to the head or nut, which causes the bolt to stretch. The stretching results in bolt tension or preload, which is the force that holds a joint together. Torque is relatively easy to measure with a torque wrench, so it is the most frequently used indicator of bolt tension. Unfortunately, a torque wrench does not measure bolt tension accurately, mainly because it does not take friction into account. The friction depends on bolt, nut, and washer material, surface smoothness, machining accuracy, degree of lubrication, and the number of times a bolt has been installed. Fastener manufacturers often provide information for determining torque requirements for tightening various bolts, accounting for friction and other effects. If this information is not available, the methods described in what follows give general guidelines for determining how much tension should be present in a bolt, and how much torque may need to be applied to arrive at that tension.

High preload tension helps keep bolts tight, increases joint strength, creates friction between parts to resist shear, and improves the fatigue resistance of bolted connections. The recommended preload F_i, which can be used for either static (stationary) or fatigue (alternating) applications, can be determined from: $F_i = 0.75 \times A_t \times S_p$ for reusable connections, and $F_i = 0.9 \times A_t \times S_p$ for permanent connections. In these formulas, F_i is the bolt preload, A_t is the tensile stress area of the bolt, and S_p is the proof strength of the bolt. Determine A_t from screw-thread tables or by means of formulas in this section. Proof strength S_p of commonly used ASTM and SAE steel fasteners is given in this section and in the section on metric screws and bolts for those fasteners. For other materials, an approximate value of proof strength can be obtained from: $S_p = 0.85 \times S_y$, where S_y is the yield strength of the material. Soft materials should not be used for threaded fasteners.

Once the required preload has been determined, one of the best ways to be sure that a bolt is properly tensioned is to measure its tension directly with a strain gage. Next best is to measure the change in length (elongation) of the bolt during tightening, using a micrometer or dial indicator. Each of the following two formulas calculates the required change in length of a bolt needed to make the bolt tension equal to the recommended preload. The change in length δ of the bolt is given by:

$$\delta = F_i \times \frac{A_d \times l_t + A_t \times l_d}{A_d \times A_t \times E} \quad (1) \qquad \text{or} \qquad \delta = \frac{F_i \times l}{A \times E} \quad (2)$$

In Equation (1), F_i is the bolt preload; A_d is the major-diameter area of the bolt; A_t is the tensile-stress area of the bolt; E is the bolt modulus of elasticity; l_t is the length of the threaded portion of the fastener within the grip; and l_d is the length of the unthreaded portion of the grip. Here, the grip is defined as the total thickness of the clamped material. Equation (2) is a simplified formula for use when the area of the fastener is constant, and gives approximately the same results as Equation (1). In Equation (2), l is the bolt length; A is the bolt area; and δ, F_i, and E are as described before.

If measuring bolt elongation is not possible, the torque necessary to tighten the bolt must be estimated. If the recommended preload is known, use the following general relation for the torque: $T = K \times F_i \times d$, where T is the wrench torque, K is a constant that depends on the bolt material and size, F_i is the preload, and d is the nominal bolt diameter. A value of K = 0.2 may be used in this equation for mild-steel bolts in the size range of ¼ to 1 inch. For other steel bolts, use the following values of K: nonplated black finish, 0.3; zinc-plated, 0.2; lubricated, 0.18; cadmium-plated, 0.16. Check with bolt manufacturers and suppliers for values of K to use with bolts of other sizes and materials.

The proper torque to use for tightening bolts in sizes up to about ½ inch may also be determined by trial. Test a bolt by measuring the amount of torque required to fracture it (use bolt, nut, and washers equivalent to those chosen for the real application). Then, use a tightening torque of about 50 to 60 per cent of the fracture torque determined by the test. The tension in a bolt tightened using this procedure will be about 60 to 70 per cent of the elastic limit (yield strength) of the bolt material.

The table that follows can be used to get a rough idea of the torque necessary to properly tension a bolt by using the bolt diameter d and the coefficients b and m from the table; the approximate tightening torque T in ft-lb for the listed fasteners is obtained by solving the equation $T = 10^{b+m \log d}$. This equation is approximate, for use with unlubricated fasteners as supplied by the mill. See the notes at the end of the table for more details on using the equation.

Wrench Torque $T = 10^{b+m \log d}$ for Steel Bolts, Studs, and Cap Screws (*see notes*)

Fastener Grade(s)	Bolt Diameter d (in.)	m	b
SAE 2, ASTM A307	¼ to 3	2.940	2.533
SAE 3	¼ to 3	3.060	2.775
ASTM A-449, A-354-BB, SAE 5	¼ to 3	2.965	2.759
ASTM A-325[a]	½ to 1½	2.922	2.893
ASTM A-354-BC	¼ to ⅝	3.046	2.837
SAE 6, SAE 7	¼ to 3	3.095	2.948
SAE 8	¼ to 3	3.095	2.983
ASTM A-354-BD, ASTM A490[a]	⅜ to 1¾	3.092	3.057
Socket Head Cap Screws	¼ to 3	3.096	3.014

[a] Values for permanent fastenings on steel structures.

Usage: Values calculated using the preceding equation are for standard, unplated industrial fasteners as received from the manufacturer; for cadmium-plated cap screws, multiply the torque by 0.9; for cadmium-plated nuts and bolts, multiply the torque by 0.8; for fasteners used with special lubricants, multiply the torque by 0.9; for studs, use cap screw values for equivalent grade.

Preload for Bolts in Loaded Joints.—The following recommendations are based on MIL-HDBK-60, a subsection of FED-STD-H28, Screw Thread Standards for Federal Service. Generally, bolt preload in joints should be high enough to maintain joint members in contact and in compression. Loss of compression in a joint may result in leakage of pressurized fluids past compression gaskets, loosening of fasteners under conditions of cyclic loading, and reduction of fastener fatigue life.

The relationship between fastener fatigue life and fastener preload is illustrated by Fig. 1. An axially loaded bolted joint in which there is no bolt preload is represented by line OAB, that is, the bolt load is equal to the joint load. When joint load varies between P_a and P_b, the bolt load varies accordingly between P_{Ba} and P_{Bb}. However, if preload P_{B1} is applied to the bolt, the joint is compressed and bolt load changes more slowly than the joint load (indicated by line P_{B1}A, whose slope is less than line OAB) because some of the load is absorbed as a reduction of compression in the joint. Thus, the axial load applied to the joint varies between $P_{Ba'}$ and $P_{Bb'}$ as joint load varies between P_a and P_b. This condition results in a considerable reduction in cyclic bolt-load variation and thereby increases the fatigue life of the fastener.

Preload for Bolts In Shear.—In shear-loaded joints, with members that slide, the joint members transmit shear loads to the fasteners in the joint and the preload must be sufficient to hold the joint members in contact. In joints that do not slide (i.e., there is no relative motion between joint members), shear loads are transmitted within the joint by frictional forces that mainly result from the preload. Therefore, preload must be great enough for the resulting friction forces to be greater than the applied shear force. With high applied shear loads, the shear stress induced in the fastener during application of the preload must also be considered in the bolted-joint design. Joints with combined axial and shear loads must be analyzed to ensure that the bolts will not fail in either tension or shear.

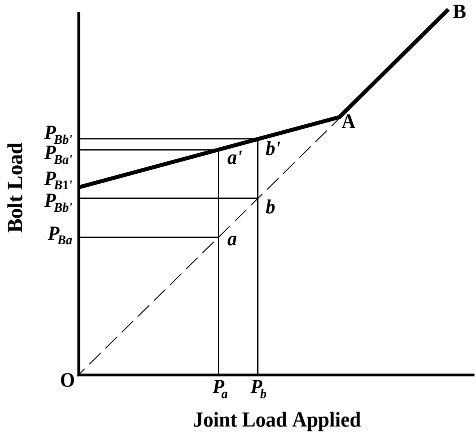


Fig. 1. Bolt Load in a Joint with Applied Axial Load

General Application of Preload.—Preload values should be based on joint requirements, as outlined before. Fastener applications are generally designed for maximum utilization of the fastener material; that is to say, the fastener size is the minimum required to perform its function and a maximum safe preload is generally applied to it. However, if a low-strength fastener is replaced by one of higher strength, for the sake of convenience or standardization, the preload in the replacement should not be increased beyond that required in the original fastener.

To utilize the maximum amount of bolt strength, bolts are sometimes tightened to or beyond the yield point of the material. This practice is generally limited to ductile materials, where there is considerable difference between the yield strength and the ultimate (breaking) strength, because low-ductility materials are more likely to fail due to unexpected overloads when preloaded to yield. Joints designed for primarily static load conditions that use ductile bolts, with a yield strain that is relatively far from the strain at fracture, are often preloaded above the yield point of the bolt material. Methods for tightening up to and beyond the yield point include tightening by feel without special tools, and the use of electronic equipment designed to compare the applied torque with the angular rotation of the fastener and detect changes that occur in the elastic properties of fasteners at yield.

Bolt loads are maintained below the yield point in joints subjected to cyclic loading and in joints using bolts of high-strength material where the yield strain is close to the strain at fracture. For these conditions, the maximum preloads generally fall within the following ranges: 50 to 80 per cent of the minimum tensile ultimate strength; 75 to 90 per cent of the minimum tensile yield strength or proof load; or 100 per cent of the observed proportional limit or onset of yield.

Bolt heads, driving recesses (in socket screws, for example), and the juncture of head and shank must be sufficiently strong to withstand the preload and any additional stress encountered during tightening. There must also be sufficient thread to prevent stripping (generally, at least three fully engaged threads). Materials susceptible to stress-corrosion cracking may require further preload limitations.

Preload Adjustments.—Preloads may be applied directly by axial loading or indirectly by turning of the nut or bolt. When preload is applied by turning of nuts or bolts, a torsion load component is added to the desired axial bolt load. This combined loading increases the tensile stress on the bolt. It is frequently assumed that the additional torsion load component dissipates quickly after the driving force is removed and, therefore, can be largely ignored. This assumption may be reasonable for fasteners loaded near to or beyond yield strength, but for critical applications where bolt tension must be maintained below yield, it is important to adjust the axial tension requirements to include the effects of the preload torsion. For this adjustment, the combined tensile stress (*von Mises* stress) F_{tc} in psi (MPa) can be calculated from the following:

$$F_{tc} = \sqrt{F_t^2 + 3F_s^2} \tag{3}$$

where F_t is the axial applied tensile stress in psi (MPa), and F_s is the shear stress in psi (MPa) caused by the torsion load application.

Some of the torsion load on a bolt, acquired when applying a preload, may be released by springback when the wrenching torque is removed. The amount of relaxation depends on the friction under the bolt head or nut. With controlled back turning of the nut, the torsional load may be reduced or eliminated without loss of axial load, reducing bolt stress and lowering creep and fatigue potential. However, calculation and control of the back-turn angle is difficult, so this method has limited application and cannot be used for short bolts because of the small angles involved.

For relatively soft work-hardenable materials, tightening bolts in a joint slightly beyond yield will work-harden the bolt to some degree. Back turning of the bolt to the desired tension will reduce embedment and metal flow and improve resistance to preload loss.

The following formula for use with single-start Unified inch screw threads calculates the combined tensile stress, F_{tc}:

$$F_{tc} = F_t\sqrt{1 + 3\left(\frac{1.96 + 2.31\mu}{1 - 0.325P/d_2} - 1.96\right)^2} \tag{4}$$

Single-start UNJ screw threads in accordance with MIL-S-8879 have a thread stress diameter equal to the bolt pitch diameter. For these threads, F_{tc} can be calculated from:

$$F_{tc} = F_t\sqrt{1 + 3\left(\frac{0.637P}{d_2} + 2.31\mu\right)^2} \tag{5}$$

where μ is the coefficient of friction between threads, P is the thread pitch ($P = 1/n$, and n is the number of threads per inch), and d_2 is the bolt-thread pitch diameter in inches. Both Equations (2) and (3) are derived from Equation (1); thus, the quantity within the radical ($\sqrt{\ }$) represents the proportion of increase in axial bolt tension resulting from preload torsion. In these equations, tensile stress due to torsion load application becomes most significant when the thread friction, μ, is high.

Coefficients of Friction for Bolts and Nuts.—Table 1 gives examples of coefficients of friction that are frequently used in determining torque requirements. Dry threads, indicated by the words "None added" in the Lubricant column, are assumed to have some residual machine oil lubrication. Table 1 values are not valid for threads that have been cleaned to remove all traces of lubrication because the coefficient of friction of these threads may be very much higher unless a plating or other film is acting as a lubricant.

Table 1. Coefficients of Friction of Bolts and Nuts

Bolt/Nut Materials	Lubricant	Coefficient of Friction, $\mu \pm 20\%$
Steel[a]	Graphite in petrolatum or oil	0.07
	Molybdenum disulfide grease	0.11
	Machine oil	0.15
Steel,[a] cadmium-plated	None added	0.12
Steel,[a] zinc-plated	None added	0.17
Steel[a]/bronze	None added	0.15
Corrosion-resistant steel or nickel-base alloys/silver-plated materials	None added	0.14
Titanium/steel[a]	Graphite in petrolatum	0.08
Titanium	Molybdenum disulfide grease	0.10

[a] "Steel" includes carbon and low-alloy steels but not corrosion-resistant steels.

Where two materials are separated by a slash (/), either may be the bolt material; the other is the nut material.

Preload Relaxation.—Local yielding, due to excess bearing stress under nuts and bolt heads (caused by high local spots, rough surface finish, and lack of perfect squareness of bolt and nut bearing surfaces), may result in preload relaxation after preloads are first applied to a bolt. Bolt tension also may be unevenly distributed over the threads in a joint, so thread deformation may occur, causing the load to be redistributed more evenly over the threaded length. Preload relaxation occurs over a period of minutes to hours after the application of the preload, so retightening after several minutes to several days may be required. As a general rule, an allowance for loss of preload of about 10 per cent may be made when designing a joint.

Increasing the resilience of a joint will make it more resistant to local yielding, that is, there will be less loss of preload due to yielding. When practical, a joint-length to bolt-diameter ratio of 4 or more is recommended (e.g., a $\frac{1}{4}$-inch bolt and a 1-inch or greater joint length). Through bolts, far-side tapped holes, spacers, and washers can be used in the joint design to improve the joint-length to bolt-diameter ratio.

Over an extended period of time, preload may be reduced or completely lost due to vibration; temperature cycling, including changes in ambient temperature; creep; joint load; and other factors. An increase in the initial bolt preload or the use of thread-locking methods that prevent relative motion of the joint may reduce the problem of preload relaxation due to vibration and temperature cycling. Creep is generally a high-temperature effect, although some loss of bolt tension can be expected even at normal temperatures. Harder materials and creep-resistant materials should be considered if creep is a problem or high-temperature service of the joint is expected.

The mechanical properties of fastener materials vary significantly with temperature, and allowance must be made for these changes when ambient temperatures range beyond 30 to 200°F. Mechanical properties that may change include tensile strength, yield strength, and modulus of elasticity. Where bolts and flange materials are generically dissimilar, such as carbon steel and corrosion-resistant steel or steel and brass, differences in thermal expansion that might cause preload to increase or decrease must be taken into consideration.

Methods of Applying and Measuring Preload.—Depending on the tightening method, the accuracy of preload application may vary up to 25 per cent or more. Care must be taken to maintain the calibration of torque and load indicators. Allowance should be made for uncertainties in bolt load to prevent overstressing the bolts or failing to obtain sufficient preload. The method of tensioning should be based on the required accuracy and relative costs.

The most common methods of bolt tension control are indirect because it is usually difficult or impractical to measure the tension produced in each fastener during assembly. Table 2 lists the most frequently used methods of applying bolt preload and the approximate accuracy of each method. For many applications, fastener tension can be satisfactorily controlled within certain limits by applying a known torque to the fastener. Laboratory tests have shown that whereas a satisfactory torque tension relationship can be established for a given set of conditions, a change of any of the variables, such as fastener material, surface finish, and the presence or absence of lubrication, may severely alter the relationship. Because most of the applied torque is absorbed in intermediate friction, a change in the surface roughness of the bearing surfaces or a change in the lubrication will drastically affect the friction and thus the torque tension relationship. Regardless of the method or accuracy of applying the preload, tension will decrease in time if the bolt, nut, or washer seating faces deform under load, if the bolt stretches or creeps under tensile load, or if cyclic loading causes relative motion between joint members.

Table 2. Accuracy of Bolt Preload Application Methods

Method	Accuracy	Method	Accuracy
By feel	±35%	Computer-controlled wrench	
Torque wrench	±25%	below yield (turn-of-nut)	±15%
Turn-of-nut	±15%	yield-point sensing	±8%
Preload indicating washer	±10%	Bolt elongation	±3–5%
Strain gages	±1%	Ultrasonic sensing	±1%

Tightening methods using power drivers are similar in accuracy to equivalent manual methods.

Elongation Measurement.—Bolt elongation is directly proportional to axial stress when the applied stress is within the elastic range of the material. If both ends of a bolt are accessible, a micrometer measurement of bolt length made before and after the application of tension will ensure the required axial stress is applied. The elongation δ in inches (mm) can be determined from the formula $\delta = F_t \times L_B \div E$, given the required axial stress F_t in psi (MPa), the bolt modulus of elasticity E in psi (MPa), and the effective bolt length L_B in inches (mm). L_B, as indicated in Fig. 2, includes the contribution of bolt area and ends (head and nut) and is calculated from:

$$L_B = \left(\frac{d_{ts}}{d}\right)^2 \times \left(L_s + \frac{H_B}{2}\right) + L_J - L_S + \frac{H_N}{2} \tag{6}$$

where d_{ts} is the thread stress diameter, d is the bolt diameter, L_s is the unthreaded length of the bolt shank, L_j is the overall joint length, H_B is the height of the bolt head, and H_N is the height of the nut.

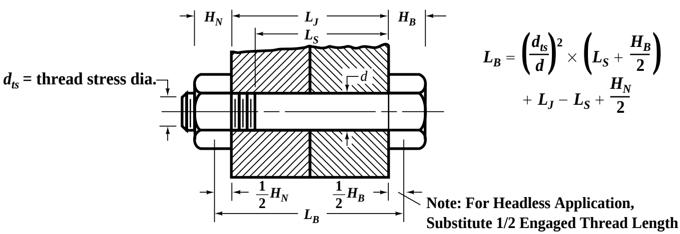


Fig. 2. Effective Length Applicable in Elongation Formulas

The micrometer method is most easily and accurately applied to bolts that are essentially uniform throughout the bolt length, that is, threaded along the entire length or that have only a few threads in the bolt grip area. If the bolt geometry is complex, such as tapered or stepped, the elongation is equal to the sum of the elongations of each section with allowances made for transitional stresses in bolt head height and nut engagement length.

The direct method of measuring elongation is practical only if both ends of a bolt are accessible. Otherwise, if the diameter of the bolt or stud is sufficiently large, an axial hole can be drilled, as shown in Fig. 3, and a micrometer depth gage or other means used to determine the change in length of the hole as the fastener is tightened. A similar method uses a special indicating bolt that has a blind axial hole containing a pin fixed at the bottom. The pin is usually made flush with the bolt head surface before load application. As the bolt is loaded, the elongation causes the end of the pin to move below the reference surface. The displacement of the pin can be converted directly into unit stress by means of a calibrated gage. In some bolts of this type, the pin is set a distance above the bolt so that the pin is flush with the bolt head when the required axial load is reached.

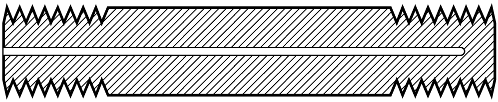


Fig. 3. Hole Drilled to Measure Elongation When One End of Stud or Bolt Is Not Accessible

The *ultrasonic method* of measuring elongation uses a sound pulse, generated at one end of a bolt, that travels the length of a bolt, bounces off the far end, and returns to the sound generator in a measured period of time. The time required for the sound pulse to return depends on the length of the bolt and the speed of sound in the bolt material. The speed of sound in the bolt depends on the material, the temperature, and the stress level. The ultrasonic measurement system can compute the stress, load, or elongation of the bolt at any time by comparing the pulse travel time in the loaded and unstressed conditions. In a similar method, measuring round-trip transit times of longitudinal and shear wave sonic pulses allows calculation of tensile stress in a bolt without consideration of bolt length. This method permits checking bolt tension at any time and does not require a record of the ultrasonic characteristics of each bolt at zero load.

To ensure consistent results, the ultrasonic method requires that both ends of the bolt be finished square to the bolt axis. The accuracy of ultrasonic measurement compares favorably with strain gage methods, but is limited by sonic velocity variations between bolts of the same material and by corrections that must be made for unstressed portions of the bolt heads and threads.

The *turn-of-nut method* applies preload by turning a nut through an angle that corresponds to a given elongation. The elongation of the bolt is related to the angle turned by the formula: $\delta_B = \theta \times l \div 360$, where δ_B is the elongation in inches (mm), θ is the turn angle of the nut in degrees, and l is the lead of the thread helix in inches (mm). Substituting $F_t \times L_B \div E$ for elongation δ_B in this equation gives the turn-of-nut angle required to attain preload F_t:

$$\theta = 360\frac{F_t L_B}{El} \tag{7}$$

where L_B is given by Equation (6), and E is the modulus of elasticity.

Accuracy of the turn-of-nut method is affected by elastic deformation of the threads, by roughness of the bearing surfaces, and by the difficulty of determining the starting point for measuring the angle. The starting point is usually found by tightening the nut enough to seat the contact surfaces firmly, and then loosening it just enough to release any tension and twisting in the bolt. The nut-turn angle will be different for each bolt size, length, material, and thread lead. The preceding method of calculating the nut-turn angle also requires elongation of the bolt without a corresponding compression of the joint material. The turn-of-nut method, as just outlined, is not valid for joints with compressible gaskets or other soft material, or if there is a significant deformation of the nut and joint material relative to that of the bolt. The nut-turn angle would then have to be determined empirically using a simulated joint and a tension-measuring device.

The Japanese Industrial Standards (JIS) Handbook, *Fasteners and Screw Threads,* indicates that the turn-of-nut tightening method is applicable in both elastic and plastic region tightening. Refer to JIS B 1083 for more detail on this subject.

Heating: causes a bolt to expand at a rate proportional to its coefficient of expansion. When a hot bolt and nut are fastened in a joint and cooled, the bolt shrinks and tension is developed. The temperature necessary to develop an axial stress, F_t, (when the stress is below the elastic limit) can be found as follows:

$$T = \frac{F_t}{Ee} + T_o \qquad (8)$$

In this equation, T is the temperature in degrees Fahrenheit needed to develop the axial tensile stress F_t in psi, E is the bolt material modulus of elasticity in psi, e is the coefficient of linear expansion in in./in.-°F, and T_o is the temperature in degrees Fahrenheit to which the bolt will be cooled. $T - T_o$ is, therefore, the temperature change of the bolt. In finite-element simulations, heating and cooling are frequently used to preload mesh elements in tension or compression. Equation (8) can be used to determine required temperature changes in such problems.

Example: A tensile stress of 40,000 psi is required for a steel bolt in a joint operating at 70°F. If E is 30×10^6 psi and e is 6.2×10^{-6} in./in.-°F, determine the temperature of the bolt needed to develop the required stress on cooling.

$$T = \frac{40{,}000}{(30 \times 10^6)(6.2 \times 10^{-6})} + 70 = 285°\text{F}$$

In practice, the bolt is heated slightly above the required temperature (to allow for some cooling while the nut is screwed down) and the nut is tightened snugly. Tension develops as the bolt cools. In another method, the nut is tightened snugly on the bolt, and the bolt is heated in place. When the bolt has elongated sufficiently, as indicated by inserting a thickness gage between the nut and the bearing surface of the joint, the nut is tightened. The bolt develops the required tension as it cools; however, preload may be lost if the joint temperature increases appreciably while the bolt is being heated.

Calculating Thread Tensile-Stress Area.—The tensile-stress area for Unified threads is based on a diameter equivalent to the mean of the pitch and minor diameters. The pitch and the minor diameters for Unified screw threads can be found from the major (nominal) diameter, d, and the screw pitch, $P = 1/n$, where n is the number of threads per inch, by use of the following formulas: the pitch diameter $d_p = d - 0.649519 \times P$; the minor diameter $d_m = d - 1.299038 \times P$. The tensile stress area, A_s, for Unified threads can then be found as follows:

$$A_s = \frac{\pi}{4}\left(\frac{d_m + d_p}{2}\right)^2 \qquad (9)$$

UNJ threads in accordance with MIL-S-8879 have a tensile thread area that is usually considered to be at the basic bolt pitch diameter, so for these threads, $A_s = \pi d^2_p/4$. The tensile stress area for Unified screw threads is smaller than this area, so the required tightening torque for UNJ threaded bolts is greater than for an equally stressed Unified threaded bolt in an equivalent joint. To convert tightening torque for a Unified fastener to the equivalent torque required with a UNJ fastener, use the following relationship:

$$\mathrm{UNJ}_{\mathrm{torque}} = \left(\frac{d \times n - 0.6495}{d \times n - 0.9743}\right)^2 \times \mathrm{Unified}_{\mathrm{torque}} \tag{10}$$

where d is the basic thread major diameter, and n is the number of threads per inch.

The tensile stress area for metric threads is based on a diameter equivalent to the mean of the pitch diameter and a diameter obtained by subtracting ⅙ the height of the fundamental thread triangle from the external-thread minor diameter. The Japanese Industrial Standard JIS B 1082 (see also ISO 898/1) defines the stress area of metric screw threads as follows:

$$A_s = \frac{\pi}{4}\left(\frac{d_2 + d_3}{2}\right)^2 \tag{11}$$

In Equation (11), A_s is the stress area of the metric screw thread in mm²; d_2 is the pitch diameter of the external thread in mm, given by $d_2 = d - 0.649515 \times P$; and d_3 is defined by $d_3 = d_1 - H/6$. Here, d is the nominal bolt diameter; P is the thread pitch; $d_1 = d - 1.082532 \times P$ is the minor diameter of the external thread in mm; and $H = 0.866025 \times P$ is the height of the fundamental thread triangle. Substituting the formulas for d_2 and d_3 into Equation (11) results in $A_s = 0.7854(d - 0.9382P)^2$.

The stress area, A_s, of Unified threads in mm² is given in JIS B 1082 as:

$$A_s = 0.7854\left(d - \frac{0.9743}{n} \times 25.4\right)^2 \tag{12}$$

Relation between Torque and Clamping Force.—The Japanese Industrial Standard JIS B 1803 defines fastener tightening torque T_f as the sum of the bearing surface torque T_w and the shank (threaded) portion torque T_s. The relationship between the applied tightening torque and bolt preload F_{ft} is as follows: $T_f = T_s + T_w = K \times F_f \times d$. In the preceding, d is the nominal diameter of the screw thread, and K is the torque coefficient defined as follows:

$$K = \frac{1}{2d}\left(\frac{P}{\pi} + \mu_s d_2 \sec\alpha' + \mu_w D_w\right) \tag{13}$$

where P is the screw thread pitch; μ_s is the coefficient of friction between threads; d_2 is the pitch diameter of the thread; μ_w is the coefficient of friction between bearing surfaces; D_w is the equivalent diameter of the friction torque bearing surfaces; and α' is the flank angle at the ridge perpendicular section of the thread ridge, defined by $\tan\alpha' = \tan\alpha\cos\beta$, where α is the thread half angle (30°, for example), and β is the thread helix, or lead, angle. β can be found from $\tan\beta = l \div 2\pi r$, where l is the thread lead, and r is the thread radius (i.e., one-half the nominal diameter d). When the bearing surface contact area is circular, D_w can be obtained as follows:

$$D_w = \frac{2}{3} \times \frac{D_o^3 - D_i^3}{D_o^2 - D_i^2} \tag{14}$$

where D_o and D_i are the outside and inside diameters, respectively, of the bearing surface contact area.

The torques attributable to the threaded portion of a fastener, T_s, and bearing surfaces of a joint, T_w, are as follows:

$$T_s = \frac{F_f}{2}\left(\frac{P}{\pi} + \mu_s d_2 \sec\alpha'\right) \tag{15}$$

$$T_w = \frac{F_f}{2}\mu_w D_w \tag{16}$$

where F_f, P, μ, d_2, α', μ_w, and D_w are as previously defined.

Tables 3 and 4 give values of torque coefficient K for coarse- and fine-pitch metric screw threads corresponding to various values of μ_s and μ_w. When a fastener material yields according to the shearing-strain energy theory, the torque corresponding to the yield clamping force (see Fig. 4) is $T_{fy} = K \times F_{fy} \times d$, where the yield clamping force F_{fy} is given by:

$$F_{fy} = \frac{\sigma_y A_s}{\sqrt{1 + 3\left[\frac{2}{d_A}\left(\frac{P}{\pi} + \mu_s d_2 \sec\alpha'\right)\right]^2}} \qquad (17)$$

Table 3. Torque Coefficients *K* for Metric Hexagon Head Bolt and Nut Coarse Screw Threads

Coefficient of Friction										
Between Threads, μ_s	Between Bearing Surfaces, μ_w									
	0.08	0.10	0.12	0.15	0.20	0.25	0.30	0.35	0.40	0.45
0.08	0.117	0.130	0.143	0.163	0.195	0.228	0.261	0.293	0.326	0.359
0.10	0.127	0.140	0.153	0.173	0.206	0.239	0.271	0.304	0.337	0.369
0.12	0.138	0.151	0.164	0.184	0.216	0.249	0.282	0.314	0.347	0.380
0.15	0.153	0.167	0.180	0.199	0.232	0.265	0.297	0.330	0.363	0.396
0.20	0.180	0.193	0.206	0.226	0.258	0.291	0.324	0.356	0.389	0.422
0.25	0.206	0.219	0.232	0.252	0.284	0.317	0.350	0.383	0.415	0.448
0.30	0.232	0.245	0.258	0.278	0.311	0.343	0.376	0.409	0.442	0.474
0.35	0.258	0.271	0.284	0.304	0.337	0.370	0.402	0.435	0.468	0.500
0.40	0.285	0.298	0.311	0.330	0.363	0.396	0.428	0.461	0.494	0.527
0.45	0.311	0.324	0.337	0.357	0.389	0.422	0.455	0.487	0.520	0.553

Values in the table are average values of torque coefficient calculated using: Equations (13) and (14) for K and D_w; diameters d of 4, 5, 6, 8, 10, 12, 16, 20, 24, 30, and 36 mm; and selected corresponding pitches P and pitch diameters d_2 according to JIS B 0205 (ISO 724) thread standard. Dimension D_i was obtained for a Class 2 fit without chamfer from JIS B 1001, Diameters of Clearance Holes and Counterbores for Bolts and Screws (equivalent to ISO 273-1979). The value of D_o was obtained by multiplying the reference dimension from JIS B 1002, width across the flats of the hexagon head, by 0.95.

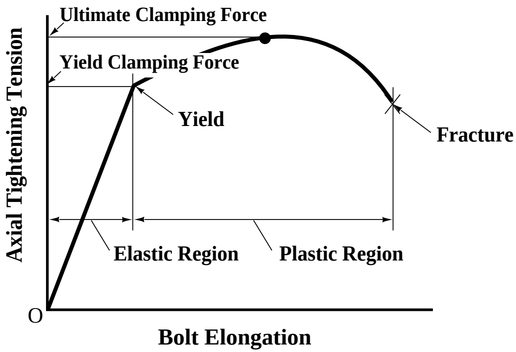


Fig. 4. The Relationship between Bolt Elongation and Axial Tightening Tension

Table 4. Torque Coefficients *K* for Metric Hexagon Head Bolt and Nut Fine-Screw Threads

Coefficient of Friction										
Between Threads, μ_s	Between Bearing Surfaces, μ_w									
	0.08	0.10	0.12	0.15	0.20	0.25	0.30	0.35	0.40	0.45
0.08	0.106	0.118	0.130	0.148	0.177	0.207	0.237	0.267	0.296	0.326
0.10	0.117	0.129	0.141	0.158	0.188	0.218	0.248	0.278	0.307	0.337
0.12	0.128	0.140	0.151	0.169	0.199	0.229	0.259	0.288	0.318	0.348
0.15	0.144	0.156	0.168	0.186	0.215	0.245	0.275	0.305	0.334	0.364
0.20	0.171	0.183	0.195	0.213	0.242	0.272	0.302	0.332	0.361	0.391
0.25	0.198	0.210	0.222	0.240	0.270	0.299	0.329	0.359	0.389	0.418
0.30	0.225	0.237	0.249	0.267	0.297	0.326	0.356	0.386	0.416	0.445
0.35	0.252	0.264	0.276	0.294	0.324	0.353	0.383	0.413	0.443	0.472
0.40	0.279	0.291	0.303	0.321	0.351	0.381	0.410	0.440	0.470	0.500
0.45	0.306	0.318	0.330	0.348	0.378	0.408	0.437	0.467	0.497	0.527

Values in the table are average values of torque coefficient calculated using Equations (13) and (14) for K and D_w; diameters d of 8, 10, 12, 16, 20, 24, 30, and 36 mm; and selected respective pitches P and pitch diameters d_2 according to JIS B 0207 thread standard (ISO 724). Dimension D_i was obtained for a Class 1 fit without chamfer from JIS B 1001, Diameters of Clearance Holes and Counterbores for Bolts and Screws (equivalent to ISO 273-1979). The value of D_o was obtained by multiplying the reference dimension from JIS B 1002 (small type series), width across the flats of the hexagon head, by 0.95.

In Equation (17), σ_y is the yield point or proof stress of the bolt, A_s is the stress area of the thread, and $d_A = (4A_s/\pi)^{1/2}$ is the diameter of a circle having an area equal to the stress area of the thread. The other variables have been identified previously.

Example: Find the torque required to tighten a 10-mm coarse-threaded (P = 1.5) grade 8.8 bolt to yield assuming that both the thread- and bearing-friction coefficients are 0.12.

Solution: From Equation (17), calculate F_{fy} and then solve $T_{fy} = KF_{fy}d$ to obtain the torque required to stress the bolt to the yield point.

$$\begin{aligned}
\sigma_y &= 800\ \text{N/mm}^2\ \text{(MPa) (minimum, based on 8.8 grade rating)}\\
A_s &= 0.7854(10 - 0.9382 \times 1.5)^2 = 57.99\ \text{mm}^2\\
d_A &= (4A_s/\pi)^{1/2} = 8.6\ \text{mm}\\
d_2 &= 9.026\ \text{mm (see JIS B 0205 or ISO 724)}
\end{aligned}$$

Find α' from $\tan\alpha' = \tan\alpha\cos\beta$ using:

$$\begin{aligned}
&\alpha = 30°;\ \tan\beta = l \div 2\pi r;\ l = P = 1.5;\ \text{and}\ r = d \div 2 = 5\ \text{mm}\\
&\tan\beta = 1.5 \div 10\pi = 0.048,\ \text{therefore}\ \beta = 2.73°\\
&\tan\alpha' = \tan\alpha\cos\beta = \tan 30° \times \cos 2.73° = 0.577,\ \text{and}\ \alpha' = 29.97°
\end{aligned}$$

Solving Equation (17) gives the yield clamping force as follows:

$$F_{fy} = \frac{800 \times 58.0}{\sqrt{1 + 3\left[\frac{2}{8.6}\left(\frac{1.5}{\pi} + 0.12 \times 9.026 \times \sec 29.97°\right)\right]^2}} = 38{,}075\ \text{N}$$

K can be determined from Tables 3 (coarse thread) and Tables 4 (fine thread) or from Equations (13) and (14). From Table 3, for μ_s and μ_w equal to 0.12, K = 0.164. The yield-point tightening torque can then be found from $T_{fy} = K \times F_{fy} \times d = 0.164 \times 38{,}075 \times 10 = 62.4 \times 10^3$ N-mm = 62.4 N-m.

Obtaining Torque and Friction Coefficients.—Given suitable test equipment, the torque coefficient K and friction coefficients between threads μ_s or between bearing surfaces μ_w can be determined experimentally as follows: Measure the value of the axial tightening tension and the corresponding tightening torque at an arbitrary point in the 50 to 80 per cent range of the bolt yield point or proof stress (for steel bolts, use the minimum value of the yield point or proof stress multiplied by the stress area of the bolt). Repeat this test several times and average the results. The tightening torque may be considered as the sum of the torque on the threads plus the torque on the bolt head- or nut-to-joint bearing surface. The torque coefficient can be found from $K = T_f \div F_f \times d$, where F_f is the measured axial tension, and T_f is the measured tightening torque.

To measure the coefficient of friction between threads or bearing surfaces, obtain the total tightening torque and that portion of the torque due to the thread or bearing surface friction. If only tightening torque and the torque on the bearing surfaces can be measured, then the difference between these two measurements can be taken as the thread-tightening torque. Likewise, if only the tightening torque and threaded-portion torque are known, the torque due to bearing can be taken as the difference between the known torques. The coefficients of friction between threads and bearing surfaces, respectively, can be obtained from the following:

$$\mu_s = \frac{2T_s \cos\alpha'}{d_2 F_f} - \cos\alpha' \tan\beta \qquad (18) \qquad\qquad \mu_w = \frac{2T_w}{D_w F_f} \qquad (19)$$

As before, T_s is the torque attributable to the threaded portion of the screw, T_w is the torque due to bearing, D_w is the equivalent diameter of friction torque on bearing surfaces according to Equation (14), and F_f is the measured axial tension.

Torque-Tension Relationships.—Torque is usually applied to develop an axial load in a bolt. To achieve the desired axial load in a bolt, the torque must overcome friction in the threads and friction under the nut or bolt head. In Fig. 5, the axial load P_B is a component of the normal force developed between threads. The normal-force component perpendicular to the thread helix is $P_{N\beta}$ and the other component of this force is the torque load $P_B \tan\beta$ that is applied in tightening the fastener. Assuming the turning force is applied at the pitch diameter of the thread, the torque T_1 needed to develop the axial load is $T_1 = P_B \times \tan\beta \times d_2/2$. Substituting $\tan\beta = l \div \pi d_2$ into the previous expression gives $T_1 = P_B \times l \div 2\pi$.

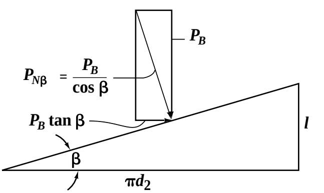


Fig. 5. Free Body Diagram of Thread Helix Forces

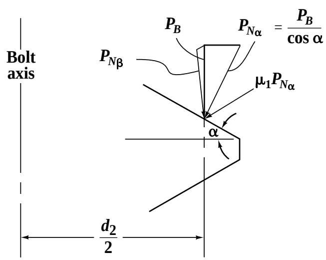


Fig. 6. Thread Friction Force

In Fig. 6, the normal-force component perpendicular to the thread flanks is $P_{N\alpha}$. With a coefficient of friction μ_1 between the threads, the friction load is equal to $\mu_1 P_{N\alpha}$, or $\mu_1 P_B \div \cos\alpha$. Assuming the force is applied at the pitch diameter of the thread, the torque T_2 to overcome thread friction is given by:

$$T_2 = \frac{d_2 \mu_1 P_B}{2\cos\alpha} \qquad (20)$$

With the coefficient of friction μ_2 between a nut or bolt-head pressure face and a component face, as in Fig. 7, the friction load is equal to $\mu_2 P_B$. Assuming the force is applied midway between the nominal (bolt) diameter d and the pressure-face diameter b, the torque T_3 to overcome the nut or bolt underhead friction is:

$$T_3 = \frac{d+b}{4}\mu_2 P_B \qquad (21)$$

The total torque, T, required to develop axial bolt load, P_B, is equal to the sum of the torques T_1, T_2, and T_3 as follows:

$$T = P_B\left(\frac{l}{2\pi} + \frac{d_2\mu_1}{2\cos\alpha} + \frac{(d+b)\mu_2}{4}\right) \qquad (22)$$

For a fastener system with 60° threads, $\alpha = 30°$ and d_2 is approximately $0.92d$. If no loose washer is used under the rotated nut or bolt head, b is approximately $1.5d$ and Equation (22) reduces to:

$$T = P_B[0.159 \times l + d(0.531\mu_1 + 0.625\mu_2)] \qquad (23)$$

In addition to the conditions of Equation (23), if the thread and bearing friction coefficients, μ_1 and μ_2, are equal (which is not necessarily so), then $\mu_1 = \mu_2 = \mu$, and the previous equation reduces to:

$$T = P_B(0.159l + 1.156\mu d) \qquad (24)$$

Example: Estimate the torque required to tighten a UNC ½-13 grade 8 steel bolt to a preload equivalent to 55 per cent of the minimum tensile bolt strength. Assume that the bolt is unplated and both the thread and bearing friction coefficients equal 0.15.

Solution: The minimum tensile strength for SAE grade 8 bolt material is 150,000 psi (from page 1488). To use Equation (24), find the stress area of the bolt using Equation (9) with $P = 1/13$, $d_m = d - 1.2990P$, and $d_p = d - 0.6495P$, and then calculate the necessary preload, P_B, and the applied torque, T.

$$A_s = \frac{\pi}{4}\left(\frac{0.4500 + 0.4001}{2}\right)^2 = 0.1419 \text{ in.}^2$$

$$P_B = \sigma_{allow} \times A_s = 0.55 \times 150{,}000 \times 0.1419 = 11{,}707 \text{ lb}_f$$

$$T = 11{,}707\left(\frac{0.159}{13} + 1.156 \times 0.15 \times 0.500\right) = 1158 \text{ lb-in.} = 96.5 \text{ lb-ft}$$

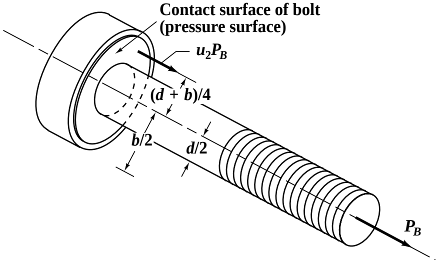
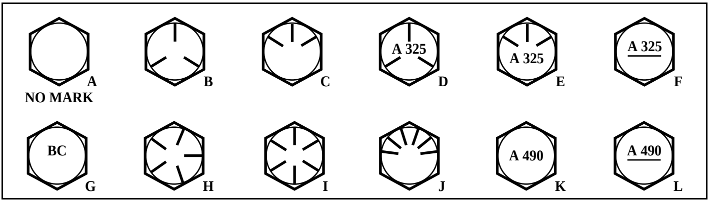


Fig. 7. Nut or Bolt Head Frictin Force

Grade Marks and Material Properties for Bolts and Screws.—Bolts, screws, and other fasteners are marked on the head with a symbol that identifies the grade of the fastener. The grade specification establishes the minimum mechanical properties that the fastener must meet. Additionally, industrial fasteners must be stamped with a registered head mark that identifies the manufacturer. The grade identification table identifies the grade markings and gives mechanical properties for some commonly used ASTM and SAE steel fasteners. Metric fasteners are identified by property grade marks, which are specified in ISO and SAE standards. These marks are discussed with metric fasteners.

Grade Identification Marks and Mechanical Properties of Bolts and Screws



Identifier	Grade	Size (in.)	Min. Strength (10^3 psi)			Material & Treatment
			Proof	Tensile	Yield	
A	SAE Grade 1	¼ to 1½	33	60	36	1
	ASTM A307	¼ to 1½	33	60	36	3
	SAE Grade 2	¼ to ¾	55	74	57	1
		⅞ to 1½	33	60	36	
	SAE Grade 4	¼ to 1½	65	115	100	2, a
B	SAE Grade 5	¼ to 1	85	120	92	2,b
	ASTM A449	1⅛ to 1½	74	105	81	
	ASTM A449	1¾ to 3	55	90	58	
C	SAE Grade 5.2	¼ to 1	85	120	92	4, b
D	ASTM A325, Type 1	½ to 1	85	120	92	2,b
		1⅛ to 1½	74	105	81	
E	ASTM A325, Type 2	½ to 1	85	120	92	4, b
		1⅛ to 1½	74	105	81	
F	ASTM A325, Type 3	½ to 1	85	120	92	5, b
		1⅛ to 1½	74	105	81	
G	ASTM A354, Grade BC	¼ to 2½	105	125	109	5,b
		2¾ to 4	95	115	99	
H	SAE Grade 7	¼ to 1½	105	133	115	7, b
I	SAE Grade 8	¼ to 1½	120	150	130	7, b
	ASTM A354, Grade BD	¼ to 1½	120	150	130	6, b
J	SAE Grade 8.2	¼ to 1	120	150	130	4, b
K	ASTM A490, Type 1	½ to 1½	120	150	130	6, b
L	ASTM A490, Type 3					5, b

Material Steel: 1—low or medium carbon; 2—medium carbon; 3—low carbon; 4—low-carbon martensite; 5—weathering steel; 6—alloy steel; 7— medium-carbon alloy. Treatment: a—cold drawn; b—quench and temper.

Detecting Counterfeit Fasteners.—Fasteners that have markings identifying them as belonging to a specific grade or property class are counterfeit if they do not meet the standards established for that class. Counterfeit fasteners may break unexpectedly at smaller loads than expected. Generally, these fasteners are made from the wrong material or they are not properly strengthened during manufacture. Either way, counterfeit fasteners can lead to dangerous failures in assemblies. The law now requires testing of fasteners used in some critical applications. Detection of counterfeit fasteners is difficult because the counterfeits look genuine. The only sure way to determine if a fastener meets its specification is to test it. However, reputable distributors will assist in verifying the authenticity of the fasteners they sell. For important applications, fasteners can be checked to determine whether they perform according to the standard. Typical laboratory checks used to detect fakes include testing hardness, elongation, and ultimate loading, and a variety of chemical tests.

Mechanical Properties and Grade Markings of Nuts.—Three grades of hex and square nuts designated Grades 2, 5, and 8 are specified by the SAE J995 standard covering nuts in the $\frac{1}{4}$- to $1\frac{1}{2}$- inch diameter range. Grades 2, 5, and 8 nuts roughly correspond to the SAE specified bolts of the same grade. Additional specifications are given for miscellaneous nuts such as hex jam nuts, hex slotted nuts, heavy hex nuts, etc. Generally speaking, use nuts of a grade equal to or greater than the grade of the bolt being used. Grade 2 nuts are not required to be marked, however, all Grades 5 and 8 nuts in the $\frac{1}{4}$- to $1\frac{1}{2}$-inch range must be marked in one of three ways: Grade 5 nuts may be marked with a dot on the face of the nut and a radial or circumferential mark at 120° counterclockwise from the dot; or a dot at one corner of the nut and a radial line at 120° clockwise from the nut, or one notch at each of the six corners of the nut. Grade 8 nuts may be identified by a dot on the face of the nut with a radial or circumferential mark at 60° counterclockwise from the dot; or a dot at one corner of the nut and a radial line at 60° clockwise from the nut, or two notches at each of the six corners of the nut.

Working Strength of Bolts.—When the nut on a bolt is tightened, an initial tensile load is placed on the bolt that must be taken into account in determining its safe working strength or external load-carrying capacity. The total load on the bolt theoretically varies from a maximum equal to the sum of the initial and external loads (when the bolt is absolutely rigid and the parts held together are elastic) to a minimum equal to either the initial or external loads, whichever is the greater (where the bolt is elastic and the parts held together are absolutely rigid). No material is absolutely rigid, so in practice the total load values fall somewhere between these maximum and minimum limits, depending upon the relative elasticity of the bolt and joint members.

Some experiments made at Cornell University to determine the initial stress due to tightening nuts on bolts sufficiently to make a packed joint steam-tight showed that experienced mechanics tighten nuts with a pull roughly proportional to the bolt diameter. It was also found that the stress due to nut tightening was often sufficient to break a $\frac{1}{2}$-inch (12.7-mm) bolt, but not larger sizes, assuming that the nut is tightened by an experienced mechanic. It may be concluded, therefore, that bolts smaller than $\frac{5}{8}$ inch (15.9 mm) should not be used for holding cylinder heads or other parts requiring a tight joint. As a result of these tests, the following empirical formula was established for the working strength of bolts used for packed joints or joints where the elasticity of a gasket is greater than the elasticity of the studs or bolts.

$$W = S_t(0.55d^2 - 0.25d)$$

In this formula, W = working strength of bolt or permissible load, in pounds, after allowance is made for initial load due to tightening; S_t = allowable working stress in tension, pounds per square inch; and d = nominal outside diameter of stud or bolt, inches. A somewhat more convenient formula, and one that gives approximately the same results, is

$$W = S_t(A - 0.25d)$$

In this formula, W, S_t, and d are as previously given, and A = area at the root of the thread, square inches.

Example: What is the working strength of a 1-inch bolt that is screwed tightly in a packed joint when the allowable working stress is 10,000 psi?

$$W = 10{,}000(0.55 \times 1 - 0.25 \times 1) = 3000 \text{ pounds approx.}$$

Formulas for Stress Areas and Lengths of Engagement of Screw Threads.— The critical areas of stress of mating screw threads are: 1) The effective cross-sectional area, or tensile-stress area, of the external thread; 2) the shear area of the external thread, which depends principally on the minor diameter of the tapped hole; and 3) the shear area of the internal thread, which depends principally on the major diameter of the external thread. The relation of these three stress areas to each other is an important factor in determining how a threaded connection will fail, whether by breakage in the threaded section of the screw (or bolt) or by stripping of either the external or internal thread.

If failure of a threaded assembly should occur, it is preferable for the screw to break rather than have either the external or internal thread strip. In other words, the length of engagement of mating threads should be sufficient to carry the full load necessary to break the screw without the threads stripping.

If mating internal and external threads are manufactured of materials having equal tensile strengths, then to prevent stripping of the external thread, the length of engagement should be not less than that given by Formula (1):

$$L_e = \frac{2 \times A_t}{3.1416 K_n \max[\tfrac{1}{2} + 0.57735n(E_s \min - K_n \max)]} \tag{1}$$

In this formula, the factor of 2 means that it is assumed that the area of the screw in shear must be twice the tensile-stress area to attain the full strength of the screw (this value is slightly larger than required and thus provides a small factor of safety against stripping); L_e = length of engagement, in inches; n = number of threads per inch; K_n max = maximum minor diameter of internal thread; E_s min = minimum pitch diameter of external thread for the class of thread specified; and A_t = tensile-stress area of screw thread given by Formula (2a) or (2b) or the thread tables for Unified threads, Tables 4a through 4k starting on page page 1740, which are based on Formula (2a).

For steels of up to 100,000 psi ultimate tensile strength,

$$A_t = 0.7854\left(D - \frac{0.9743}{n}\right)^2 \tag{2a}$$

For steels of over 100,000 psi ultimate tensile strength,

$$A_t = 3.1416\left(\frac{E_s \min}{2} - \frac{0.16238}{n}\right)^2 \tag{2b}$$

In these formulas, D = basic major diameter of the thread and the other symbols have the same meanings as before.

Stripping of Internal Thread: If the internal thread is made of material of lower strength than the external thread, stripping of the internal thread may take place before the screw breaks. To determine whether this condition exists, it is necessary to calculate the factor J for the relative strength of the external and internal threads given by Formula (3):

$$J = \frac{A_s \times \text{tensile strength of external thread material}}{A_n \times \text{tensile strength of internal thread material}} \tag{3}$$

If J is less than or equal to 1, the length of engagement determined by Formula (1) is adequate to prevent stripping of the internal thread; if J is greater than 1, the required length of engagement Q to prevent stripping of the internal thread is obtained by multiplying the length of engagement L_e, Formula (1), by J:

$$Q = JL_e \tag{4}$$

In Formula (3), A_s and A_n are the shear areas of the external and internal threads, respectively, given by Formulas (5) and (6):

$$A_s = 3.1416nL_eK_n\text{max}\left[\frac{1}{2n} + 0.57735(E_s\text{min} - K_n\text{max})\right] \tag{5}$$

$$A_n = 3.1416nL_eD_s\text{min}\left[\frac{1}{2n} + 0.57735(D_s\text{min} - E_n\text{max})\right] \tag{6}$$

In these formulas, n = threads per inch; L_e = length of engagement from Formula (1); K_n max = maximum minor diameter of internal thread; E_s min = minimum pitch diameter of the external thread for the class of thread specified; D_s min = minimum major diameter of the external thread; and E_n max = maximum pitch diameter of internal thread.

Load to Break Threaded Portion of Screws and Bolts.—The direct tensile load P to break the threaded portion of a screw or bolt (assuming that no shearing or torsional stresses are acting) can be determined from the following formula:

$$P = SA_t$$

where P = load in pounds to break screw; S = ultimate tensile strength of material of screw or bolt in pounds per square inch; and A_t = tensile-stress area in square inches from Formula (2a), (2b), or from the screw thread tables.

Lock Wire Procedure Detail.—Wire ties are frequently used as a locking device for bolted connections to prevent loosening due to vibration and loading conditions, or tampering. The use of safety wire ties is illustrated in Figs. 1 and 2 below. The illustrations assume the use of right-hand threaded fasteners and the following additional rules apply:

1) No more that three (3) bolts may be tied together; 2) Bolt heads may be tied as shown only when the female thread receiver is captive; 3) Pre-drilled nuts may be tied in a fashion similar to that illustrated with the following conditions. a) Nuts must be heat-treated;

and b) Nuts are factory drilled for use with lock wire.

4) Lock wire must fill a minimum of 75% of the drilled hole provided for the use of lock wire; and 5) Lock wire must be aircraft quality stainless steel of 0.508 mm (0.020 inch) diameter, 0.8128 mm (0.032 inch) diameter, or 1.067 mm 0.042 inch) diameter. Diameter of lock wire is determined by the thread size of the fastener to be safe-tied. a) Thread sizes of 6 mm (0.25 inch) and smaller use 0.508mm (0.020 inch) wire; b) Thread sizes of 6 mm (0.25 inch) to 12 mm (0.5 inch) use 0.8128 mm (0.032 inch) wire; c) Thread sizes > 12 mm (0.5 inch) use 1.067 mm (0.042 inch) wire; and d) The larger wire may be used in smaller bolts in cases of convenience, but smaller wire must not be used in larger fastener sizes.

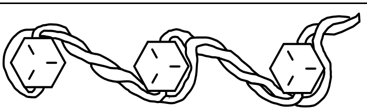


Fig. 1. Three (3) Bolt Procedure

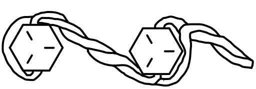


Fig. 2. Two (2) Bolt Procedure