Unbrako

Engineering Guide

Socket Products

Unbrako

SOCKET HEAD CAP SCREWS. . . Why Socket Screws? Why UNBRAKO?

The most important reasons for the increasing use of socket head cap screws in industry are safety, reliability and economy. All three reasons are directly traceable to the superior performance of socket screws vs. other fasteners, and that is due to their superior strength and advanced design.

- Reliability, higher pressures, stresses and speeds in todays machines and equipment demand stronger, more reliable joints and stronger, more reliable fasteners to hold them together.
- Rising costs make failure and downtime intolerable. Bigger, more complex units break down more frequently despite every effort to prevent it.
- This is why the reliability of every component has become critical. Components must stay together to function properly, and to keep them together joints must stay tight.
- Joint reliability and safety with maximum strength and fatigue resistance. UNBRAKO socket cap screws offer this to a greater degree than any other threaded fastener you can purchase "off-the-self."
- UNBRAKO socket cap screws offer resistance to a greater degree than any other threaded fasteners you can purchase "off-the-shelf."

TENSILE STRENGTH

- U.S. standard alloy steel socket head cap screws are made to strength levels of 180,000 and 170,000 psi to current industry standards. However, UNBRAKO socket cap screws are consistently maintained at 190,000 and 180,000 psi (depending on screw diameter).
- The higher tensile strength of UNBRAKO socket screws can be translated into savings. Using fewer socket screws of the same size can achieve the same clamping force in the joint. A joint requiring twelve 1-3/8" Grade 5 hex heads would need only 7 UNBRAKO socket head cap screws. Use them size for size and there are fewer holes to drill and tap and fewer screws to buy and handle. Smaller diameter socket head cap screws vs. larger hex screws cost less to drill and tap, take less energy to drive, and there is also weight saving.
- The size of the component parts can be reduced since the cylindrical heads of socket screws need less space than hex heads and require no additional wrench space.

FATIGUE STRENGTH

- Joints that are subject to external stress loading are susceptible to fatigue failure. UNBRAKO socket screws have distinct advantages that give you an extra bonus of protection against this hazard.
- Three major factors account for the greater fatigue resistance of UNBRAKO socket screws – design improvements, mechanical properties and closely controlled manufacturing processes.

AUSTENITIC STAINLESS STEEL STANDARD SERIES

UNBRAKO stainless socket screws are made from austenitic stainless steel. UNBRAKO stainless screws offer excellent resistance to rust and corrosion from acids, organic substances, salt solutions and atmospheres. Superior properties attained with stainless steel include retention of a high percentage of tensile strength and good creep resistance up to 800°F. without scaling or oxidation, and good shock and impact resistance to temperatures as low as –300°F.

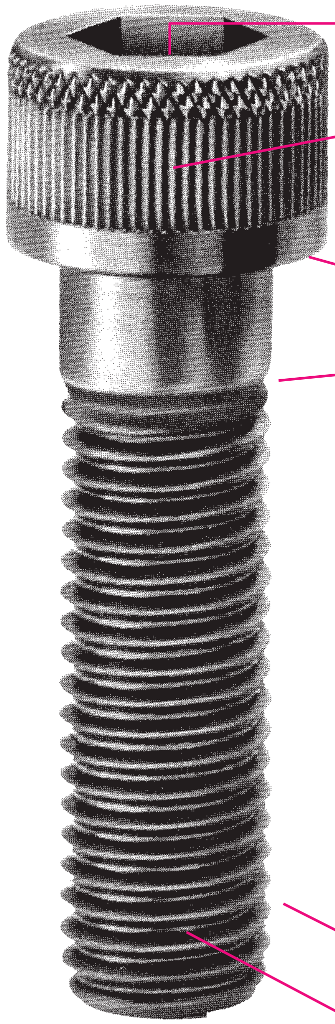
non-magnetic – Valuable in certain electrical applications. Maximum permeability is 1.2 Can be reduced to 1.02 by bright annealing.

cleanliness – Corrosion resistant characteristics of UNBRAKO screws are useful in chemical, food processing, appliance, paper, textile, packaging and pharmaceutical industries, as well as laboratories, hospitals, etc.

eye-appeal – Bright, non-tarnishing qualities add to appearance and salability of many products; are valuable assets to designers.

Standard processing of UNBRAKO stainless steel socket screws includes a passivation surface treatment which removes any surface contaminations.

PROFILE OF EXTRA STRENGTH

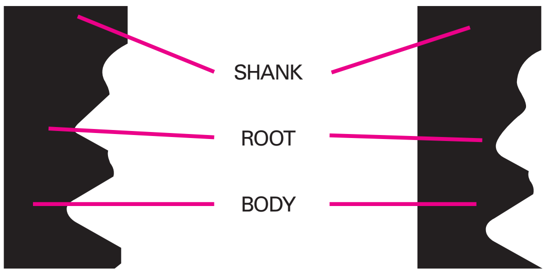


Deep, accurate socket for high torque wrenching. Knurls for easier handling. Marked for easier identification.

Head with increased bearing area for greater loading carrying capacity. Precision forged for symmetrical grain flow, maximum strength.

Elliptical fillet doubles fatigue life at critical head-shank juncture.

"3-R" (radiused-root runout) increases fatigue life in this critical head-shank juncture.

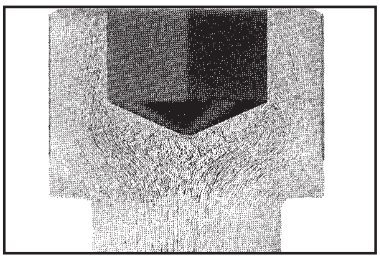


CONVENTIONAL THREAD RUNOUT – Note sharp angle at root where high stress concentration soon develops crack which penetrates into body of the screw.

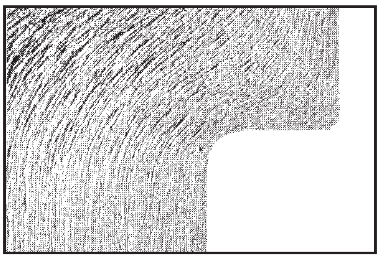
UNBRAKO "3-R" (RADIUSED ROOT RUNOUT) THREAD – Controlled radius of runout root provides a smooth form that distributes stress and increases fatigue life of thread run-out as a much as 300% in certain sizes.

Fully formed radiused thread increases fatigue life 100% over flat root thread forms.

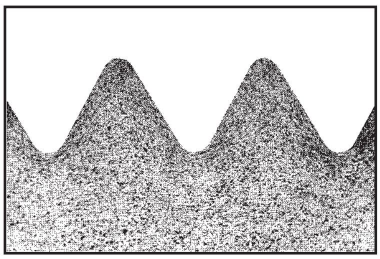
Controlled heat treatment produces maximum strength without brittleness.



Accurate control of socket depth gives more wrench engagement than other screws, permits full tightening without cracking or reaming the socket, yet provides ample metal in the crucial fillet area for maximum head strength.



Controlled head forging, uniform grain flow, unbroken flow lines; makes heads stronger; minimizes failure in vital fillet area; adds to fatigue strength.



Contour-following flow lines provide extra shear strength in threads, resist stripping and provide high fatigue resistance. The large root radius UNBRAKO socket screw development doubles fatigue life compared to flat root thread forms.

Unbrako

UNBRAKO Metric Fasteners

UNBRAKO Metric Fasteners are the strongest off-the-shelf threaded fasteners you can buy. Their exclusive design features and closely controlled manufacturing processes insure the dimensional accuracy, strength and fatigue resistance needed for reliability in today's advanced technology. They are manufactured with the same methods and features as their inch-series counterpart.

Strength

UNBRAKO metric socket head cap screws are made into property class 12.9 with a minimum ultimate tensile strength of 1300 or 1250 MPa depending on screw diameter. Precision in manufacturing and careful control in stress areas insure strength in such critical areas as heads, sockets, threads, fillets, and bearing areas.

When you purchase UNBRAKO metric socket screw products, you can be sure that they meet or exceed the strength levels of all current standards, including the three most common-ANSI, ISO and DIN. Unbrako is represented on several ASME, ANSI, ASTM and ISO committees.

- ANSI (American National Standards Institute) documents are published by ASME (The American Society of Mechanical Engineers) and are familiar to almost all users of socket screw products in the U.S.A.
- ASTM (American Society for Testing and Materials). Many ANSI documents list dimensional information but refer to ASTM specifications for materials, mechanical properties, and test criteria.
- ISO (International Standards Organization) is a standards group comprising 70 member nations. Its objective is to provide standards that will be completely universal and common to all countries subscribing.
- DIN (Deutsche Industries Normen) is the German standards group.

NOTE: The proper tightening of threaded fasteners can have a significant effect on their performance.

A WARNING TO METRIC FASTENER USERS

Metric socket cap screws are NOT sold in a single strength level like U.S. inch socket screws.

Property Class	General Material	Strength Level, UTS min. MPa (KSI)
	International Standards Organization, ISO	
Property Class 8.8	Carbon Steel	800 (116) < M16 830 (120) ≥ M16
Property Class 10.9	Alloy Steel	1040 (151)
Property Class 12.9	Alloy Steel	1220 (177)
USA Standards ASTM A574M	Alloy Steel	1220 (177)
Unbrako Standards ASTM A574M	Alloy Steel	1300 (189) ≤ M16 1250 (181) > M16

STANDARDS

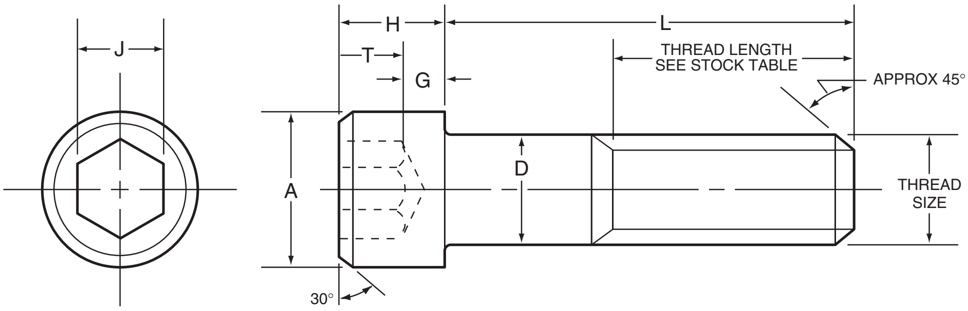
The use of metric fasteners in the worldwide market has led to the creation of many standards. These standards specify the fastener requirements: dimensions, material, strength levels, inspection, etc. Different standards are the responsibility of various organizations and are not always identical. Unbrako supplies metric fasteners for maximum interchangeability with all standards. This Engineering Guide was published with the most current values, which are however subject to change by any standards organization at any time.

METRIC SOCKET HEAD CAP SCREWS

Dimensions

Threads: ANSI B1.13M, ISO 261, ISO 262 (coarse series only)

Property Class: 12.9-ISO 898/1



NOTES

1. **Material:** ASTM A574M, DIN ENISO4762-alloy steel
2. **Hardness:** Rc 38-43
3. **Tensile Stress:** 1300 MPa thru M16 size. 1250 MPa over M16 size.
4. **Yield Stress:** 1170 MPa thru M16 size. 1125 MPa over M16 size.
5. **Thread Class:** 4g 6g

LENGTH TOLERANCE

nominal screw length	nominal screw diameter		
	M1.6 thru M10	M12 thru M20	over 20
	tolerance on lgth., mm		
Up to 16 mm, incl.	±0.3	±0.3	–
Over 16 to 50 mm, incl.	±0.4	±0.4	±0.7
Over 50 to 120 mm, incl.	±0.7	±1.0	±1.5
Over 120 to 200 mm, incl.	±1.0	±1.5	±2.0
Over 200 mm	±2.0	±2.5	±3.0

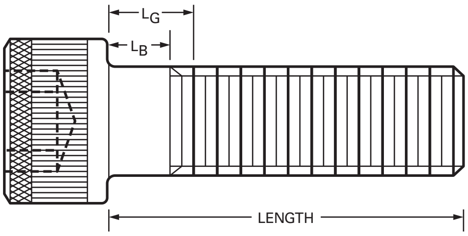
DIMENSIONS | MECHANICAL PROPERTIES | APPLICATION DATA

thread size nom.	pitch	A max.	D max.	H max.	J nom.	G min.	T min.	UTS min. MPa	tensile strength min.		single shear strength of body min.		recommended ** seating torque plain finish	
									kN	lbs.	kN	lbs.	N-m	in-lbs.
M1.6	0.35	3.0	1.6	1.6	1.5	0.54	0.80	1300	1.65	370	1.57	352.5	0.29	2.6
M2	0.40	3.8	2.0	2.0	1.5	0.68	1.0	1300	2.69	605	2.45	550	0.60	5.3
M2.5	0.45	4.5	2.5	2.5	2.0	0.85	1.25	1300	4.41	990	3.83	860	1.21	11
M3	0.5	5.5	3.0	3.0	2.5	1.02	1.5	1300	6.54	1,470	5.5	1240	2.1	19
M4	0.7	7.0	4.0	4.0	3.0	1.52	2.0	1300	11.4	2,560	9.8	2,205	4.6	41
M5	0.8	8.5	5.0	5.0	4.0	1.90	2.5	1300	18.5	4,160	15.3	3,445	9.5	85
M6	1.0	10.0	6.0	6.0	5.0	2.28	3.0	1300	26.1	5,870	22.05	4,960	16	140
M8	1.25	13.0	8.0	8.0	6.0	3.2	4.0	1300	47.6	10,700	39.2	8,800	39	350
M10	1.5	16.0	10.0	10.0	8.0	4.0	5.0	1300	75.4	17,000	61	13,750	77	680
M12	1.75	18.0	12.0	12.0	10.0	4.8	6.0	1300	110	24,700	88	19,850	135	1,200
*(M14)	2.0	21.0	14.0	14.0	12.0	5.6	7.0	1300	150	33,700	120	27,000	215	1,900
M16	2.0	24.0	16.0	16.0	14.0	6.4	8.0	1300	204	45,900	157	35,250	330	2,900
M20	2.5	30.0	20.0	20.0	17.0	8.0	10.0	1250	306	68,800	235.5	53,000	650	5,750
M24	3.0	36.0	24.0	24.0	19.0	9.6	12.0	1250	441	99,100	339	76,500	1100	9,700
*M30	3.5	45.0	30.0	30.0	22.0	12.0	15.0	1250	701	158,000	530	119,000	2250	19,900
*M36	4.0	54.0	36.0	36.0	27.0	14.4	18.0	1250	1020	229,000	756	171,500	3850	34,100
*M42	4.5	63.0	42.0	42.0	32.0	16.8	21.0	1250	1400	315,000	1040	233,500	6270	55,580
*M48	5.0	72.0	48.0	48.0	36.0	19.2	24.0	1250	1840	413,000	1355	305,000	8560	75,800

All dimensions in millimeters.
Sizes in brackets not preferred for new designs.
*Non-stock diameter.
**Torque calculated in accordance with VDI 2230, "Systematic Calculation of High Duty Bolted Joints," to induce approximately 800 MPa stress in screw threads. Torque values listed are for plain screws. (See Note, page 1.)

Unbrako

SOCKET HEAD CAP SCREWS ■ Metric ■ Body and Grip Lengths



BODY and GRIP LENGTHS

L_G is the maximum grip length and is the distance from the bearing surface to the first complete thread.

L_B is the minimum body length and is the length of the unthreaded cylindrical portion of the shank.

BODY AND GRIP LENGTH DIMENSIONS FOR METRIC SOCKET HEAD CAP SCREWS

Nominal Size	M1.6		M2		M2.5		M3		M4		M5		M6		M8		M10		M12		M14		M16		M20		M24	
Nominal Length	L_G	L_B	L_G	L_B	L_G	L_B	L_G	L_B	L_G	L_B	L_G	L_B	L_G	L_B	L_G	L_B	L_G	L_B	L_G	L_B	L_G	L_B	L_G	L_B	L_G	L_B	L_G	L_B
20	4.8	3.0	4.0	2.0																								
25	9.8	8.0	9.0	7.0	8.0	5.7	7.0	4.5																				
30	14.8	13.0	14.0	12.0	13.0	10.7	12.0	9.5	10.0	6.5																		
35	...	...	19.0	17.0	18.0	15.7	17.0	14.5	15.0	11.5	13.0	9.0	11.0	6.0														
40	...	...	24.0	22.0	23.0	20.7	22.0	19.5	20.0	16.5	18.0	14.0	16.0	11.0														
45	...	...	...	...	28.0	25.7	27.0	24.5	25.0	21.5	23.0	19.0	21.0	16.0	17.0	10.7												
50	...	...	...	...	33.0	30.7	32.0	29.5	30.0	26.5	28.0	24.0	26.0	21.0	22.0	15.7	18.0	10.5										
55	...	...	...	...	...	...	37.0	34.5	35.0	31.5	33.0	29.0	31.0	26.0	27.0	20.7	23.0	15.5										
60	...	...	...	...	...	...	42.0	39.5	40.0	36.5	38.0	34.0	36.0	31.0	32.0	25.7	28.0	20.5	24.0	15.2								
65	...	...	...	...	...	...	47.0	44.5	45.0	41.5	43.0	39.0	41.0	36.0	37.0	30.7	33.0	25.5	29.0	20.2	25.0	15.0						
70	...	...	...	...	...	...	...	...	50.0	46.5	48.0	44.0	46.0	41.0	42.0	35.7	38.0	30.5	34.0	25.2	30.0	20.0	26.0	16.0				
80	...	...	...	...	...	...	...	...	60.0	56.5	58.0	54.0	56.0	51.0	52.0	45.7	48.0	40.5	44.0	35.2	40.0	30.0	36.0	26.0				
90	...	...	...	...	...	...	...	...	...	...	68.0	64.0	66.0	61.0	62.0	55.7	58.0	50.5	54.0	45.2	50.0	40.0	46.0	36.0	38.0	25.5		
100	...	...	...	...	...	...	...	...	...	...	78.0	74.0	76.0	71.0	72.0	65.7	68.0	60.5	64.0	55.2	60.0	50.0	56.0	46.0	48.0	35.5	40.0	25.0
110	...	...	...	...	...	...	...	...	...	...	...	...	86.0	81.0	82.0	75.7	78.0	70.5	74.0	65.2	70.0	60.0	66.0	56.0	58.0	45.5	50.0	35.0
120	...	...	...	...	...	...	...	...	...	...	...	...	96.0	91.0	92.0	85.7	88.0	80.5	84.0	75.2	80.0	70.0	76.0	66.0	68.0	55.5	60.0	45.0
130	...	...	...	...	...	...	...	...	...	...	...	...	...	...	102.0	95.7	98.0	90.5	94.0	85.2	90.0	80.0	86.0	76.0	78.0	65.5	70.0	55.0
140	...	...	...	...	...	...	...	...	...	...	...	...	...	...	112.0	105.7	108.0	100.5	104.0	95.2	100.0	90.0	96.0	86.0	88.0	75.5	80.0	65.0
150	...	...	...	...	...	...	...	...	...	...	...	...	...	...	122.0	115.7	118.0	110.5	114.0	105.2	110.0	100.0	106.0	96.0	98.0	85.5	90.0	75.0
160	...	...	...	...	...	...	...	...	...	...	...	...	...	...	132.0	125.7	128.0	120.5	124.0	115.2	120.0	110.0	116.0	106.0	108.0	95.5	100.0	85.0
180	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	148.0	140.5	144.0	135.2	140.0	130.0	136.0	126.0	128.0	115.5	120.0	105.0
200	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	168.0	160.5	164.0	155.2	160.0	150.0	156.0	146.0	148.0	135.5	140.0	125.0
220	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	184.0	175.2	180.0	170.0	176.0	166.0	168.0	155.5	160.0	145.0
240	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	204.0	195.2	200.0	190.0	196.0	186.0	188.0	175.5	180.0	165.0
260	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	220.0	210.0	216.0	206.0	208.0	195.5	200.0	185.0
300	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	256.0	246.0	248.0	235.5	240.0	225.0

SOCKET HEAD CAP SCREWS (METRIC SERIES)

PER ASME/ANSI B18.3.1M-1986

METRIC SOCKET FLAT HEAD CAP SCREWS

Dimensions

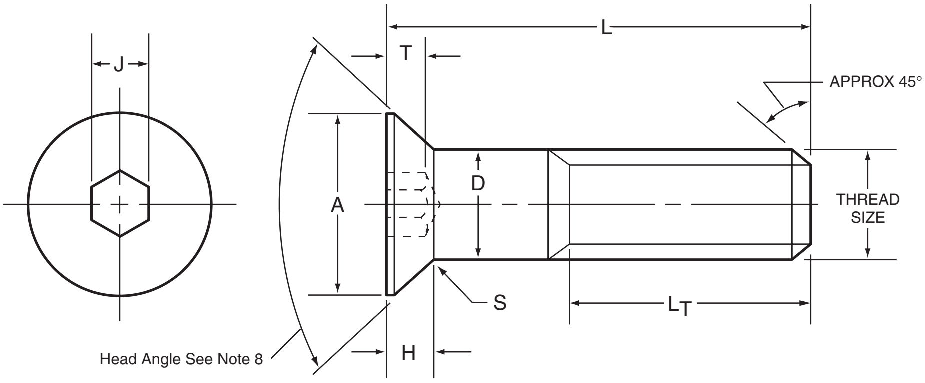
Threads: ANSI B1.13M, ISO 262 (coarse series only)

Applicable or Similar Specification: DIN ENISO10642

General Note: Flat, countersunk head cap screws and button head cap screws are designed and recommended for moderate fastening applications: machine guards, hinges, covers, etc. They are not suggested for use in critical high strength applications where socket head cap screws should be used.

NOTES

1. **Material:** ASTM F835M
2. **Dimensions:** B18.3.5M
3. **Property Class:** 12.9
4. **Hardness:** Rc 38-43 (alloy steel)
5. **Tensile Stress:** 1040MPa
6. **Shear Stress:** 630 MPa
7. **Yield Stress:** 945 MPa
8. **Sizes:** For sizes up to and including M20, head angle shall be 92°/90°. For larger sizes head angle shall be 62°/60°.
9. **Thread Class:** 4g 6g



LENGTH TOLERANCE

nominal screw length	nominal screw diameter M3 thru M24 tolerance on lgth., mm
Up to 16 mm, incl.	±0.3
Over 16 to 60 mm, incl.	±0.5
Over 60 mm	±0.8

DIMENSIONS | **APPLICATION DATA**

nom. thread size	pitch	A max.***	D max.	H ref.	T min.	S ref.	L_T min.	J nom.	recommended seating torque** plain N-m	recommended seating torque** plain in-lbs.
M3	0.5	6.72	3	1.7	1.10	0.50	18	2	1.2	11
M4	0.7	8.96	4	2.3	1.55	0.70	20	2.5	2.8	25
M5	0.8	11.20	5	2.8	2.05	0.70	22	3	5.5	50
M6	1.0	13.44	6	3.3	2.25	0.85	24	4	9.5	85
M8	1.25	17.92	8	4.4	3.20	1.20	28	5	24	210
M10	1.50	22.40	10	5.5	3.80	1.50	32	6	47	415
M12	1.75	26.88	12	6.5	4.35	1.85	36	8	82	725
M16	2.00	33.60	16	7.5	4.89	1.85	44	10	205	1800
M20	2.50	40.32	20	8.5	5.45	1.85	52	12	400	3550
*M24	3.00	40.42	24	14.0	10.15	2.20	60	14	640	5650

All dimensions in millimeters.
*Non-stock Diameter
**Torque calculated to induce 420 MPa in the screw threads.
Torque values are for plain screws. (See Note, page 1.)

***Maximum to theoretical sharp corner

METRIC SOCKET BUTTON HEAD CAP SCREWS

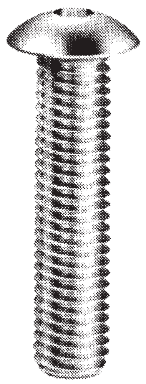
Threads: ANSI B1.13M, ISO 262(coarse series only)

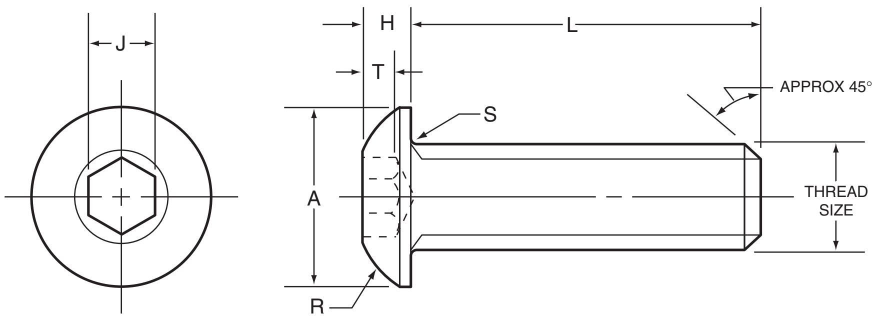
Similar Specifications: ISO 7380

General Note: Flat, countersunk head cap screws and button head cap screws are designed and recommended for moderate fastening applications: machine guards, hinges, covers, etc. They are not suggested for use in critical high strength applications where socket head cap screws should be used.

NOTES

1. **Material:** ASTM F835M
2. **Dimensions:** ANSI B18.3.4M
3. **Property Class:** 12.9
4. **Hardness:** Rc 38-43
5. **Tensile Stress:** 1040 MPa
6. **Shear Stress:** 630 MPa
7. **Yield Stress:** 945 MPa
8. Bearing surface of head square with body within 2°.
9. **Thread Class:** 4g 6g





LENGTH TOLERANCE

nominal screw length	nominal screw diameter M3 thru M16 tolerance on lgth., mm
Up to 16 mm, incl.	±0.3
Over 16 to 60 mm, incl.	±0.5
Over 60 mm	±0.8

DIMENSIONS

APPLICATION DATA

nom. thread size	pitch	A	H	T	R	S	J	recommended seating torque** plain	
		max.	max.	min.	ref.	ref.	nom.	N-m	in-lbs.
M3	0.5	5.70	1.65	1.05	2.95	.35	2.0	1.2	11
M4	0.7	7.60	2.20	1.35	4.10	.35	2.5	2.8	25
M5	0.8	9.50	2.75	1.92	5.20	.45	3.0	5.5	50
M6	1.0	10.50	3.30	2.08	5.60	.45	4.0	9.5	85
M8	1.285	14.00	4.40	2.75	7.50	.45	5.0	24.0	210
M10	1.50	18.00	5.50	3.35	10.00	.60	6.0	47.0	415
M12	1.75	21.00	6.60	4.16	11.00	.60	8.0	82.0	725
*M16	2.0	28.00	8.60	5.20	15.00	.60	10.0	205.0	1800

All dimensions in millimeters.
*Non-stock Diameter
**Torque calculated to induce 420 MPa in the screw threads.
Torque values are for plain screws. (See Note, page 1.)

METRIC SOCKET HEAD SHOULDER SCREWS

Threads: ANSI B 1.13 M, ISO 262

Similar Specifications: ANSI B18.3.3M, ISO 7379, DIN 9841

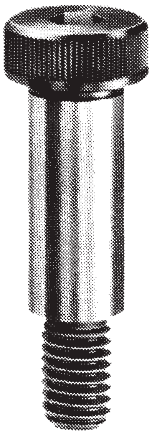
NOTES

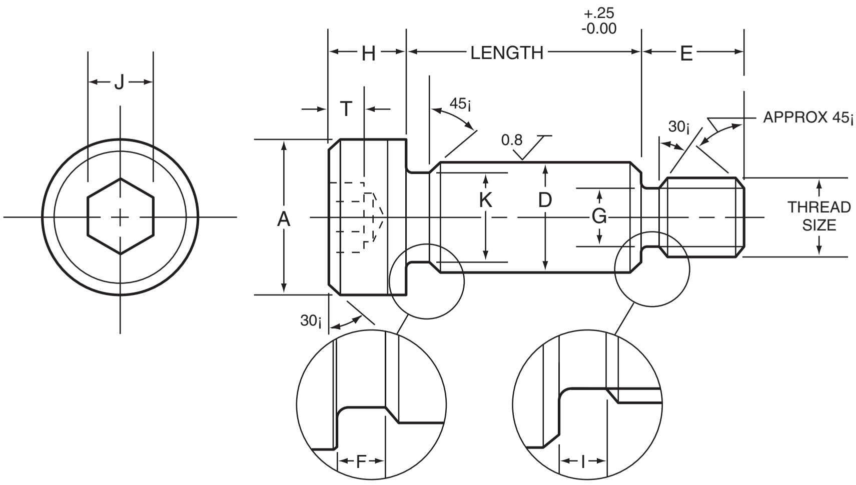
1. **Material:** ASTM A574M alloy steel
2. **Hardness:** Rc 36-43
3. **Tensile Stress:** 1100 MPa based on minimum thread neck area (G min.).
4. **Shear Stress:** 660 MPa
5. **Concentricity:** Body to head O.D. within 0.15 TIR when checked in a “V” block.

Body to thread pitch diameter within 0.1 TIR when checked at a distance of 5mm from the shoulder at the threaded end.

Squareness, concentricity, parallelism, and bow of body to thread pitch diameter shall be within 0.05 TIR per centimeter of body length with a maximum of 0.5 when seated against the shoulder in a threaded bushing and checked on the body at a distance of 2.5 “D” from the underside of the head.

6. **Squareness:** The bearing surface of the head shall be perpendicular to the axis of the body within a maximum deviation of 2°.
7. **Thread Class:** 4g 6g





DIMENSIONS

APPLICATION DATA

nom. size	thread size	pitch	A max.	T min.	D* max.	D* min.	K min.	H max.	G min.	F max.	I max.	E max.	J nom.	recommended seating torque** N-m	recommended seating torque** in-lbs.
6	M5	0.8	10.00	2.4	6.0	5.982	5.42	4.50	3.68	2.5	2.40	9.75	3	7	60
8	M6	1.0	13.00	3.3	8.0	7.978	7.42	5.50	4.40	2.5	2.60	11.25	4	12	105
10	M8	1.25	16.00	4.2	10.0	9.978	9.42	7.00	6.03	2.5	2.80	13.25	5	29	255
12	M10	1.5	18.00	4.9	12.0	11.973	11.42	8.00	7.69	2.5	3.00	16.40	6	57	500
16	M12	1.75	24.00	6.6	16.0	15.973	15.42	10.00	9.35	2.5	4.00	18.40	8	100	885
20	M16	2.0	30.00	8.8	20.0	19.967	19.42	14.00	12.96	2.5	4.80	22.40	10	240	2125
24	M20	2.5	36.00	10.0	24.0	23.967	23.42	16.00	16.30	3.0	5.60	27.40	12	470	4160

All dimensions in millimeters.
*Shoulder diameter tolerance h8 (ISO R 286)

**See Note, page 1.

Unbrako

METRIC SOCKET SET SCREWS ■ Knurled Cup Point and Plain Cup Point ■ Dimensions

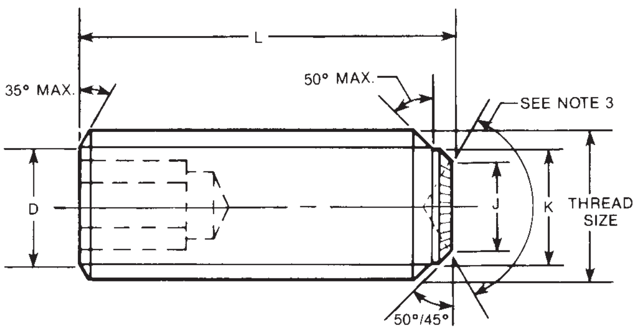
Threads: ANSI B 1.13M, ISO 261, ISO 262 (coarse series only)

Grade: 45H

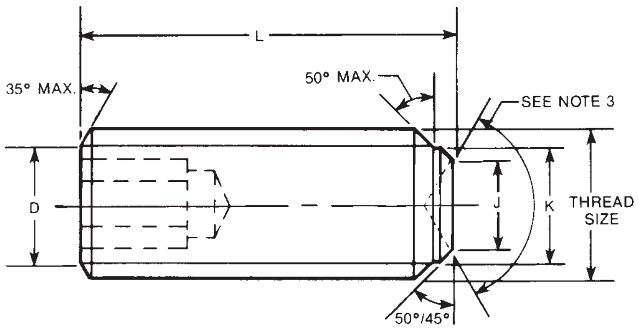
Applicable or Similar Specifications: ANSI B 18.3.6M, ISO 4029, DIN 916, DIN 915, DIN 914, DIN 913

NOTES

1. **Material:** ASTM F912M
2. **Hardness:** Rockwell C45-53
3. **Angle:** The cup angle is 135 maximum for screw lengths equal to or smaller than screw diameter. For longer lengths, the cup angle will be 124 maximum
4. **Thread Class:** 6g



KNURLED CUP POINT



PLAIN CUP POINT

LENGTH TOLERANCE

nominal screw length	nominal screw diameter M1.6 thru M24 tolerance on lgth., mm
Up to 12 mm, incl.	±0.3
Over 12 to 50 mm, incl.	±0.5
Over 50 mm	±0.8

DIMENSIONS

APPLICATION DATA

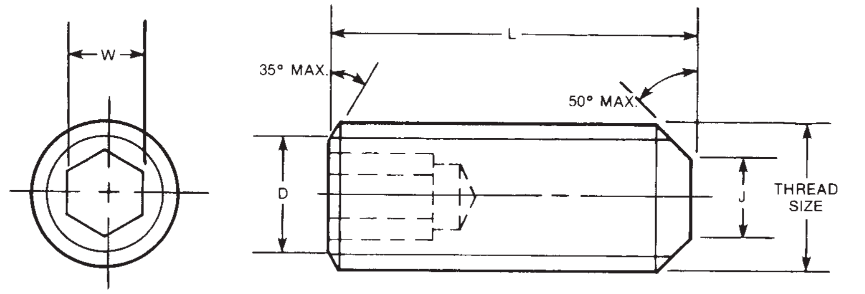
nom. thread size	pitch	D max.	J max. plain cup	J max. knurled cup	K max.	L min. preferred plain cup	L min. preferred knurled cup	W nom.	recommended* seating torque N-m	recommended* seating torque in-lbs.
MICROSIZE – Plain Cup Only										
M1.6	0.35	1.0	0.80	–	–	2.0	–	0.7	0.09	0.8
M2	0.40	1.32	1.00	–	–	2.5	–	0.9	0.21	1.8
M2.5	0.45	1.75	1.25	–	–	3.0	–	1.3	0.57	5.0
STANDARD SIZE – Knurled Cup Point Supplied Unless Plain Cup Point Is Specified										
M3	0.5	2.10	1.50	1.40	2.06	3.0	3.0	1.5	0.92	8.0
M4	0.7	2.75	2.00	2.10	2.74	3.0	3.0	2.0	2.2	19.0
M5	0.8	3.70	2.50	2.50	3.48	4.0	4.0	2.5	4.0	35.0
M6	1.0	4.35	3.00	3.30	4.14	4.0	5.0	3.0	7.2	64
M8	1.25	6.00	5.00	5.00	5.62	5.0	6.0	4.0	17.0	150.0
M10	1.5	7.40	6.00	6.00	7.12	6.0	8.0	5.0	33.0	290
M12	1.75	8.60	8.00	8.00	8.58	8.0	10.0	6.0	54.0	480
M16	2.0	12.35	10.00	10.00	11.86	12.0	14.0	8.0	134	1190
M20	2.5	16.00	14.00	14.00	14.83	16.0	18.0	10.0	237	2100
M24	3.0	18.95	16.00	16.00	17.80	20.0	20.0	12.0	440	3860

All dimensions in millimeters.
*Not applicable to screws with a length equal to or less than the diameter. See Note, page 1.

METRIC SOCKET SET SCREW

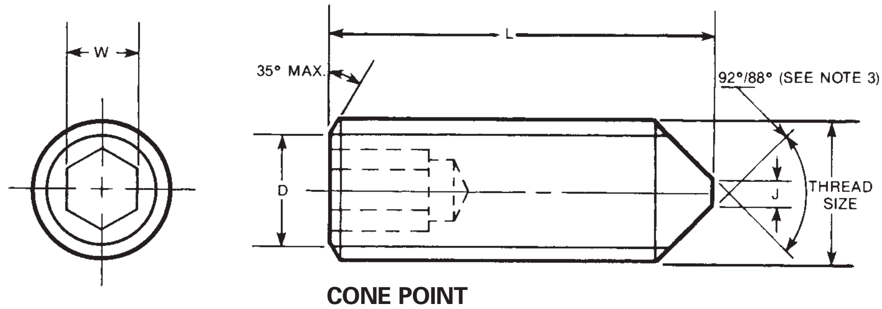
Flat Point, Cone Point, Dog Point Styles ■ Dimensions

REF. ISO 4026



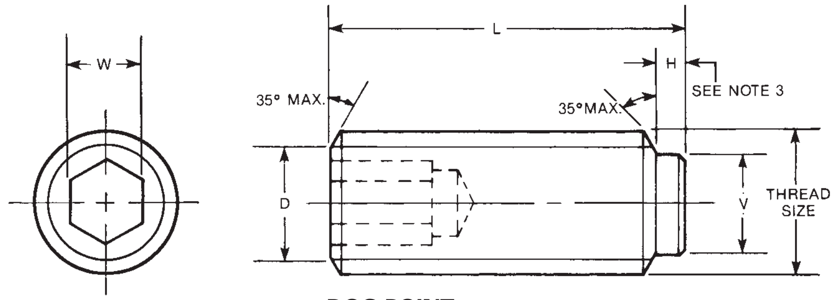
FLAT POINT

REF. ISO 4027



CONE POINT

REF. ISO 4028
ISO 7435



DOG POINT

DIMENSIONS

			flat point		cone point		dog point			
							H nom.			
nom. thread size	pitch	D max.	J max.	L min. preferred	J max.	L min. preferred	short lgth.	long lgth.	L min. preferred	V max.
M3	0.5	2.10	2.0	3.0	0.3	4.0	0.75	1.5	5.0	2.00
M4	0.7	2.75	2.5	3.0	0.4	4.0	1.00	2.0	5.0	2.50
M5	0.8	3.70	3.5	4.0	0.5	5.0	1.25	2.5	6.0	3.50
M6	1.00	4.25	4.0	4.0	1.5	6.0	1.50	3.0	6.0	4.00
M8	1.25	6.00	5.5	5.0	2.0	6.0	2.00	4.0	8.0	5.50
M10	1.50	7.40	7.0	6.0	2.5	8.0	2.50	5.0	8.0	7.00
M12	1.75	8.60	8.5	8.0	3.0	10.0	3.00	6.0	12.0	8.50
M16	2.00	12.35	12.0	12.0	4.0	14.0	4.00	8.0	16.0	12.00
M20	2.50	16.00	15.0	14.0	6.0	18.0	5.00	10.0	20.0	15.00
M24	3.00	18.95	18.0	20.0	8.0	20.0	6.00	12.0	22.0	18.00

METRIC LOW HEAD CAP SCREWS

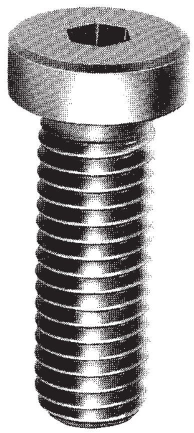
Threads: ANSI B 1.13M, ISO 262 (coarse series only)

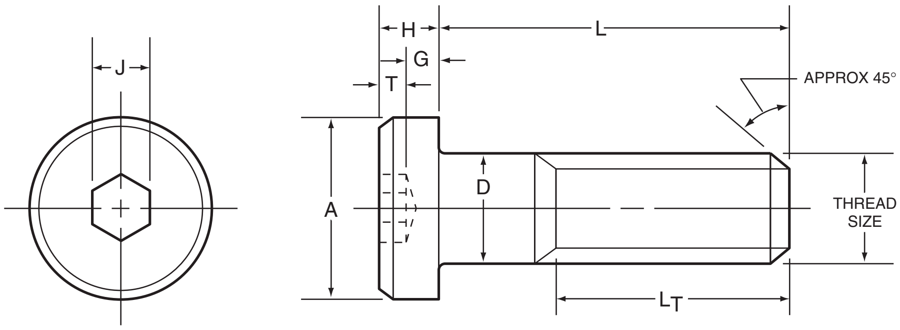
Property Class: 10.9

Similar Specifications: DIN 7984, DIN 6912

NOTES

1. **Material:** ASTM A574M-alloy steel
2. **Hardness:** Rc 33-39
3. **Tensile Stress:** 1040 MPa
4. **Yield Stress:** 940 MPa
5. **Thread Class:** 6g





DIMENSIONS

APPLICATION DATA

nom. thread size	pitch	A	D	G	T	H	L_T	J	recommended* seating torque	
									plain	
		max.	max.	min.	min.	max.	min.	nom.	N-m	in-lbs.
M4	0.7	7	4	1.06	1.48	2.8	20	3	4.5	40
M5	0.8	8.5	5	1.39	1.85	3.5	22	4	8.5	75
M6	1.0	10	6	1.65	2.09	4.0	24	5	14.5	130
M8	1.25	13	8	2.24	2.48	5.0	28	6	35	310
M10	1.5	16	10	2.86	3.36	6.5	32	8	70	620
M12	1.75	18	12	3.46	4.26	8.0	36	10	120	1060
M16	2.0	24	16	4.91	4.76	10.0	44	12	300	2650
M20	2.5	30	20	6.10	6.07	12.5	52	14	575	5100

All dimensions in millimeters.
*Torque calculated to induce 620 MPa in the screw threads.
Torque values are for plain screws. (See Note, page 1.)

METRIC HEXAGON KEYS

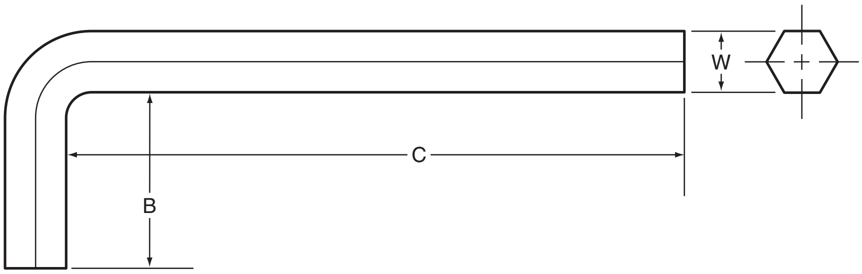
Dimensions ■ Mechanical Properties ■ Socket Applications

These UNBRAKO keys are made to higher requirements than ISO or DIN keys, which may not properly torque Class 12.9 cap screws. The strength and dimensional requirements are necessary to properly install the products in this catalog.

Material: ANSI B18.3.2.M alloy steel

Dimensions: ANSI B18.3.2.M

Similar Specifications: DIN 911, ISO 2936



METRIC KEY APPLICATION CHART

size W	socket cap screws: std. head height	socket cap screws: low head	socket cap screws	flat head socket cap screws	button head shoulder screws	socket set screws
0.7						M1.6
0.9						M2
1.3						M2.5
1.5	M1.6/M2					M3
2.0	M2.5		M3	M3		M4
2.5	M3		M4	M4		M5
3.0	M4	M4	M5	M5	M6	M6
4.0	M5	M5	M6	M6	M8	M8
5.0	M6	M6	M8	M8	M10	M10
6.0	M8	M8	M10	M10	M12	M12
8.0	M10	M10	M12	M12	M16	M16
10.0	M12	M12	M16	M16	M20	M20
12.0	M14	M16	M20		M24	M24
14.0	M16	M20	M24			
17.0	M20	M24				
19.0	M24					
22.0	M30					
27.0	M36					
32.0	M42					
36.0	M48					

DIMENSIONS

MECHANICAL PROPERTIES

key size W		B	C nominal		torsional shear strength minimum		torsional yield strength minimum	
max.	min.	mominal	short arm	long arm	N-m	In-lbs.	N-m	In-lbs.
0.711	0.698	5.5	31	*69	0.12	1.1	0.1	0.9
0.889	0.876	9	31	71	0.26	2.3	0.23	2.
1.270	1.244	13.5	42	75	0.73	6.5	.63	5.6
1.500	1.470	14	45	78	1.19	10.5	1.02	9.
2.000	1.970	16	50	83	2.9	26	2.4	21
2.500	2.470	18	56	90	5.4	48	4.4	39
3.000	2.960	20	63	100	9.3	82	8.	71
4.000	3.960	25	70	106	22.2	196	18.8	166
5.000	4.960	28	80	118	42.7	378	36.8	326
6.000	5.960	32	90	140	74	655	64	566
8.000	7.950	36	100	160	183	1,620	158	1,400
10.000	9.950	40	112	170	345	3,050	296	2,620
12.000	11.950	45	125	212	634	5,610	546	4,830
14.000	13.930	55	140	236	945	8,360	813	7,200
17.000	16.930	60	160	250	1,690	15,000	1,450	12,800
19.000	18.930	70	180	280	2,360	20,900	2,030	18,000
22.000	21.930	80	*200	*335	3,670	32,500	3,160	28,000
24.000	23.930	90	*224	*375	4,140	36,600	3,560	31,500
27.000	26.820	100	*250	*500	5,870	51,900	5,050	44,700
32.000	31.820	125	*315	*630	8,320	73,600	7,150	63,300
36.000	35.820	140	*355	*710	11,800	104,000	10,200	90,300

All dimensions in millimeters.
*Non-stock sizes

Unbrako

ISO TOLERANCES FOR METRIC FASTENERS

nominal dimension		tolerance zone in mm (external measurements)													tolerance zone in mm					
over	**to**	**h6**	**h8**	**h10**	**h11**	**h13**	**h14**	**h15**	**h16**	**js14**	**js15**	**js16**	**js17**	**m6**	**H7**	**H8**	**H9**	**H11**	**H13**	**H14**
0	**1**	0	0	0	0	0								+0.002	+0.010	+0.0014	+0.025	+0.060	+0.14	
		–0.006	–0.014	–0.040	–0.060	–0.14								+0.008	0	0	0	0	0	
1	**3**	0	0	0	0	0	0	0	0	±0.125	±0.20	±0.30	±0.50	+0.002	+0.010	+0.014	+0.025	+0.060	+0.14	+0.25
		–0.006	–0.014	–0.040	–0.060	–0.14	–0.25	–0.40	–0.60					+0.008	0	0	0	0	0	0
3	6	0	0	0	0	0	0	0	0	±0.15	±0.24	±0.375	±0.60	+0.004	+0.012	+0.018	+0.030	+0.075	+0.18	+0.30
		–0.008	–0.018	–0.048	–0.075	–0.18	–0.30	–0.48	–0.75					+0.012	0	0	0	0	0	0
6	10	0	0	0	0	0	0	0	0	±0.18	±0.29	±0.45	±.075	+0.006	+0.015	+0.022	+0.036	+0.090	+0.22	+0.36
		–0.009	–0.022	–0.058	–0.090	–0.22	–0.36	–0.58	–0.90					+0.0015	0	0	0	0	0	0
10	18	0	0	0	0	0	0	0	0	±0.215	±0.35	±0.55	±0.90	+0.007	+0.018	+0.027	+0.043	+0.110	+0.27	+0.43
		–0.011	–0.027	–0.070	–0.110	–0.27	–0.43	–0.70	–1.10					+0.018	0	0	0	0	0	0
18	30	0	0	0	0	0	0	0	0	± 0.26	±0.42	±0.65	±1.05	+0.008	+0.021	+0.033	+0.052	+0.130	+0.33	+0.52
		–0.030	–0.033	–0.084	–0.130	–0.33	–0.52	–0.84	–1.30					+0.021	0	0	0	0	0	0
30	50					0	0	0	0	±0.31	±0.50	±0.80	±1.25						+0.39	+0.62
						–0.39	–0.62	–1.00	–1.60										0	0
50	80					0	0	0	0	±0.37	±0.60	±0.95	±1.50						+0.46	+0.74
						–0.46	–0.74	–1.20	–1.90										0	0
80	120					0	0	0	0	±0.435	±0.70	±1.10	±1.75						+0.54	+0.87
						–0.54	–0.87	–1.40	–2.20										0	0
120	180									±0.50	±0.80	±1.25	±2.00							
180	250									±0.575	±0.925	±1.45	±2.30							
250	315									±0.65	±1.05	±1.60	±2.60							
315	400									±0.70	±1.15	±1.80	±2.85							
400	500									±0.775	±1.25	±2.00	±3.15							

ISO TOLERANCES FOR SOCKET SCREWS

nominal dimension		tolerance zone in mm										
over	**to**	**C13**	**C14**	**D9**	**D10**	**D11**	**D12**	**EF8**	**E11**	**E12**	**Js9**	**K9**
	3	+0.20	+0.31	+0.045	+0.060	+0.080	+0.12	+0.024	+0.074	+0.100		0
		+0.06	+0.06	+0.020	+0.020	+0.020	+0.02	+0.010	+0.014	+0.014	±0.0125	-0.025
3	**6**	+0.24	+0.37	+0.060	+0.078	+0.115	+0.15	+0.028	+0.095	+0.140		0
		+0.06	+0.07	+0.030	+0.030	+0.030	+0.03	+0.014	+0.020	+0.020	±0.015	-0.030
6	**10**					+0.130	+0.19	+0.040	+0.115	+0.115		0
						+0.040	+0.40	+0.018	+0.025	+0.025	±0.018	-0.036
10	**18**						+0.2		+0.142	+0.212		
							+0.05		+0.032	+0.032		
18	**30**						+0.275					
							+0.065					
30	**50**						+0.33					
							+0.08					
50	**80**						+0.40					
							+0.10					

References
ISO R 286
ISO 4759/I
ISO 4759/II
ISO 4759/III

Notes

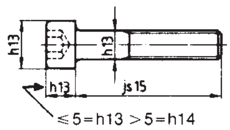
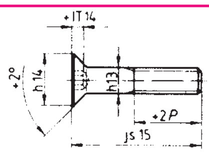
ANSI standards allow slightly wider tolerances for screw lengths than ISO and DIN.

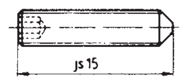
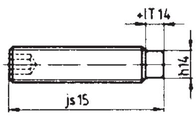
The table is intended to assist in the design with metric fasteners. For tolerances not listed here refer to the complete standards.

Tolerances for Metric Fasteners

The tolerances in the tables below are derived from ISO standard: ISO 4759

The tables show tolerances on the most common metric fasteners. However, occasionally some slight modifications are made.

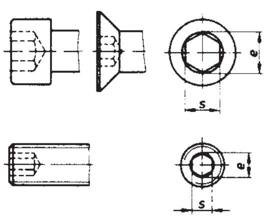
Item	DIN
	912
	7991

Item	DIN
	913 914 916
	915 966

Notes

Product grade A applies to sizes up to M24 and length not exceeding 10 x diameter or 150 mm, whatever is shorter.

Product grade B applies to the sizes above M24 and all sizes with lengths, greater than 10 x diameter or 150 mm, whichever is shorter.

Feature			
Hexagon Sockets 	s	Tolerance *	Tolerance **
	0.7	EF8	
	0.9	JS9	
	1.3	K9	
	1.5	D9	D10
	2		
	2.5	D10	D11
	3	D11	
	4	E11	E11
	5		E12
	6		
	8		
	10		
	12		
	14		
	>14	D12	

*Tolerance zones for socket set screws

**Tolerance zones for socket head cap screws

Note: For S 0.7 to 1.3 the actual allowance in the product standards has been slightly modified for technical reasons.

CONVERSION CHART

SI UNITS & CONVERSIONS FOR CHARACTERISTICS OF MECHANICAL FASTENERS						
			conversion			
property	**unit**	**symbol**	**from**	**to**	**multiply by**	**approximate equivalent**
length	meter centimeter millimeter	m cm mm	inch inch foot	mm cm mm	25.4 2.54 304.8	25mm = 1 in. 300mm = 1ft. 1m = 39.37 in.
mass	kilogram gram tonne (megagram)	kg g t	once pound ton (2000 lb)	g kg kg	28.35 .4536 907.2	28g = 1 oz. 1kg = 2.2 lb. = 35 oz. 1t = 2200 lbs.
density	kilogram per cubic meter	kg/m^3	pounds per cu. ft.	kg/m	16.02	16kg/m = 1 lb/ft.^{3}
temperature	deg. Celsius	°C	deg. Fahr.	°C	(°F – 32) x 5/9	0°C = 32°F 100°C = 212°F
area	square meter square millimeter	m^2 mm^2	sq. in. sq. ft.	mm^2 m^2	645.2 .0929	645mm^2 = 1 in.^{2} 1m^2 = 11 ft.^{2}
volume	cubic meter cubic centimeter cubic millimeter	m^3 cm^3 mm^3	cu. in. cu.ft. cu. yd.	mm^3 m^3 m^3	16387 .02832 .7645	16400mm^3 = 1 in.^{3} 1m^3 = 35 ft.^{3} 1m^3 = 1.3 yd.^{3}
force	newton kilonewton meganewton	N kN MN	ounce (Force) pound (Force) Kip	N kN MN	.278 .00445 .00445	1N = 3.6 ozf 4.4N = 1 lbf 1kN = 225 lbf
stress	megapascal newtons/sq.m	MPa N/m^2	pound/in^2 (psi) Kip/in^2 (ksi)	MPa MPa	.0069 6.895	1MPa = 145 psi 7MPa = 1 ksi
torque	newton-meter	N•m	inch-ounce inch-pound foot-pound	N-m N-m N-m	.00706 .113 1.356	1N•m = 140 in. oz. 1N•m = 9 in. lb. 1N•m = .75 ft. lb. 1.4 N•m = 1 ft. lb.

TABLE OF CONTENTS

Technical Section

IMPORTANT

The technical discussions represent typical applications only. The use of the information is at the sole discretion of the reader. Because applications vary enormously, UNBRAKO does not warrant the scenarios described are appropriate for any specific application. The reader must consider all variables prior to using this information.

INSTALLATION CONTROL

Several factors should be considered in designing a joint or selecting a fastener for a particular application.

JOINT DESIGN AND FASTENER SELECTION.

Joint Length

The longer the joint length, the greater the total elongation will occur in the bolt to produce the desired clamp load or preload. In design, if the joint length is increased, the potential loss of preload is decreased.

Joint Material

If the joint material is relatively stiff compared to the bolt material, it will compress less and therefore provide a less sensitive joint, less sensitive to loss of preload as a result of brinelling, relaxation and even loosening.

Thread Stripping Strength

Considering the material in which the threads will be tapped or the nut used, there must be sufficient engagement length to carry the load. Ideally, the length of thread engagement should be sufficient to break the fastener in tension. When a nut is used, the wall thickness of the nut as well as its length must be considered.

An estimate, a calculation or joint evaluation will be required to determine the tension loads to which the bolt and joint will be exposed. The size bolt and the number necessary to carry the load expected, along with the safety factor, must also be selected.

The safety factor selected will have to take into consideration the consequence of failure as well as the additional holes and fasteners. Safety factors, therefore, have to be determined by the designer.

SHEAR APPLICATIONS

Shear Strength of Material

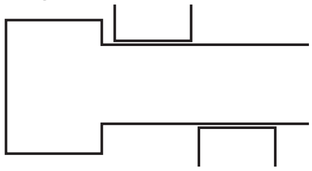
Not all applications apply a tensile load to the fastener. In many cases, the load is perpendicular to the fastener in shear. Shear loading may be single, double or multiple loading.

There is a relationship between the tensile strength of a material and its shear strength. For alloy steel, the shear strength is 60% of its tensile strength. Corrosion resistant steels (e.g. 300-Series stainless steels) have a lower tensile/shear relationship and it is usually 50-55%

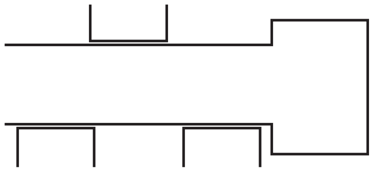
Single/Double Shear

Single shear strength is exactly one-half the double shear value. Shear strength listed in pounds per square inch (psi) is the shear load in pounds multiplied by the cross sectional area in square inches.

Single Shear



Double Shear



OTHER DESIGN CONSIDERATIONS

Application Temperature

For elevated temperature, standard alloy steels are useful to about 550°F–600°F. However, if plating is used, the maximum temperature may be less (eg. cadmium should not be used over 450°F.

Austenitic stainless steels (300 Series) may be useful to 800°F. They can maintain strength above 800°F but will begin to oxidize on the surface.

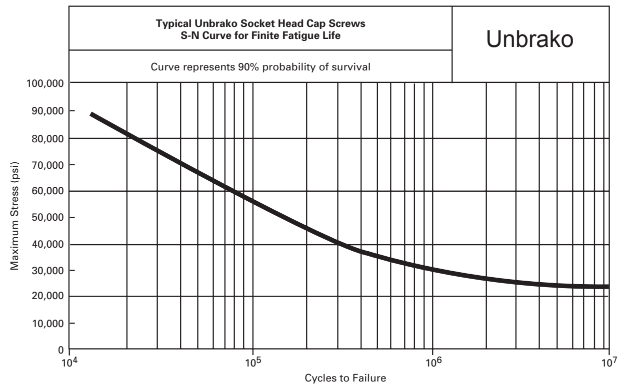
Corrosion Environment

A plating may be selected for mild atmospheres or salts. If plating is unsatisfactory, a corrosion resistant fastener may be specified. The proper selection will be based upon the severity of the corrosive environment.

FATIGUE STRENGTH

S/N Curve

Most comparative fatigue testing and specification fatigue test requirements are plotted on an S/N curve. In this curve, the test stress is shown on the ordinate (y-axis) and the number of cycles is shown on the abscissa (x-axis) in a lograthmic scale. On this type curve, the high load to low load ratio must be shown. This is usually R =.1, which means the low load in all tests will be 10% of the high load.



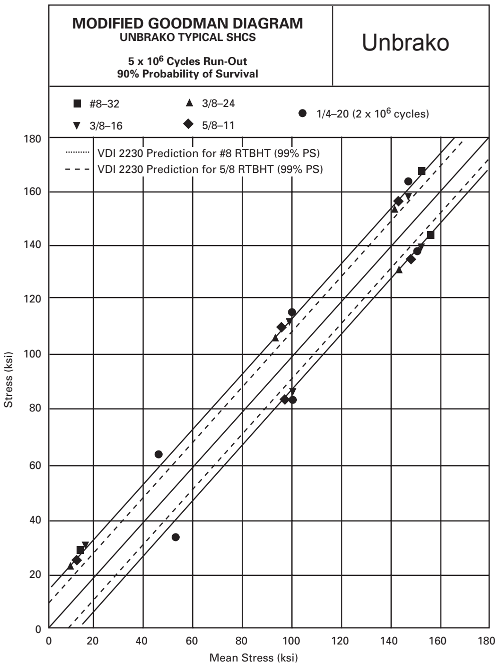
Effect of Preload

Increasing the R to .2, .3 or higher will change the curve shape. At some point in this curve, the number of cycles will reach 10 million cycles. This is considered the endurance limit or the stress at which infinite life might be expected.

Modified Goodman/ Haigh Soderberg Curve

The S/N curve and the information it supplies will not provide the information needed to determine how an individual fastener will perform in an actual application. In application, the preload should be higher than any of the preloads on the S/N curve.

Therefore, for application information, the modified Goodman Diagram and/or the Haigh Soderberg Curve are more useful. These curves will show what fatigue performance can be expected when the parts are properly preloaded.



METHODS OF PRELOADING

Elongation

The modulus for steel of 30,000,000 (thirty million) psi means that a fastener will elongate .001 in/in of length for every 30,000 psi in applied stress. Therefore, if 90,000 psi is the desired preload, the bolt must be stretched .003 inches for every inch of length in the joint.

This method of preloading is very accurate but it requires that the ends of the bolts be properly prepared and also that all measurements be very carefully made. In addition, direct measurements are only possible where both ends of the fastener are available for measurement after installation. Other methods of measuring length changes are ultrasonic, strain gages and turn of the nut.

Torque

By far, the most popular method of preloading is by torque. Fastener manufacturers usually have recommended seating torques for each size and material fastener. The only requirement is the proper size torque wrench, a conscientious operator and the proper torque requirement.

Strain

Since stress/strain is a constant relationship for any given material, we can use that relationship just as the elongation change measurements were used previously.

Now, however, the strain can be detected from strain gages applied directly to the outside surface of the bolt or by having a hole drilled in the center of the bolt and the strain gage installed internally. The output from these gages need instrumentation to convert the gage electrical measurement method. It is, however, an expensive method and not always practical.

Turn of the Nut

The nut turn method also utilizes change in bolt length. In theory, one bolt revolution (360° rotation) should increase the bolt length by the thread pitch. There are at least two variables, however, which influence this relationship. First, until a snug joint is obtained, no bolt elongation can be measured. The snugging produces a large variation in preload. Second, joint compression is also taking place so the relative stiffnesses of the joint and bolt influences the load obtained.

VARIABLES IN TORQUE

Coefficient of Friction

Since the torque applied to a fastener must overcome all friction before any loading takes place, the amount of friction present is important.

In a standard unlubricated assembly, the friction to be overcome is the head bearing area and the thread-to-thread friction. Approximately 50% of the torque applied will be used to overcome this head-bearing friction and approximately 35% to overcome the thread friction. So 85% of the torque is overcoming friction and only 15% is available to produce bolt load.

If these interfaces are lubricated (cadmium plate, molybdenum disulfide, anti-seize compounds, etc.), the friction is reduced and thus greater preload is produced with the same torque.

The change in the coefficient of friction for different conditions can have a very significant effect on the slope of the torque tension curve. If this is not taken into consideration, the proper torque specified for a plain unlubricated bolt may be sufficient to yield or break a lubricated fastener.

Thread Pitch

The thread pitch must be considered when a given stress is to be applied, since the cross-sectional area used for stress calculations is the thread tensile stress area and is different for coarse and fine threads. The torque recommendations, therefore, are slightly higher for fine threads than for coarse threads to achieve the same stress.

Differences between coarse and fine threads.

Coarse Threads are…

- more readily available in industrial fasteners.
- easier to assemble because of larger helix angle.
- require fewer turns and reduce cross threading.
- higher thread stripping strength per given length.
- less critical of tap drill size.
- not as easily damaged in handling.

Their disadvantages are…
- lower tensile strength.
- reduced vibrational resistance.
- coarse adjustment.

Fine Threads provide...
- higher tensile strength.
- greater vibrational resistance.
- finer adjustment.

Their disadvantages are…
- easier cross threaded.
- threads damaged more easily by handling.
- tap drill size slightly more critical.
- slightly lower thread stripping strength.

Other Design Guidelines

In addition to the joint design factors discussed, the following considerations are important to the proper use of high-strength fasteners.

- Adequate thread engagement should be guaranteed by use of the proper mating nut height for the system. Minimum length of engagement recommended in a tapped hole depends on the strength of the material, but in all cases should be adequate to prevent stripping.
- Specify nut of proper strength level. The bolt and nut should be selected as a system.
- Specify compatible mating female threads. 2B tapped holes or 3B nuts are possibilities.
- Corrosion, in general, is a problem of the joint, and not just of the bolt alone. This can be a matter of galvanic action between dissimilar metals. Corrosion of the fastener material surrounding the bolt head or nut can be critical with high-strength bolting. Care must be exercised in the compatibility of joint materials and/or coatings to protect dissimilar metals.

PROCESSING CONTROL

The quality of the raw material and the processing control will largely affect the mechanical properties of the finished parts.

MATERIAL SELECTION

The selection of the type of material will depend on its end use. However, the control of the analysis and quality is a critical factor in fastener performance. The material must yield reliable parts with few hidden defects such as cracks, seams, decarburization and internal flaws.

FABRICATION METHOD

Head

There are two general methods of making bolt heads, forging and machining. The economy and grain flow resulting from forging make it the preferred method.

The temperature of forging can vary from room temperature to 2000°F. By far, the greatest number of parts are cold upset on forging machines known as headers or boltmakers. For materials that do not have enough formability for cold forging, hot forging is used. Hot forging is also used for bolts too large for cold upsetting due to machine capacity. The largest cold forging machines can make bolts up to 1-1/2 inch diameter. For large quantities of bolts, hot forging is more expensive then cold forging.

Some materials, such as stainless steel, are warm forged at temperatures up to 1000°F. The heating results in two benefits, lower forging pressures due to lower yield strength and reduced work hardening rates.

Machining is the oldest method and is used for very large diameters or small production runs.

The disadvantage is that machining cuts the metal grain flow, thus creating planes of weakness at the critical head-to-shank fillet area. This can reduce tension fatigue performance by providing fracture planes.

Fillets

The head-to-shank transition (fillet) represents a sizable change in cross section at a critical area of bolt performance. It is important that this notch effect be minimized. A generous radius in the fillet reduces the notch effect. However, a compromise is necessary because too large a radius will reduce load-bearing area under the head.

Composite radii such as elliptical fillets, maximize curvature on the shank side of the fillet and minimize it on the head side to reduce loss of bearing area on the load-bearing surface.

Critical Fastener Features

Head-Shank-Fillet: This area on the bolt must not be restricted or bound by the joint hole. A sufficient chamfer or radius on the edge of the hole will prevent interference that could seriously reduce fatigue life. Also, if the bolt should seat on an unchamfered edge, there might be serious loss of preload if the edge breaks under load.

Threads

Threads can be produced by grinding, cutting or rolling.

In a rolled thread, the material is caused to flow into the thread die contour, which is ground into the surface during the manufacture of the die. Machines with two or three circular dies or two flat dies are most common.

Thread cutting requires the least tooling costs and is by far the most popular for producing internal threads. It is the most practical method for producing thin wall parts and the only technique available for producing large diameter parts (over 3 inches in diameter).

Thread grinding yields high dimensional precision and affords good control of form and finish. It is the only practical method for producing thread plug gages.

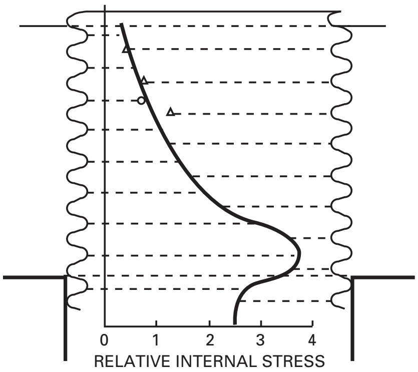
Both machining and grinding have the disadvantage of cutting material fibers at the most critical point of performance.

The shape or contour of the thread has a great effect on the resulting fatigue life. The thread root should be large and well rounded without sharp corners or stress risers. Threads with larger roots should always be used for harder materials.

In addition to the benefits of grain flow and controlled shape in thread rolling, added fatigue life can result when the rolling is performed after heat treatment.

This is the accepted practice for high fatigue performance bolts such as those used in aircraft and space applications.

FASTENER POINT END



RELATIVE INTERNAL STRESS
AT FIRST ENGAGED THREAD
FASTENER HEAD END

EVALUATING PERFORMANCE

Mechanical Testing

In the fastener industy, a system of tests and examinations has evolved which yields reliable parts with proven performance.

Some tests are conducted on the raw material; some on the finished product.

There always seems to be some confusion regarding mechanical versus metallurgical properties. Mechanical properties are those associated with elastic or inelastic reaction when force is applied, or that involve the relationship between stress and strain. Tensile testing stresses the fastener in the axial direction. The force at which the fastener breaks is called the breaking load or ultimate tensile strength. Load is designated in pounds, stress in pounds per square inch and strain in inches per inch.

When a smooth tensile specimen is tested, the chart obtained is called a Stress-Strain Curve. From this curve, we can obtain other useful data such as yield strength. The method of determining yield is known as the offset method and consists of drawing a straight line parallel to the stress strain curve but offset from the zero point by a specified amount. This value is usually 0.2% on the strain ordinate. The yield point is the intersection of the stress-strain curve and the straight line. This method is not applicable to fasteners because of the variables introduced by their geometry.

When a fastener tensile test is plotted, a load/elongation curve can be obtained. From this curve, a yield determination known as Johnson's 2/3 approximate method for determination of yield strength is used to establish fastener yield, which will be acceptable for design purposes. It is not recommended for quality control or specification requirements.

Torque-tension testing is conducted to correlate the required torque necessary to induce a given load in a mechanically fastened joint. It can be performed by hand or machine. The load may be measured by a tensile machine, a load cell, a hydraulic tensile indicator or by a strain gage.

Fatigue tests on threaded fasteners are usually alternating tension-tension loading. Most testing is done at more severe strain than its designed service load but ususally below the material yield strength.

Shear testing, as previously mentioned, consists of loading a fastener perpendicular to its axis. All shear testing should be accomplished on the unthreaded portion of the fastener.

Checking hardness of parts is an indirect method for testing tensile strength. Over the years, a correlation of tensile strength to hardness has been obtained for most materials. See page 83 for more detailed information. Since hardness is a relatively easy and inexpensive test, it makes a good inspection check. In hardness checking, it is very important that the specimen be properly prepared and the proper test applied.

Stress durability is used to test parts which have been subjected to any processing which may have an embrittling effect. It requires loading the parts to a value higher than the expected service load and maintaining that load for a specified time after which the load is removed and the fastener examined for the presence of cracks.

Impact testing has been useful in determining the ductile brittle transformation point for many materials. However, because the impact loading direction is transverse to a fastener's normal longitude loading, its usefulness for fastener testing is minimal. It has been shown that many fastener tension impact strengths do not follow the same pattern or relationship of Charpy or Izod impact strength.

Metallurgical Testing

Metallurgical testing includes chemical composition, microstructure, grain size, carburization and decarburization, and heat treat response.

The chemical composition is established when the material is melted. Nothing subsequent to that process will influence the basic composition.

The microstructure and grain size can be influenced by heat treatment. Carburization is the addition of carbon to the surface which increases hardness. It can occur if heat treat furnace atmospheres are not adequately controlled. Decarburization is the loss of carbon from the surface, making it softer. Partial decarburization is preferable to carburization, and most industrial standards allow it within limits.

In summary, in order to prevent service failures, many things must be considered:

The Application Requirements

Strength Needed – Safety Factors
- Tension/Shear/Fatigue
- Temperature
- Corrosion
- Proper Preload

The Fastener Requirements

- Material
- Fabrication Controls
- Performance Evaluations

Unbrako

AN EXPLANATION OF JOINT DIAGRAMS

When bolted joints are subjected to external tensile loads, what forces and elastic deformation really exist? The majority of engineers in both the fastener manufacturing and user industries still are uncertain. Several papers, articles, and books, reflecting various stages of research into the problem have been published and the volume of this material is one reason for confusion. The purpose of this article is to clarify the various explanations that have been offered and to state the fundamental concepts which apply to forces and elastic deformations in concentrically loaded joints. The article concludes with general design formulae that take into account variations in tightening, preload loss during service, and the relation between preloads, external loads and bolt loads.

The Joint Diagram

Forces less than proof load cause elastic strains. Conversely, changes in elastic strains produce force variations. For bolted joints this concept is usually demonstrated by joint diagrams.

The most important deformations within a joint are elastic bolt elongation and elastic joint compression in the axial direction. If the bolted joint in Fig. 1 is subjected to the preload F_i the bolt elongates as shown by the line OB in Fig. 2A and the joint compresses as shown by the line OJ. These two lines, representing the spring characteristics of the bolt and joint, are combined into one diagram in Fig. 2B to show total elastic deformation.

If a concentric external load F_e is applied under the bolt head and nut in Fig. 1, the bolt elongates an additional amount while the compressed joint members partially relax. These changes in deformation with external loading are the key to the interaction of forces in bolted joints.

In Fig. 3A the external load F_e is added to the joint diagram Fe is located on the diagram by applying the upper end to an extension of OB and moving it in until the lower end contacts OJ. Since the total amount of elastic deformation (bolt plus joint) remains constant for a given preload, the external load changes the total bolt elongation to $\Delta l_B + \lambda$ and the total joint compression to $\Delta l_J - \lambda$.

In Fig. 3B the external load F_e is divided into an additional bolt load F_{eB} and the joint load F_{eJ}, which unloads the compressed joint members. The maximum bolt load is the sum of the load preload and the additional bolt load:

$$F_{B\,max} = F_i + F_{eB}$$

If the external load Fe is an alternating load, F_{eB} is that part of F_e working as an alternating bolt load, as shown in Fig. 3B. This joint diagram also illustrates that the joint absorbs more of the external load than the bolt subjected to an alternating external load.

The importance of adequate preload is shown in Fig. 3C. Comparing Fig. 3B and Fig. 3C, it can be seen that F_{eB} will remain relatively small as long as the preload F_i is greater than F_{eJ}. Fig. 3C represents a joint with insufficient preload. Under this condition, the amount of external load that the joint can absorb is limited, and the excess load must then be applied to the bolt. If the external load is alternating, the increased stress levels on the bolt produce a greatly shortened fatigue life.

When seating requires a certain minimum force or when transverse loads are to be transformed by friction, the minimum clamping load $F_{J\,min}$ is important.

$$F_{J\,min} = F_{B\,max} - F_e$$

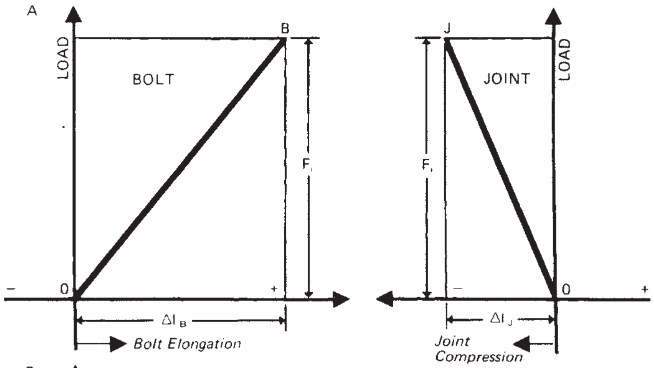


Fig. 1 (above) Joint components

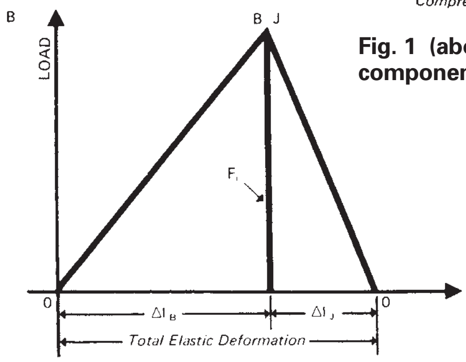
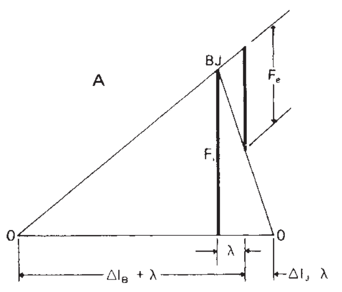


Fig. 2 Joint diagram is obtained by combining load vs. deformation diagrams of bolt and joints.



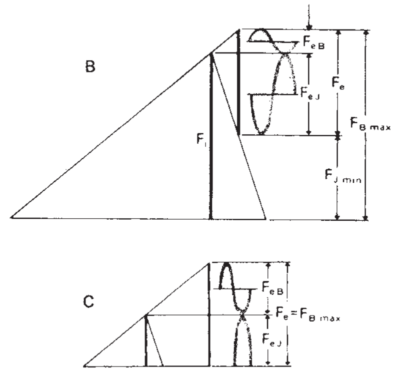


Fig. 3 The complete simple joint diagrams show external load F_e added (A), and external load divided into an additional bolt load F_{eB} and reduction in joint compression F_{eJ} (B). Joint diagram (C) shows how insufficient preload F_i causes excessive additional bolt load F_{eB}.

Spring Constants

To construct a joint diagram, it is necessary to determine the spring rates of both bolt and joint. In general, spring rate is defined as:

$$K = \frac{F}{\Delta l}$$

From Hook's law:

$$\Delta l = \frac{lF}{EA}$$

Therefore:

$$K = \frac{EA}{l}$$

To calculate the spring rate of bolts with different cross sections, the reciprocal spring rates, or compliances, of each section are added:

$$\frac{1}{K_B} = \frac{1}{K_1} + \frac{1}{K_2} + \dots + \frac{1}{K_n}$$

Thus, for the bolt shown in Fig. 4:

$$\frac{1}{K_B} = \frac{1}{E}\left(\frac{0.4d}{A_1} + \frac{l_1}{A_1} + \frac{l_2}{A_2} + \frac{l_3}{A_m} + \frac{0.4d}{A_m}\right)$$

where

d = the minor thread diameter and

A_m = the area of the minor thread diameter

This formula considers the elastic deformation of the head and the engaged thread with a length of 0.4d each.

Calculation of the spring rate of the compressed joint members is more difficult because it is not always obvious which parts of the joint are deformed and which are not. In general, the spring rate of a clamped part is:

$$K_J = \frac{EA_S}{l_J}$$

where A_s is the area of a substitute cylinder to be determined.

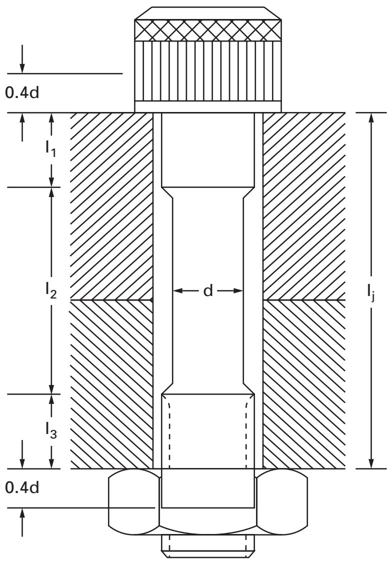


Fig. 4 Analysis of bolt lengths contributing to the bolt spring rate.

When the outside diameter of the joint is smaller than or equal to the bolt head diameter, i.e.,as in a thin bushing, the normal cross sectioned area is computed:

$$A_s = \frac{\pi}{4}(D_c^2 - D_h^2)$$

where

D_c = OD of cylinder or bushing and

D_h = hole diameter

When the outside diameter of the joint is larger than head or washer diameter D_H, the stress distribution is in the shape of a barrel, Fig 5. A series of investigations proved that the areas of the following substitute cylinders are close approximations for calculating the spring contents of concentrically loaded joints.

When the joint diameter D_J is greater than D_H but less than $3D_H$;

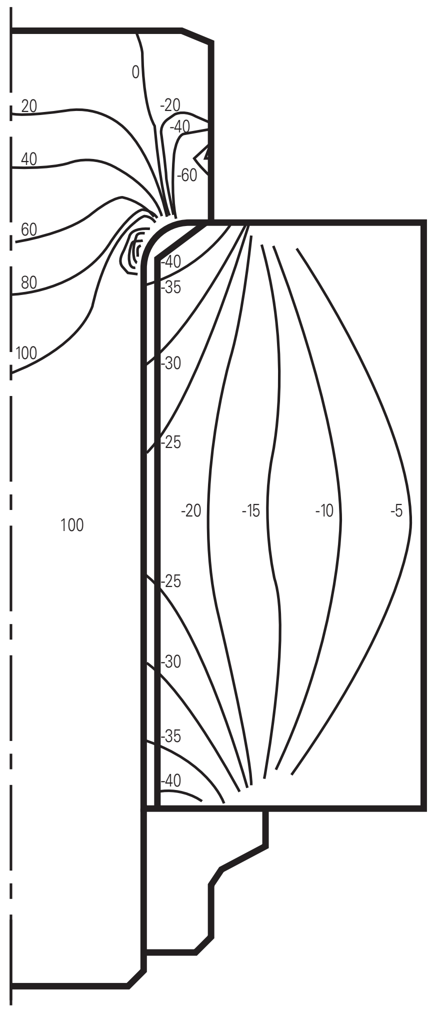


Fig. 5 Lines of equal axial stresses in a bolted joint obtained by the axisymmetric finite element method are shown for a 9/16–18 bolt preloaded to 100 KSI. Positive numbers are tensile stresses in KSI; negative numbers are compressive stresses in KSI.

$$A_s = \frac{\pi}{4}\left(D_H^2 - D_h^2\right) + \frac{\pi}{8}\left(\frac{D_J}{DH} - 1\right)\left(\frac{D_H l_J}{5} + \frac{l_J^2}{100}\right)$$

When the joint diameter D_J is equal to or greater than $3D_H$:

$$A_s = \frac{\pi}{4}\left[\left(D_H + 0.1\, l_J\right)^2 - D_h^2\right]$$

These formulae have been verified in laboratories by finite element method and by experiments.

Fig. 6 shows joint diagrams for springy bolt and stiff joint and for a stiff bolt and springy joint. These diagrams demonstrate the desirability of designing with springy bolt and a stiff joint to obtain a low additional bolt load F_{eB} and thus a low alternating stress.

The Force Ratio

Due to the geometry of the joint diagram, Fig. 7,

$$F_{eB} = \frac{F_e\, K_B}{K_B + K_J}$$

Defining $\Phi = \dfrac{K_B}{K_B + K_J}$

$F_{eB} = F_e\Phi$ and

Φ, called the Force Ratio, $= \dfrac{F_{eB}}{F_e}$

For complete derivation of Φ, see Fig. 7.

To assure adequate fatigue strength of the selected fastener the fatigue stress amplitude of the bolt resulting from an external load F_e is computed as follows:

$$\sigma_B = \pm \frac{F_{eB}/2}{A_m} \quad \text{or}$$

$$\sigma_B = \pm \frac{\Phi\, F_e}{2\, A_m}$$

Effect of Loading Planes

The joint diagram in Fig 3, 6 and 7 is applicable only when the external load F_e is applied at the same loading planes as the preloaded F_i, under the bolt head and the nut. However, this is a rare case, because the external load usually affects the joint somewhere between the center of the joint and the head and the nut.

When a preloaded joint is subjected to an external load F_e at loading planes 2 and 3 in Fig. 8, F_e relieves the compression load of the joint parts between planes 2 and 3. The remainder of the system, the bolt and the joint parts between planes 1-2 and 3-4, feel additional load due to F_e applied planes 2 and 3, the joint material between planes 2 and 3 is the clamped part and all other joint members, fastener and remaining joint material, are clamping parts. Because of the location of the loading planes, the joint diagram changes from black line to the blue line. Consequently, both the additional bolt load $F_{B\,max}$ decrease significantly when the loading planes of F_e shift from under the bolt head and nut toward the joint center.

Determination of the length of the clamped parts is, however, not that simple. First, it is assumed that the external load is applied at a plane perpendicular to the bolt axis. Second, the distance of the loading planes from each other has to be estimated. This distance may be expressed as the ratio of the length of clamped parts to the total joint length. Fig. 9 shows the effect of two different loading planes on the bolt load, both joints having the same preload F_i and the same external load F_e. The lengths of the clamped parts are estimated to be $0.75 l_J$ for joint A, and $0.25 l_J$ for joint B.

In general, the external bolt load is somewhere between $F_{eB} = 1\Phi F_e$ for loading planes under head and nut and $F_{eB} = 0\Phi F_e = 0$ when loading planes are in the joint center, as shown in Fig. 10. To consider the loading planes in calculations, the formula:

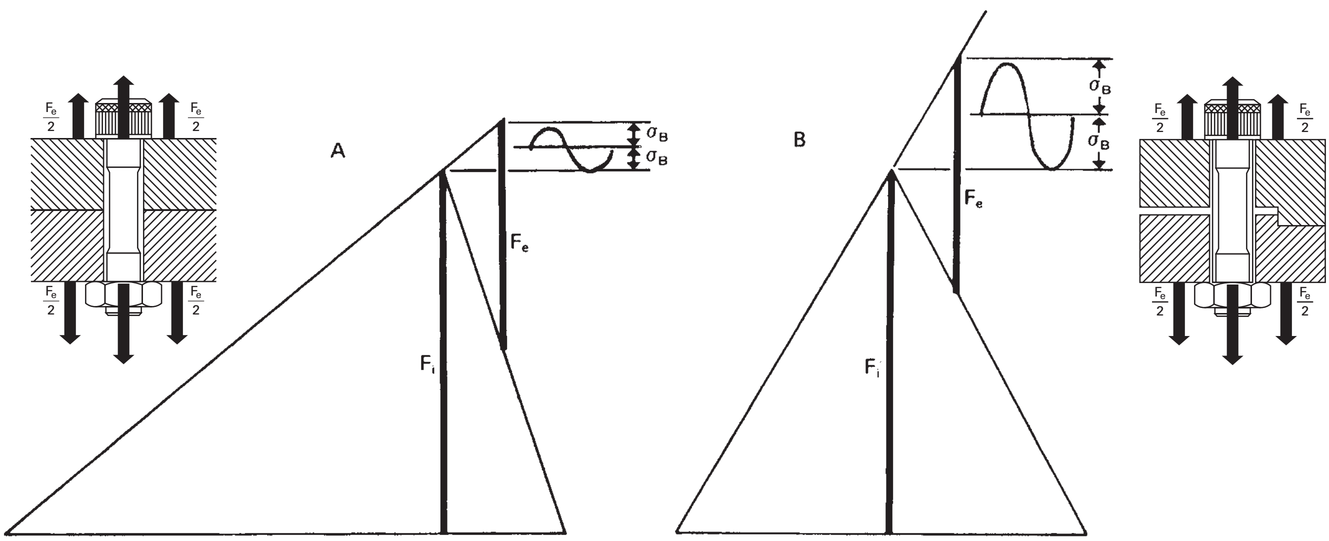


Fig. 6 Joint diagram of a springy bolt in a stiff joint (A), is compared to a diagram of a stiff bolt in a springy joint (B). Preload F_i and external load F_e are the same but diagrams show that alternating bolt stresses are significantly lower with a spring bolt in a stiff joint.

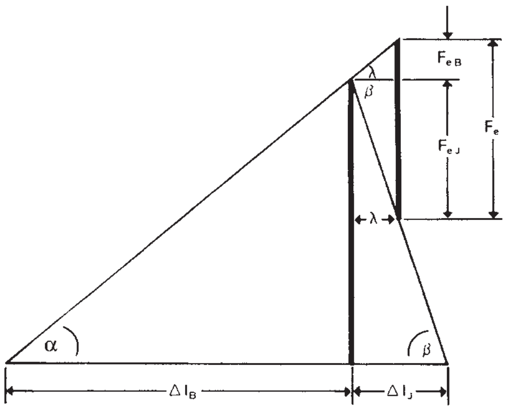


Fig. 7 Analysis of external load F_e and derivation of Force Ratio Φ.

$$\tan\alpha = \frac{F_i}{\Delta l_B} = K_B \text{ and } \tan\beta = \frac{F i}{\Delta l_J} = K_J$$

$$\lambda = \frac{F_{eB}}{\tan\alpha} = \frac{F_{eJ}}{\tan\beta} = \frac{FeB}{K_B} = \frac{F_{eJ}}{K_J} \quad \text{or}$$

$$F_{eJ} = \lambda \tan\beta \text{ and } F_{eB} = \lambda \tan\alpha$$

Since $F_e = F_{eB} + F_{eJ}$
$F_e = F_{eB} + \lambda \tan\beta$

Substituting $\frac{F_{eB}}{\tan\alpha}$ for λ produces:

$$F_e = F_{eB} + \frac{F_{eB}\tan\beta}{\tan\alpha}$$

Multiplying both sides by tan α:

$$F_e \tan\alpha = F_{eB}(\tan\alpha + \tan\beta) \text{ and}$$

$$F_{eB} = \frac{F_e \tan\alpha}{\tan\alpha \tan\beta}$$

Substituting K_B for tan α and K_J for tan β

$$F_{eB} = F_e \frac{F_B}{K_B + K_J}$$

Defining $\Phi = \frac{K_B}{K_B + K_J}$

$$F_{eB} = \Phi F_e$$

$\Phi = \frac{F_{eB}}{F_e}$ and it becomes obvious why Φ is called force ratio.

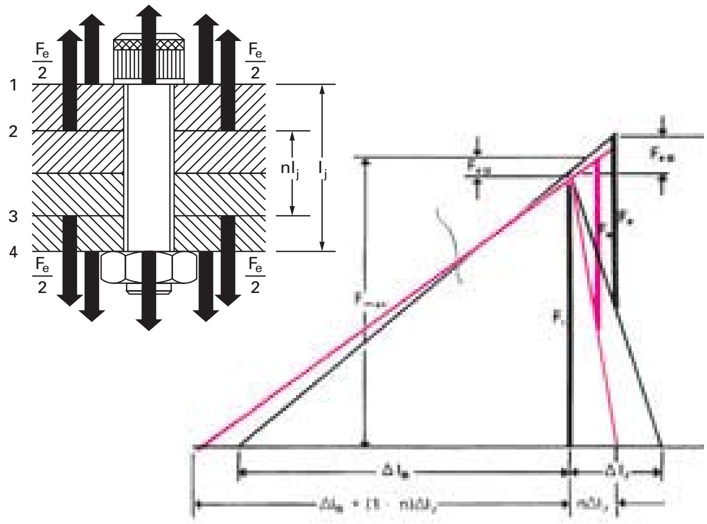


Fig. 8 Joint diagram shows effect of loading planes of F_e on bolt loads F_{eB} and $F_{B\,max}$. Black diagram shows F_{eB} and $F_{B\,max}$ resulting from F_e applied in planes 1 and 4. Orange diagram shows reduced bolt loads when Fe is applied in planes 2 and 3.

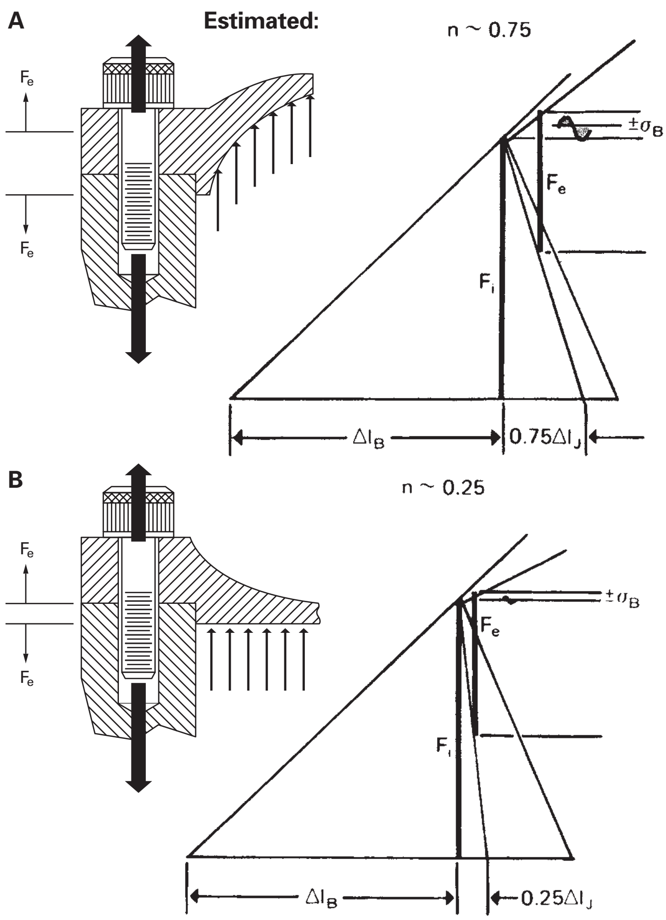


Fig. 9 When external load is applied relatively near bolt head, joint diagram shows resulting alternating stress α_B (A). When same value external load is applied relatively near joint center, lower alternating stress results (B).

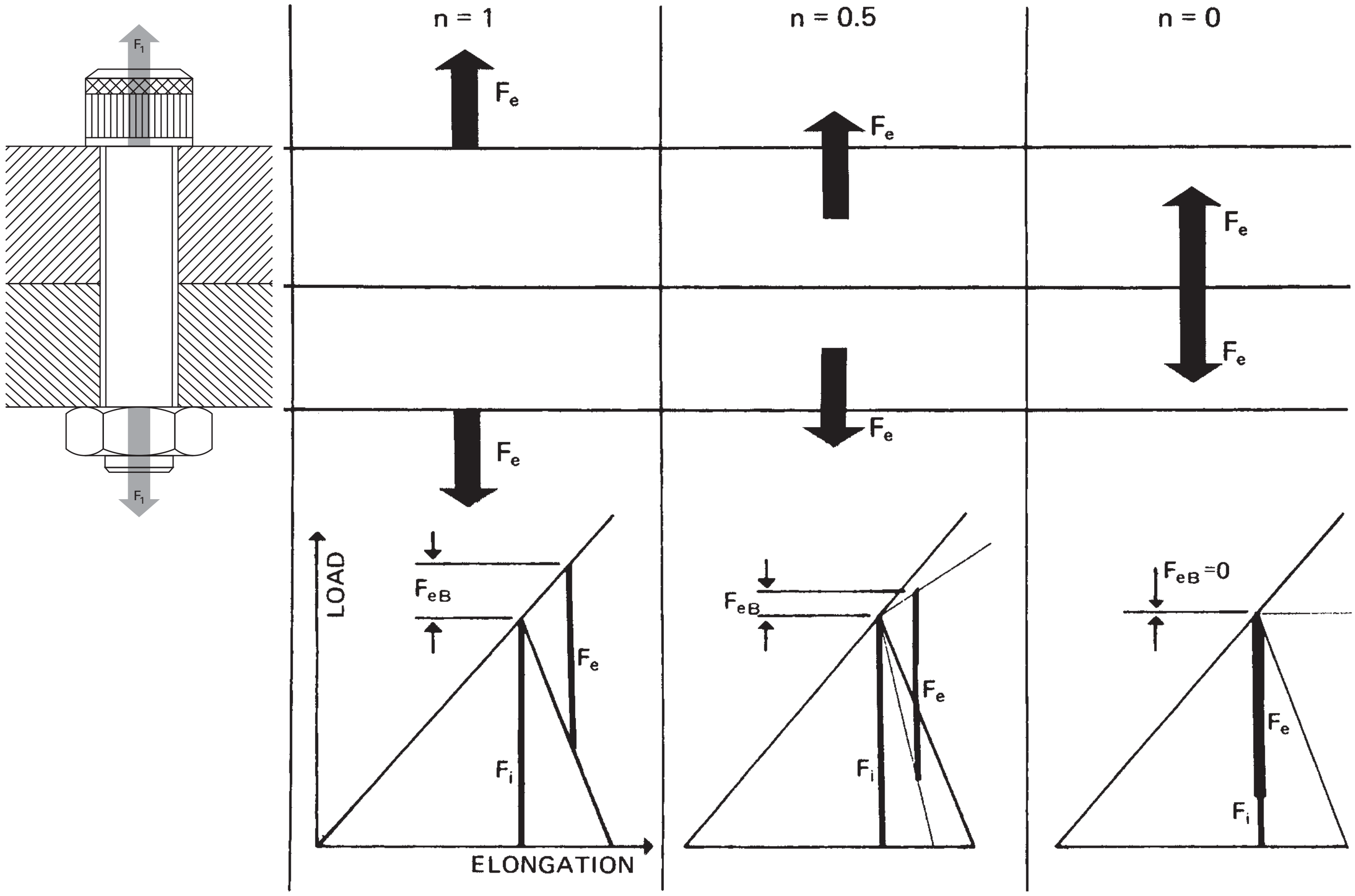


Fig. 10 Force diagrams show the effect of the loading planes of the external load on the bolt load.

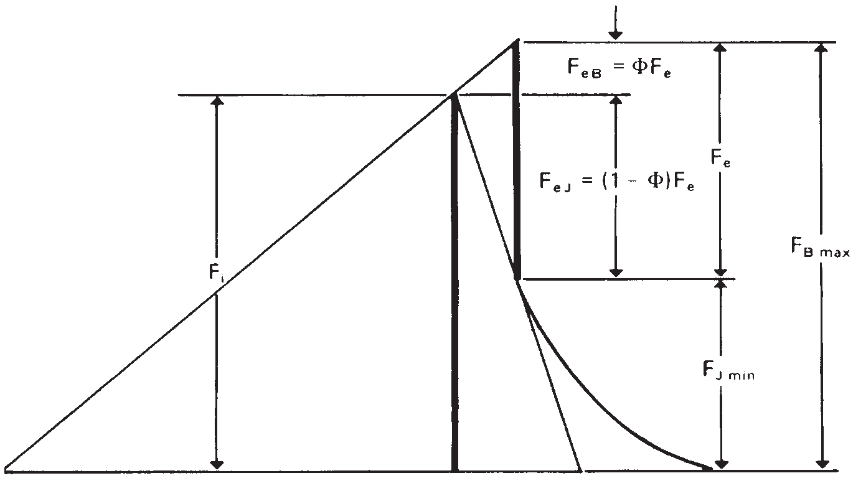


Fig. 11 Modified joint diagram shows nonlinear compression of joint at low preloads.

$F_{eB} = \Phi F_e$ must be modified to :

$F_{eB} = n \Phi F_e$

where n equals the ratio of the length of the clamped parts due to F_e to the joint length l_j. The value of n can range from 1, when F_e is applied under the head and nut, to O, when F_e is applies at the joint center. Consequently the stress amplitude:

$$\sigma_B = \pm \frac{\Phi F_e}{2 A_m} \quad \text{becomes}$$

$$\sigma_B = \pm \frac{n \Phi F_e}{2 A_m}$$

General Design Formulae

Hitherto, construction of the joint diagram has assumed linear resilience of both bolt and joint members. However, recent investigations have shown that this assumption is not quite true for compressed parts.

Taking these investigations into account, the joint diagram is modified to Fig. 11. The lower portion of the joint spring rate is nonlinear, and the length of the linear portion depends on the preload level F_i. The higher F_i the longer the linear portion. By choosing a sufficiently high minimum load, F_{min}>$2F_e$, the non-linear range of the joint spring rate is avoided and a linear relationship between F_{eB} and F_e is maintained.

Also from Fig. 11 this formula is derived:

$$F_{i\,min} = F_{J\,min} + (1 - \Phi) F_e + \Delta F_i$$

where ΔF_i is the amount of preload loss to be expected. For a properly designed joint, a preload loss $\Delta F_i = -$ (0.005 to 0.10) F_i should be expected.

The fluctuation in bolt load that results from tightening is expressed by the ratio:

$$a = \frac{F_{i\,max}}{F_{i\,min}}$$

where a varies between 1.25 and 3.0 depending on the tightening method.

Considering a the general design formulae are:

$$F_{i\,nom} = F_{J\,min} = (1 - \Phi) F_e$$

$$F_{i\,max} = a\,[\,F_{j\,min} + (1 - \Phi) F_e + \Delta F_i\,]$$

$$F_{B\,max} = a\,[\,F_{j\,min} + (1 - \Phi) F_e + \Delta F_i\,] + \Phi F_e$$

Conclusion

The three requirements of concentrically loaded joints that must be met for an integral bolted joint are:

1. The maximum bolt load FB max must be less than the bolt yield strength.

2. If the external load is alternating, the alternating stress must be less than the bolt endurance limit to avoid fatigue failures.

3. The joint will not lose any preload due to permanent set or vibration greater than the value assumed for ΔF_i .

SYMBOLS

A	Area (in.^{2})
Am	Area of minor thread diameter (in.^{2})
A_s	Area of substitute cylinder (in.^{2})
A_x	Area of bolt part 1_x (in.^{2})
d	Diameter of minor thread (in.)
D_c	Outside diameter of bushing (cylinder) (in.)
D_H	Diameter of Bolt head (in.)
D_h	Diameter of hole (in.)
D_J	Diameter of Joint
E	Modulus of Elasticity (psi)
F	Load (lb)
F_e	External load (lb.)
F_{eB}	Additinal Bolt Load due to external load (lb)
F_{eJ}	Reduced Joint load due to external load (lb)
F_i	Preload on Bolt and Joint (lb)
ΔF_i	Preload loss (–lb)
$F_{i\,min}$	Minimum preload (lb)
$F_{i\,max}$	Maximum preload (lb)
$F_{j\,nom}$	Nominal preload (lb)
$F_{B\,max}$	Maximum Bolt load (lb)
$F_{J\,min}$	Minimum Joint load (lb)
K	Spring rate (lb/in.)
K_B	Spring rate of Bolt (lb/in.)
K_J	Spring rate of Joint (lb/in.)
K_x	Spring rate of Bolt part l_x (lb/in.)
l	Length (in.)
Δl	Change in length (in.)
l_B	Length of Bolt (in.)
Δl_B	Bolt elongation due to F_i (in.)
l_J	Length of Joint (in.)
Δl_J	Joint compression to F_i (in.)
l_x	Length of Bolt part x (in.)
n	$\frac{\text{Length of clamped parts}}{\text{Total Joint Length}}$
α	Tightening factor
Φ	Force ratio
λ	Bolt and Joint elongation due to F_e (in.)
σ_B	Bolt stress amplitude (± psi)

TIGHTENING TORQUES AND THE TORQUE-TENSION RELATIONSHIP

All of the analysis and design work done in advance will have little meaning if the proper preload is not achieved. Several discussions in this technical section stress the importance of preload to maintaining joint integrity. There are many methods for measuring preload (see Table 12). However, one of the least expensive techniques that provides a reasonable level of accuracy versus cost is by measuring torque. The fundamental characteristic required is to know the relationship between torque and tension for any particular bolted joint. Once the desired design preload must be identified and specified first, *then* the torque required to induce that preload is determined.

Within the elastic range, before permanent stretch is induced, the relationship between torque and tension is essentially linear (see figure 13). Some studies have found up to 75 variables have an effect on this relationship: materials, temperature, rate of installation, thread helix angle, coefficients of friction, etc. One way that has been developed to reduce the complexity is to depend on empirical test results. That is, to perform experiments under the application conditions by measuring the induced torque and recording the resulting tension. This can be done with relatively simple, calibrated hydraulic pressure sensors, electric strain gages, or piezoelectric load cells. Once the data is gathered and plotted on a chart, the slope of the curve can be used to calculate a correlation factor. This technique has created an accepted formula for relating torque to tension.

$T = K \times D \times P$

T = torque, lbf.-in.
D = fastener nominal diameter, inches
P = preload, lbf.
K = “nut factor,” “tightening factor,” or “k-value”

If the preload and fastener diameter are selected in the design process, and the K-value for the application conditions is known, then the necessary torque can be calculated. It is noted that even with a specified torque, actual conditions at the time of installation can result in variations in the actual preload achieved (see Table 12).

One of the most critical criteria is the selection of the K-value. Accepted nominal values for many industrial applications are:

K = 0.20 for as-received steel bolts into steel holes

K = 0.15 steel bolts with cadmium plating, which acts like a lubricant,

K = 0.28 steel bolts with zinc plating.

The K-value is not the coefficient of the friction (μ); it is an empirically derived correlation factor.

It is readily apparent that if the torque intended for a zinc plated fastener is used for cadmium plated fastener, the preload will be almost two times that intended; it may actually cause the bolt to break.

Another influence is where friction occurs. For steel bolts holes, approximately 50% of the installation torque is consumed by friction under the head, 35% by thread friction, and only the remaining 15% inducing preload tension. Therefore, if lubricant is applied just on the fastener underhead, full friction reduction will not be achieved. Similarly, if the material against which the fastener is bearing, e.g. aluminum, is different than the internal thread material, e.g. cast iron, the effective friction may be difficult to predict, These examples illustrate the importance and the value of identifying the torque-tension relationship. It is a recommend practice to contact the lubricant manufacturer for K-value information if a lubricant will be used.

The recommended seating torques for Unbrako headed socket screws are based on inducing preloads reasonably expected in practice for each type. The values for Unbrako metric fasteners are calculated using VDI2230, a complex method utilized extensively in Europe. All values assume use in the received condition in steel holes. It is understandable the designer may need preloads higher than those listed. The following discussion is presented for those cases.

TORSION-TENSION YIELD AND TENSION CAPABILITY AFTER TORQUING

Once a headed fastener has been seated against a bearing surface, the inducement of torque will be translated into both torsion and tension stresses. These stresses combine to induce twist. If torque continues to be induced, the stress along the angle of twist will be the largest stress *while the bolt is being torqued.* Consequently, the stress along the bolt axis (axial tension) will be something less. This is why a bolt can fail at a lower tensile stress *during installation* than when it is pulled in straight tension alone, eg . a tensile test. Research has indicated the axial tension can range from 135,000 to 145,000 PSI for industry socket head cap screws at torsion-tension yield, depending on diameter. Including the preload variation that can occur with various installation techniques, eg. up to 25%, it can be understood why some recommended torques induce preload reasonably lower than the yield point.

Figure 13 also illustrates the effect of straight tension applied after installation has stopped. Immediately after stopping the installation procedure there will be some relaxation, and the torsion component will drop toward zero. This leaves only the axial tension, which keeps the joint clamped together. Once the torsion is relieved, the axial tension yield value and ultimate value for the fastener will be appropriate.

Table 12
Industrial Fasteners Institute's Torque-Measuring Method

Preload Measuring Method	Accuracy Percent	Relative Cost
Feel (operator's judgement)	±35	1
Torque wrench	±25	1.5
Turn of the nut	±15	3
Load-indicating washers	±10	7
Fastener elongation	±3 to 5	15
Strain gages	±1	20

THE TORQUE-TENSION RELATIONSHIP

Fig. 14

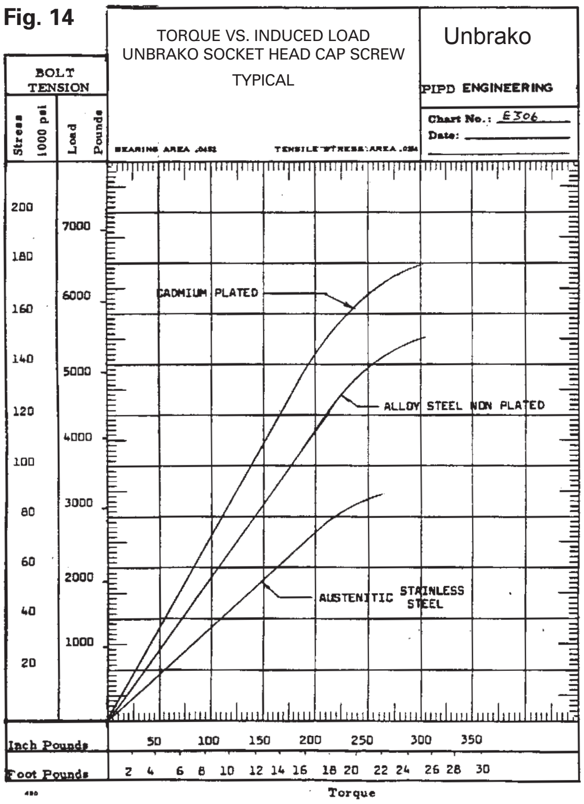


Fig. 13 Torque/Tension Relationship

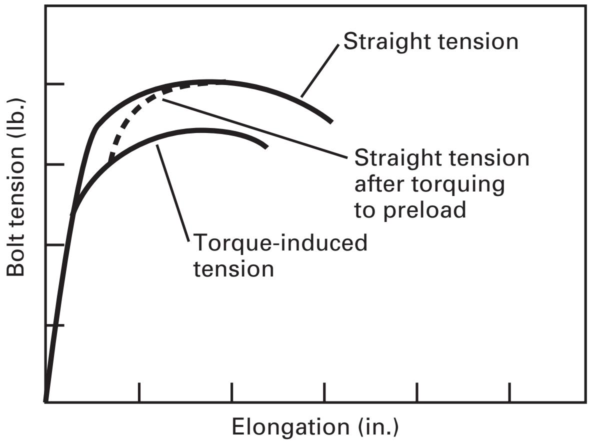


Fig. 15 Recommended Seating Torques (Inch-Lb.) for Application in Various Materials
UNBRAKO pHd (1960 Series) Socket Head Cap Screws

	mild steel Rb 87 cast iron Rb 83 note 1		brass Rb 72 note 2		aluminum Rb 72 (2024-T4) note 3	
	UNC	UNF	UNC	UNF	UNC	UNF
screw size	plain	plain	plain	plain	plain	plain
#0	–	*2.1	–	*2.1	–	*2.1
#1	*3.8	*4.1	*3.8	*4.1	*3.8	*4.1
#2	*6.3	*6.8	*6.3	*6.8	*6.3	*6.8
#3	*9.6	*10.3	*9.6	*10.3	*9.6	*10.3
#4	*13.5	*14.8	*13.5	*14.8	*13.5	*14.8
#5	*20	*21	*20	*21	*20	*21
#6	*25	*28	*25	*28	*25	*28
#8	*46	*48	*46	*48	*46	*48
#10	*67	*76	*67	*76	*67	*76
1/4	*158	*180	136	136	113	113
5/16	*326	*360	228	228	190	190
3/8	*580	635	476	476	397	397
7/16	*930	*1,040	680	680	570	570
1/2	*1,420	*1,590	1,230	1,230	1,030	1,030
9/16	*2,040	2,250	1,690	1,690	1,410	1,410
5/8	*2,820	3,120	2,340	2,340	1,950	1,950
3/4	*5,000	5,340	4,000	4,000	3,340	3,340
7/8	*8,060	8,370	6,280	6,280	5,230	5,230
1	*12,100	12,800	9,600	9,600	8,000	8,000
1 1/8	*13,800	*15,400	13,700	13,700	11,400	11,400
1 1/4	*19,200	*21,600	18,900	18,900	15,800	15,800
1 3/8	*25,200	*28,800	24,200	24,200	20,100	20,100
1 1/2	*33,600	*36,100	32,900	32,900	27,400	27,400

NOTES:
1. Torques based on 80,000 psi bearing stress under head of screw.
2. Torques based on 60,000 psi bearing stress under head of screw.
3. Torques based on 50,000 psi bearing stress under head of screw.

*Denotes torques based on 100,000 psi tensile stress in screw threads up to 1" dia., and 80,000 psi for sizes 1 1/8" dia. and larger.

To convert inch-pounds to inch-ounces – multiply by 16.

To convert inch-pounds to foot-pounds – divide by 12.

STRIPPING STRENGTH OF TAPPED HOLES

Charts and sample problems for obtaining minimum thread engagement based on applied load, material, type of thread and bolt diameter.

Knowledge of the thread stripping strength of tapped holes is necessary to develop full tensile strength of the bolt or, for that matter, the minimum engagement needed for any lesser load.

Conversely, if only limited length of engagement is available, the data help determine the maximum load that can be safely applied without stripping the threads of the tapped hole.

Attempts to compute lengths of engagement and related factors by formula have not been entirely satisfactory-mainly because of subtle differences between various materials. Therefore, strength data has been empirically developed from a series of tensile tests of tapped specimens for seven commonly used metals including steel, aluminum, brass and cast iron.

The design data is summarized in the six accompanying charts, (Charts E504-E509), and covers a range of screw thread sizes from #0 to one inch in diameter for both coarse and fine threads. Though developed from tests of Unbrako socket head cap screws having minimum ultimate tensile strengths (depending on the diameter) from 190,000 to 180,000 psi , these stripping strength values are valid for all other screws or bolts of equal or lower strength having a standard thread form. Data are based on static loading only.

In the test program, bolts threaded into tapped specimens of the metal under study were stressed in tension until the threads stripped. Load at which stripping occurred and the length of engagement of the specimen were noted. Conditions of the tests, all of which are met in a majority of industrial bolt applications, were:

- Tapped holes had a basic thread depth within the range of 65 to 80 per cent. Threads of tapped holes were Class 2B fit or better.
- Minimum amount of metal surrounding the tapped hole was 2 1/2 times the major diameter.
- Test loads were applied slowly in tension to screws having standard Class 3A threads. (Data, though, will be equally applicable to Class 2A external threads as well.)
- Study of the test results revealed certain factors that greatly simplified the compilation of thread stripping strength data:
- Stripping strengths are almost identical for loads applied either by pure tension or by screw torsion. Thus data are equally valid for either condition of application.
- Stripping strength values vary with diameter of screw. For a given load and material, larger diameter bolts required greater engagement.
- Minimum length of engagement (as a percent of screw diameter) is a straight line function of load. This permits easy interpolation of test data for any intermediate load condition.
- When engagement is plotted as a percentage of bolt diameter, it is apparent that stripping strengths for a wide range of screw sizes are close enough to be grouped in a single curve. Thus , in the accompanying charts, data for sizes #0 through #12 have been represented by a single set of curves.

With these curves, it becomes a simple matter to determine stripping strengths and lengths of engagement for any condition of application. A few examples are given below:

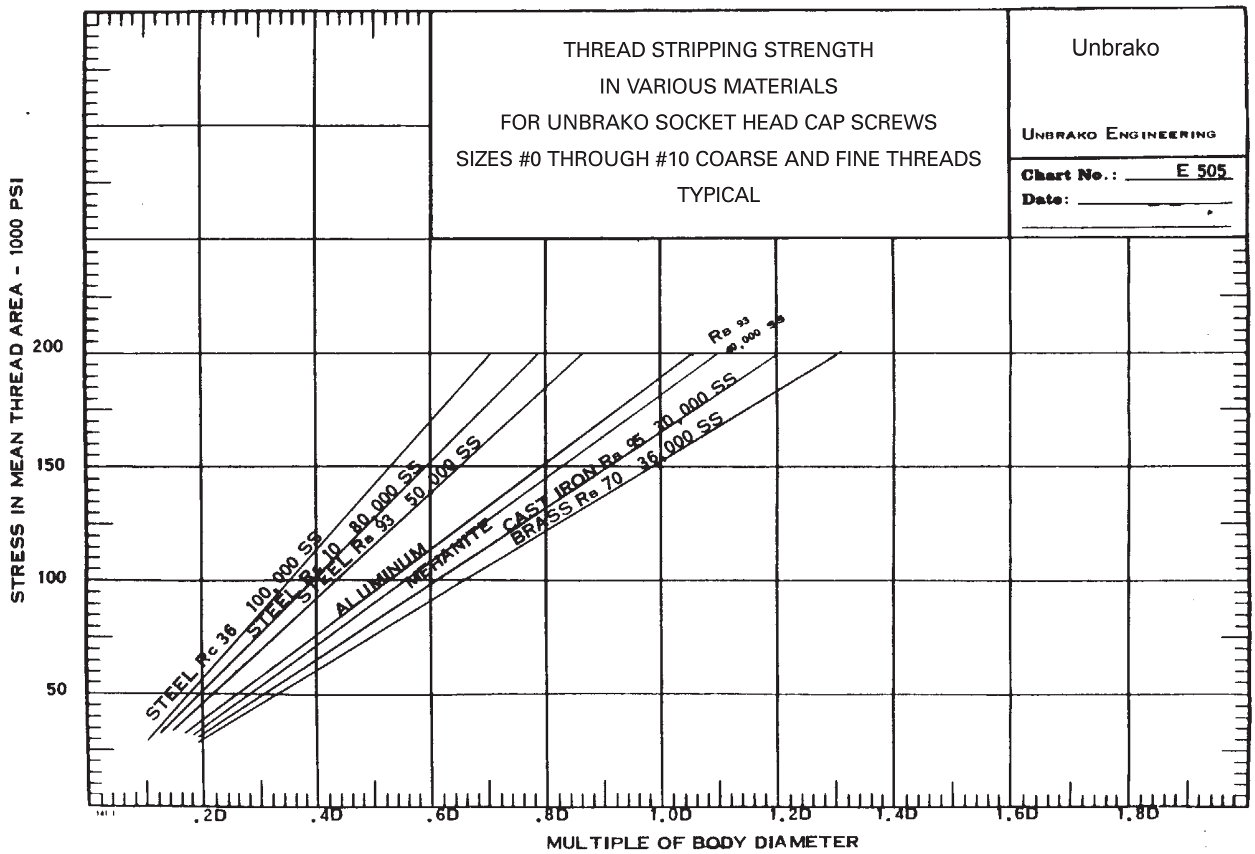
Example 1. Calculate length of thread engagement necessary to develop the minimum ultimate tensile strength (190,000 psi) of a 1/2–13 (National Coarse) Unbrako cap screw in cast iron having an ultimate shear strength of 30,000 psi. E505 is for screw sizes from #0 through #10; E506 and E507 for sizes from 1/4 in. through 5/8 in.; E508 and E509 for sizes from 3/4 in. through 1 in. Using E506 a value 1.40D is obtained. Multiplying nominal bolt diameter (0.500 in.) by 1.40 gives a minimum length of engagement of 0.700 in.

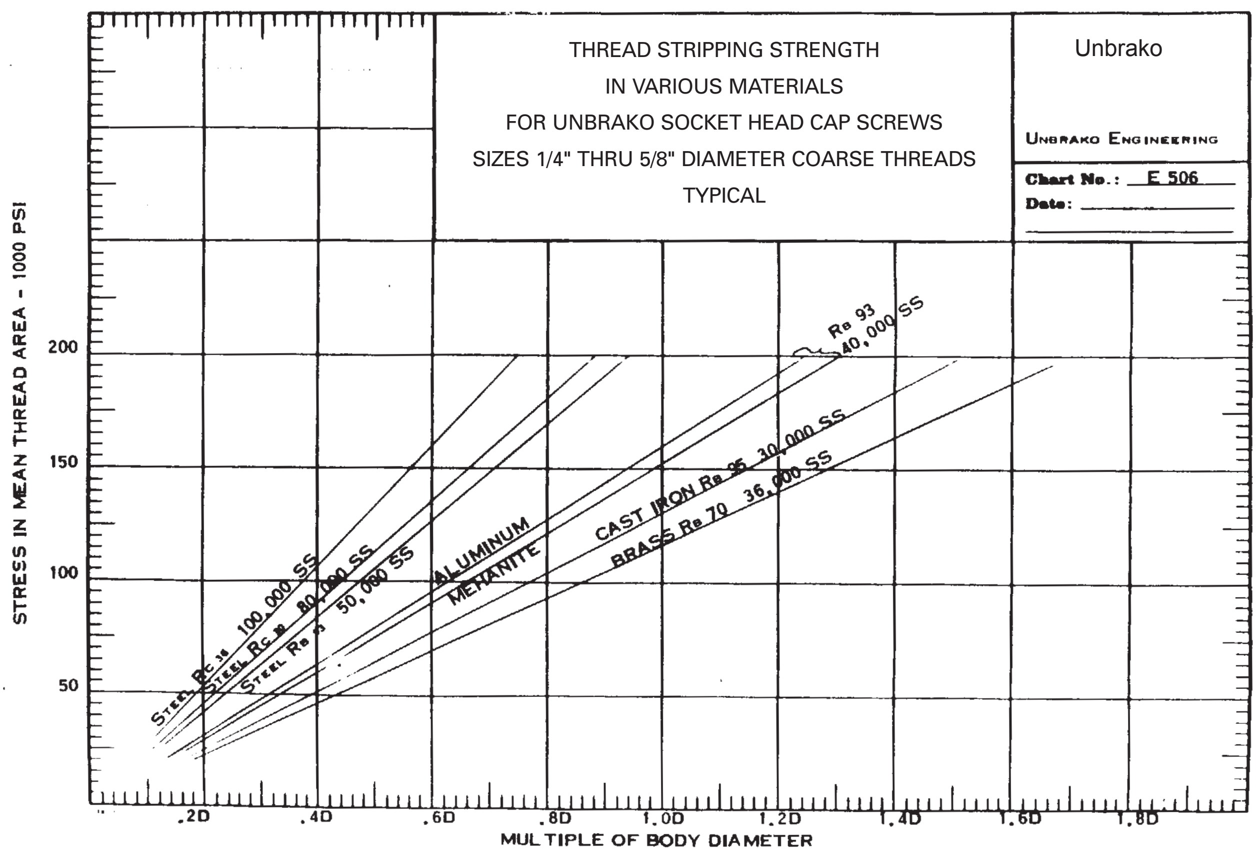
Example 2. Calculate the length of engagement for the above conditions if only 140,000 psi is to be applied. (This is the same as using a bolt with a maximum tensile strength of 140,000psi.) From E506 obtain value of 1.06D Minimum length of engagement = (0.500) (1.06) = 0.530.

Example 3. Suppose in Example 1 that minimum length of engagement to develop full tensile strength was not available because the thickness of metal allowed a tapped hole of only 0.600 in. Hole depth in terms of bolt dia. = 0.600/0.500 = 1.20D. By working backwards in Chart E506, maximum load that can be carried is approximately 159,000 psi.

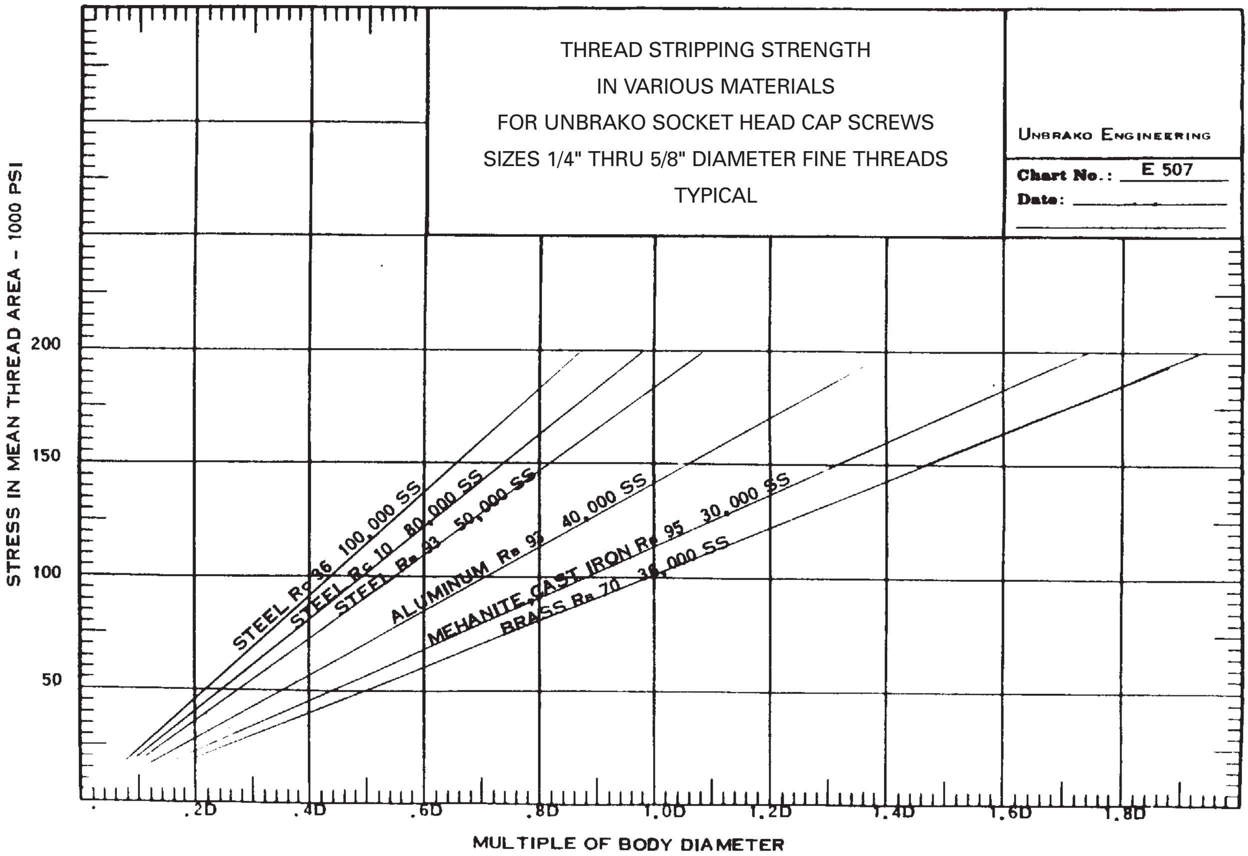
Example 4. Suppose that the hole in Example 1 is to be tapped in steel having an ultimate shear strength 65,000 psi. There is no curve for this steel in E506 but a design value can be obtained by taking a point midway between curves for the 80,000 psi and 50,000 psi steels that are listed. Under the conditions of the example, a length of engagement of 0.825D or 0.413 in. will be obtained.

STRIPPING STRENGTH OF TAPPED HOLES

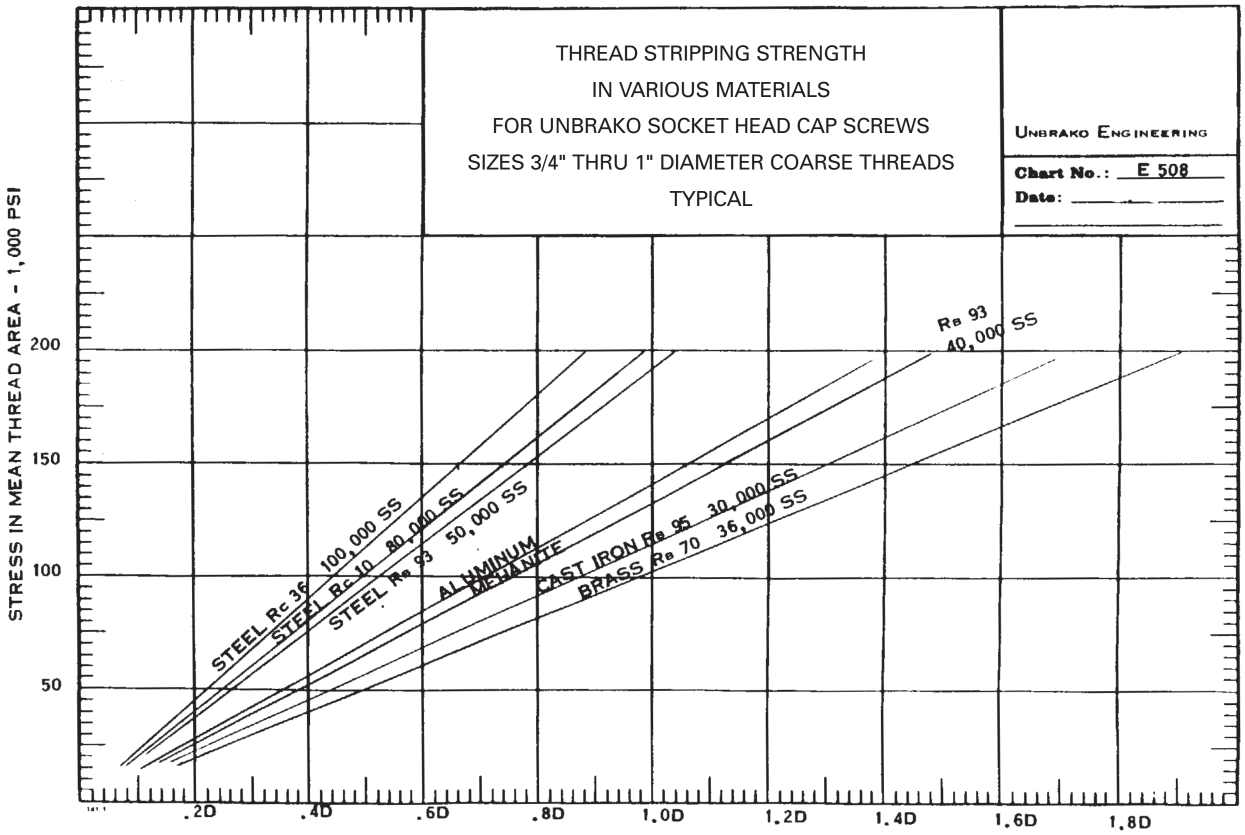




Unbrako

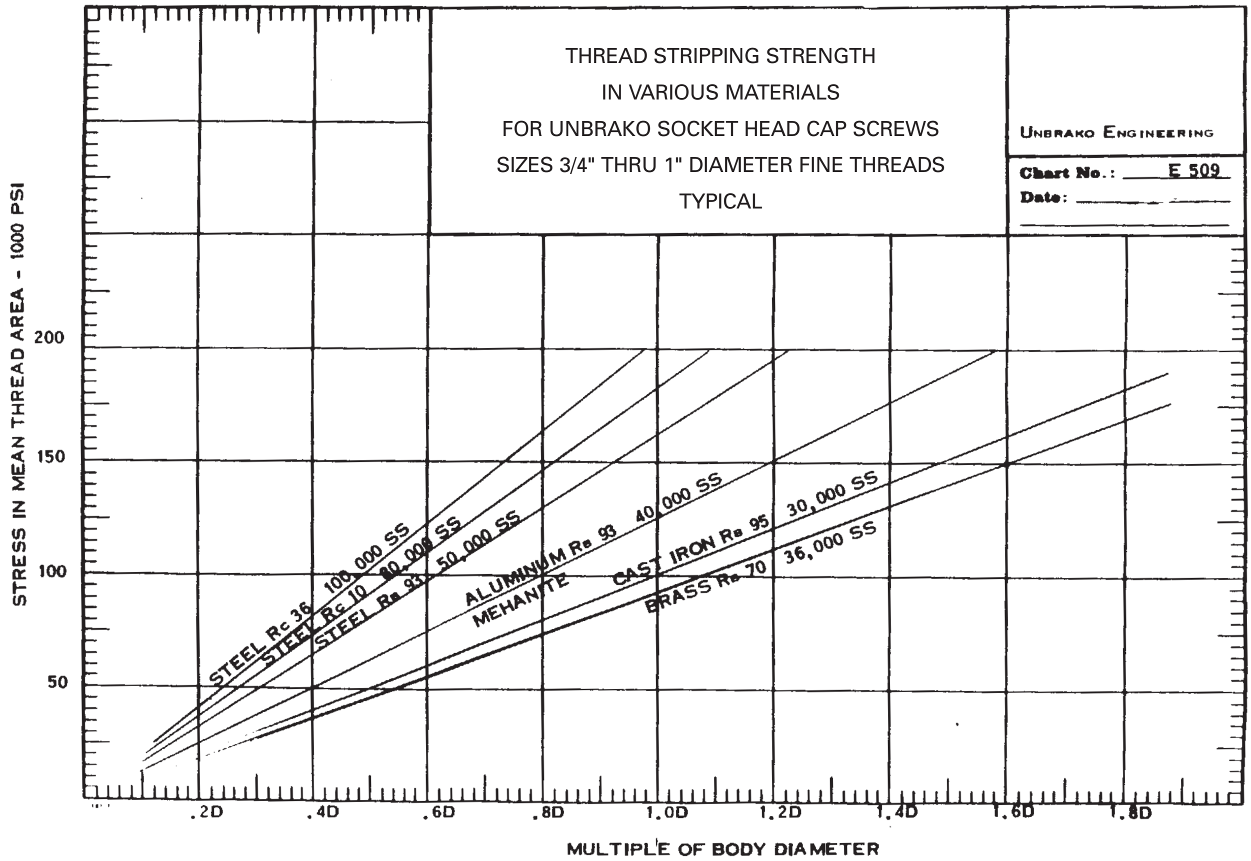


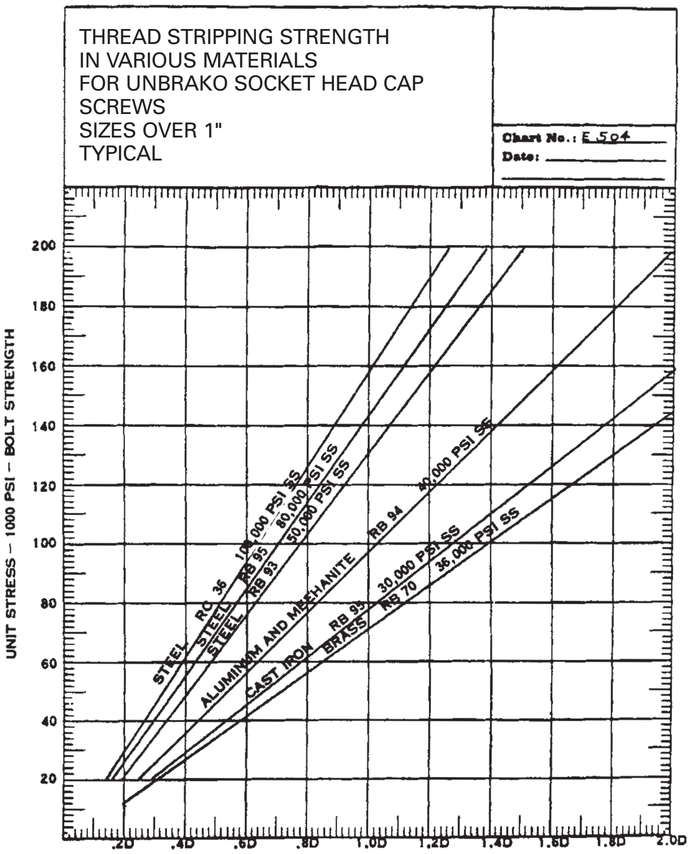
THREAD STRIPPING STRENGTH
IN VARIOUS MATERIALS
FOR UNBRAKO SOCKET HEAD CAP SCREWS
SIZES 1/4" THRU 5/8" DIAMETER FINE THREADS
TYPICAL
UNBRAKO ENGINEERING
Chart No.: E 507
Date:
STRESS IN MEAN THREAD AREA - 1000 PSI
200
150
100
50
STEEL Rc 36 100,000 SS
STEEL Rc 10 80,000 SS
STEEL Rs 93 50,000 SS
ALUMINUM Rs 92 40,000 SS
MEHANITE CAST IRON Rs 95 30,000 SS
BRASS Rs 70 36,000 SS
.2D
.4D
.6D
.8D
1.0D
1.2D
1.4D
1.6D
1.8D
MULTIPLE OF BODY DIAMETER



THREAD STRIPPING STRENGTH
IN VARIOUS MATERIALS
FOR UNBRAKO SOCKET HEAD CAP SCREWS
SIZES 3/4" THRU 1" DIAMETER COARSE THREADS
TYPICAL
UNBRAKO ENGINEERING
Chart No.: E 508
Date:
STRESS IN MEAN THREAD AREA - 1,000 PSI
200
150
100
50
Rs 93
40,000 SS
STEEL Rc 36 100,000 SS
STEEL Rc 10 80,000 SS
STEEL Rs 93 50,000 SS
ALUMINUM
MEHANITE
CAST IRON Rs 95 30,000 SS
BRASS Rs 70 36,000 SS
.2D
.4D
.6D
.8D
1.0D
1.2D
1.4D
1.6D
1.8D

STRIPPING STRENGTH OF TAPPED HOLES





HIGH-TEMPERATURE JOINTS

Bolted joints subjected to cyclic loading perform best if an initial preload is applied. The induced stress minimizes the external load sensed by the bolt, and reduces the chance of fatigue failure. At high temperature, the induced load will change, and this can adeversely affect the fastener performance. It is therefore necessary to compensate for high-temperature conditions when assembling the joint at room temperature. This article describes the factors which must be considered and illustrates how a high-temperature bolted joint is designed.

In high-temperature joints, adequate clamping force or preload must be maintained in spite of temperature-induced dimensional changes of the fastener relative to the joint members. the change in preload at any given temperature for a given time can be calculated, and the affect compensated for by proper fastener selection and initial preload.

Three principal factors tend to alter the initial clamping force in a joint at elevated temperatures, provided that the fastener material retains requisite strength at the elevated temperature. These factors are: Modulus of elasticity, coefficient of thermal expansion, and relaxation.

Modulus Of Elasticity: As temperature increases, less stress or load is needed to impart a given amount of elongation or strain to a material than at lower temperatures. This means that a fastener stretched a certain amount at room temperature to develop a given preload will exert a lower clamping force at higher temperature if there is no change in bolt elongation.

Coefficient of Expansion: With most materials, the size of the part increases as the temperature increases. In a joint, both the structure and the fastener grow with an increase in temperature, and this can result,depending on the materials, in an increase or decrease in the clamping force. Thus, matching of materials in joint design can assure sufficient clamping force at both room and elevated temperatures. Table 16 lists mean coefficient of thermal expansion of certain fastener alloys at several temperatures.

Relaxation: At elevated temperatures, a material subjected to constant stress below its yield strength will flow plastically and permanently change size. This phenomenon is called creep. In a joint at elevated temperature, a fastener with a fixed distance between the bearing surface of the head and nut will produce less and less clamping force with time. This characteristic is called relaxation. It differs from creep in that stress changes while elongation or strain remains constant. Such elements as material, temperature, initial stress, manufacturing method, and design affect the rate of relaxation.

Relaxation is the most important of the three factors. It is also the most critical consideration in design of elevated-temperature fasteners. A bolted joint at 1200°F can lose as much as 35 per cent of preload. Failure to compensate for this could lead to fatigue failure through a loose joint even though the bolt was properly tightened initially.

If the coefficient of expansion of the bolt is greater than that of the joined material, a predictable amount of clamping force will be lost as temperature increases. Conversely, if the coefficient of the joined material is greater, the bolt may be stressed beyond its yield or even fracture strength. Or, cyclic thermal stressing may lead to thermal fatigue failure.

Changes in the modulus of elasticity of metals with increasing temperature must be anticipated, calculated, and compensated for in joint design. Unlike the coefficient of expansion, the effect of change in modulus is to reduce clamping force whether or not bolt and structure are the same material, and is strictly a function of the bolt metal.

Since the temperature environment and the materials of the structure are normally "fixed," the design objective is to select a bolt material that will give the desired clamping force at all critical points in the operating range of the joint. To do this, it is necessary to balance out the three factors-relaxation, thermal expansion, and modulus-with a fourth, the amount of initial tightening or clamping force.

In actual joint design the determination of clamping force must be considered with other design factors such as ultimate tensile, shear, and fatigue strength of the fastener at elevated temperature. As temperature increases the inherent strength of the material decreases. Therefore, it is important to select a fastener material which has sufficient strength at maximum service temperature.

Example

The design approach to the problem of maintaining satisfactory elevated-temperature clamping force in a joint can be illustrated by an example. The example chosen is complex but typical. A cut-and-try process is used to select the right bolt material and size for a given design load under a fixed set of operating loads and environmental conditions, Fig.17.

The first step is to determine the change in thickness, Δt, of the structure from room to maximum operating temperature.

For the AISI 4340 material:

$\Delta t_1 = t_1(T_2 - T_1)\alpha$

$\Delta t_1 = (0.50)(800 - 70)\ (7.4 \times 10^{-6})$

$\Delta t_1 = 0.002701$ in.

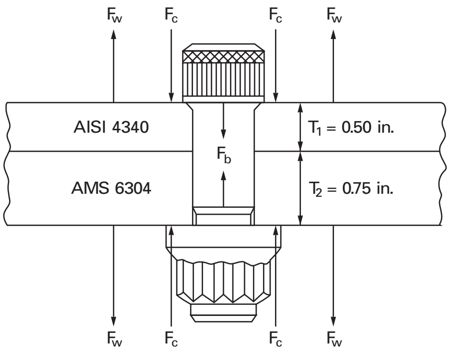
For the AMS 6304 material:

$\Delta t_2 = (0.75)(800 - 70)(7.6 \times 10^{-6})$

$\Delta t_2 = 0.004161$ in.

The total increase in thickness for the joint members is 0.00686 in.

The total effective bolt length equals the total joint thickness plus one-third of the threads engaged by the nut. If it is assumed that the smallest diameter bolt should be used for weight saving, then a 1/4-in. bolt should be tried. Thread engagement is approximately one diameter, and the effective bolt length is:



- d = Bolt diam, in.
- E = Modulus of elasticity, psi
- F_b = Bolt preload, lb
- F_c = Clamping force, lb (F_b=F_c)
- F_w = Working load=1500 lb static + 100 lb cyclic
- L = Effective bolt length, inc.
- T_1 = Room temperature= 70°F
- T_2 = Maximum operatng temperature for 1000 hr=800°F
- t = Panel thickness, in.
- a = Coefficient of thermal expansion

Fig. 17 – Parameters for joint operating at 800°F.

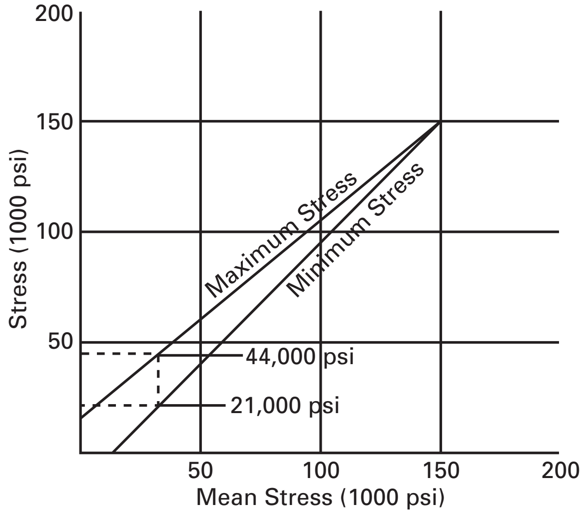


Fig. 18 – Goodman diagram of maximum and minimum operating limits for H-11 fastener at 800°F. Bolts stressed within these limits will give infinite fatigue life.

$L = t_1 + t_2 + (1/3\ d)$

$L = 0.50 + 0.75 + (1/3 \times 0.25)$

$L = 1.333$ in.

The ideal coefficient of thermal expansion of the bolt material is found by dividing the total change in joint thickness by the bolt length times the change in temperature.

$$\alpha b = \frac{\Delta t}{L \times \Delta t}$$

$$\alpha = \frac{.00686}{(1.333)(800 - 70)} = 7.05 \times 10^{-6} \text{ in./in./deg. F}$$

The material, with the nearest coefficient of expansion is with a value of 9,600,000 at 800°F.

To determine if the bolt material has sufficient strength and resistance to fatigue, it is necessary to calculate the stress in the fastener at maximum and minimum load. The bolt load plus the cyclic load divided by the tensile stress of the threads will give the maximum stress. For a 1/4-28 bolt, tensile stress area,from thread handbook H 28, is 0.03637 sq. in. The maximum stress is

$$S_{max} = \frac{\text{Bolt load}}{\text{Stress area}} = \frac{1500 + 100}{0.03637}$$

$S_{max} = 44{,}000$ psi

and the minimum bolt stress is 41,200 psi.

H-11 has a yield strength of 175,000 psi at 800°F, Table 3, and therefore should be adequate for the working loads.

A Goodman diagram, Fig. 18, shows the extremes of stress within which the H-11 fastener will not fail by fatigue. At the maximum calculated load of 44,000 psi, the fastener will withstand a minimum cyclic loading at 800°F of about 21,000 psi without fatigue failure.

Because of relaxation, it is necessary to determine the initial preload required to insure 1500-lb. clamping force in the joint after 1000 hr at 800°F.

When relaxation is considered, it is necessary to calculate the maximum stress to which the fastener is subjected. Because this stress is not constant in dynamic joints, the resultant values tend to be conservative. Therefore, a maximum stress of 44,000 psi should be considered although the necessary stress at 800°F need be only 41,200 psi. Relaxation at 44,000 psi can be interpolated from the figure, although an actual curve could be constructed from tests made on the fastener at the specific conditions.

The initial stress required to insure a clamping stress of 44,000 psi after 1000 hr at 800°F can be calculated by interpolation.

$x = 61{,}000 - 44{,}000 = 17{,}000$
$y = 61{,}000 - 34{,}000 = 27{,}000$
$B = 80{,}000 - 50{,}000 = 30{,}000$
$A = 80{,}000 - C$

$$\frac{x}{y} = \frac{A}{B} \quad \frac{17{,}000}{27{,}000} = \frac{80{,}000 - C}{30{,}000}$$

$C = 61{,}100$ psi

The bolt elongation required at this temperature is calculated by dividing the stress by the modulus at temperature and multiplying by the effective length of the bolt. That is: $(61{,}000 \times 1.333)/24.6 \times 10^6 = 0.0033$

Since the joint must be constructed at room temperature, it is necessary to determine the stresses at this state. Because the modulus of the fastener material changes with temperature, the clamping force at room temperature will not be the same as at 800°F. To determine the clamping stress at assembly conditions, the elongation should be multiplied by the modulus of elasticity at room temperature.

$.0033 \times 30.6 \times 10^6 = 101{,}145$ psi

The assembly conditions will be affected by the difference between th ideal and actual coefficients of expansion of the joint. The ideal coeffienct for the fastener material was calculated to be 7.05 but the closest material – H-11 – has a coefficient of 7.1. Since this material has a greater expansion than calculated, there will be a reduction in clamping force resulting from the increase in temperature. This amount equals the difference between the ideal and the actual coefficients multiplied by the change in temperature, the length of the fastener, and the modulus of elasticity at 70°F.

$[(7.1 - 7.05) \times 10^{-6}][800 - 70][1.333] \times$

$[30.6 \times 10^6] = 1{,}490$ psi

The result must be added to the initial calculated stresses to establish the minimum required clamping stress needed for assembling the joint at room temperature.

$101{,}145 + 1{,}490 = 102{,}635$ psi

Finally, the method of determining the clamping force or preload will affect the final stress in the joint at operating conditions. For example, if a torque wrench is used to apply preload (the most common and simplest method available), a plus or minus 25 per cent variation in induced load can result. Therefore, the maximum load which could be expected in this case would be 1.5 times the minimum, or:

$(1.5)(102{,}635) = 153{,}950$ psi

This value does not exceed the room-temperature yield strength for H-11 given in Table 19.

Since there is a decrease in the clamping force with an increase in temperature and since the stress at operating temperature can be higher than originally calculated because of variations in induced load, it is necessary to ascertain if yield strength at 800°F will be exceeded

$$\frac{(\text{max stress at } 70°\text{F} + \text{change in stress}) \times \text{E at } 800°\text{F}}{E \text{ at } 70°\text{F}}$$

$$\frac{[153{,}950 + (-1490)] \times 24.6 \times 10^6}{30.6 \times 10^6} = 122{,}565$$

This value is less than the yield strength for H-11 at 800°F, Table 19. Therefore, a 1/4-28 H-11 bolt stressed between 102,635 psi and 153,950 psi at room temperature will maintain a clamping load 1500 lb at 800°F after 1000 hr of operation. A cyclic loading of 100 lb, which results in a bolt loading between 1500 and 1600 lb will not cause fatigue failure at the operating conditions.

Table 16

PHYSICAL PROPERTIES OF MATERIALS USED TO MANUFACTURE ALLOY STEEL SHCS'S

Coefficient of Thermal Expansion, µm/m/°K[1]

20°C to 68°F to	100 212	200 392	300 572	400 752	500 932	600 1112
Material						
5137M, 51B37M[2]	–	12.6	13.4	13.9	14.3	14.6
4137[3]	11.2	11.8	12.4	13.0	13.6	–
4140[3]	12.3	12.7	–	13.7	–	14.5
4340[3]	–	12.4	–	13.6	–	14.5
8735[3]	11.7	12.2	12.8	13.5	–	14.1
8740[3]	11.6	12.2	12.8	13.5	–	14.1

Modulus of Elongation (Young's Modulus)

E = 30,000,000 PSI/in/in

Table 19 - Yield Strength at Various Temperatures

Alloy	Temperature (F) 70	800	1000	1200
Stainless Steels				
Type 302	35,000	35,000	34,000	30,000
Type 403	145,000	110,000	95,000	38,000
PH 15-7 Mo	220,000	149,000	101,000	–
High Strength Iron-Base Stainless Alloys				
A 286	95,000	95,000	90,000	85,000
AMS 5616	113,000	80,000	60,000	40,000
Unitemp 212	150,000	140,000	135,000	130,000
High Strength Iron-Base Alloys				
AISI 4340	200,000	130,000	75,000	–
H-11 (AMS 6485)	215,000	175,000	155,000	–
AMS 6340	160,000	100,000	75,000	–
Nickel-Base Alloys				
Iconel X	115,000	-	-	98,000
Waspaloy	115,000	-	106,000	100,000

NOTES:

1. Developed from ASM, Metals HDBK, 9th Edition, Vol. 1 (°C = °K for values listed)
2. ASME SA574
3. AISI
4. Multiply values in table by .556 for µin/in/°F.

CORROSION IN THREADED FASTENERS

All fastened joints are, to some extent, subjected to corrosion of some form during normal service life. Design of a joint to prevent premature failure due to corrosion must include considerations of the environment, conditions of loading , and the various methods of protecting the fastener and joint from corrosion.

Three ways to protect against corrosion are:

1. Select corrosion-resistant material for the fastener.
2. Specify protective coatings for fastener, joint interfaces, or both.
3. Design the joint to minimize corrosion.

The solution to a specific corrosion problem may require using one or all of these methods. Economics often necessitate a compromise solution.

Fastener Material

The use of a suitably corrosion-resistant material is often the first line of defense against corrosion. In fastener design, however, material choice may be only one of several important considerations. For example, the most corrosion-resistant material for a particular environment may just not make a suitable fastener.

Basic factors affecting the choice of corrosion resistant threaded fasteners are:

- Tensile and fatigue strength.
- Position on the galvanic series scale of the fastener and materials to be joined.
- Special design considerations: Need for minimum weight or the tendency for some materials to gall.
- Susceptibility of the fastener material to other types of less obvious corrosion. For example, a selected material may minimize direct attack of a corrosive environment only to be vulnerable to fretting or stress corrosion.

Some of the more widely used corrosion-resistant materials, along with approximate fastener tensile strength ratings at room temperature and other pertinent properties, are listed in Table 1. Sometimes the nature of corrosion properties provided by these fastener materials is subject to change with application and other conditions. For example, stainless steel and aluminum resist corrosion only so long as their protective oxide film remains unbroken. Alloy steel is almost never used, even under mildly corrosive conditions, without some sort of protective coating. Of course, the presence of a specific corrosive medium requires a specific corrosion-resistant fastener material, provided that design factors such as tensile and fatigue strength can be satisfied.

Protective Coating

A number of factors influence the choice of a corrosion-resistant coating for a threaded fastener. Frequently, the corrosion resistance of the coating is not a principal consideration. At times it is a case of economics. Often, less-costly fastener material will perform satisfactorily in a corrosive environment if given the proper protective coating.

Factors which affect coating choice are:

- Corrosion resistance
- Temperature limitations
- Embrittlement of base metal
- Effect on fatigue life
- Effect on locking torque
- Compatibility with adjacent material
- Dimensional changes
- Thickness and distribution
- Adhesion characteristics

Conversion Coatings: Where cost is a factor and corrosion is not severe, certain conversion-type coatings are effective. These include a black-oxide finish for alloy-steel screws and various phosphate base coatings for carbon and alloy-steel fasteners. Frequently, a rust-preventing oil is applied over a conversion coating.

Paint: Because of its thickness, paint is normally not considered for protective coatings for mating threaded fasteners. However, it is sometimes applied as a supplemental treatment at installation. In special cases, a fastener may be painted and installed wet, or the entire joint may be sealed with a coat of paint after installation.

TABLE 1 – TYPICAL PROPERTIES OF CORROSION RESISTANT FASTENER MATERIALS

Materials Stainless Steels	Tensile Strength (1000 psi)	Yield Strength at 0.2% offset (1000 psi)	Maximum Service Temp (F)	Mean Coefficient of Thermal Expan. (in./in./deg F)	Density (lbs/cu in.)	Base Cost Index	Position on Galvanic Scale
303, passive	80	40	800	10.2	0.286	Medium	8
303, passive, cold worked	125	80	800	10.3	0.286	Medium	9
410, passive	170	110	400	5.6	0.278	Low	15
431, passive	180	140	400	6.7	0.280	Medium	16
17-4 PH	200	180	600	6.3	0.282	Medium	11
17-7 PH	200	185	600	6.7	0.276	Medium	14
AM 350	200	162	800	7.2	0.282	Medium	13
15-7 Mo	200	155	600	–	0.277	Medium	12
A-286	150	85	1200	9.72	0.286	Medium	6
A-286, cold worked	220	170	1200	–	0.286	High	7

Electroplating: Two broad classes of protective electroplating are: 1. The barrier type-such as chrome plating-which sets up an impervious layer or film that is more noble and therefore more corrosion resistant than the base metal. 2. The sacrificial type, zinc for example, where the metal of the coating is less noble than the base metal of the fastener. This kind of plating corrodes sacrificially and protects the fastener.

Noble-metal coatings are generally not suitable for threaded fasteners-especially where a close-tolerance fit is involved. To be effective, a noble-metal coating must be at least 0.001 in. thick. Because of screw-thread geometry, however, such plating thickness will usually exceed the tolerance allowances on many classes of fit for screws.

Because of dimensional necessity, threaded fastener coatings, since they operate on a different principle, are effective in layers as thin as 0.0001 to 0.0002 in.

The most widely used sacrificial platings for threaded fasteners are cadmium, zinc, and tin. Frequently, the cadmium and zinc are rendered even more corrosion resistant by a post-plating chromate-type conversion treatment. Cadmium plating can be used at temperatures to 450°F. Above this limit, a nickel cadmium or nickel-zinc alloy plating is recommended. This consists of alternate deposits of the two metals which are heat-diffused into a uniform alloy coating that can be used for applications to 900°F. The alloy may also be deposited directly from the plating bath.

Fastener materials for use in the 900 to 1200°F range (stainless steel, A-286), and in the 1200° to 1800°F range (high-nickel-base super alloys) are highly corrosion resistant and normally do not require protective coatings, except under special environment conditions.

Silver plating is frequently used in the higher temperature ranges for lubrication to prevent galling and seizing, particularly on stainless steel. This plating can cause a galvanic corrosion problem, however, because of the high nobility of the silver.

Hydrogen Embrittlement: A serious problem, known as hydrogen embrittlement, can develop in plated alloy steel fasteners. Hydrogen generated during plating can diffuse into the steel and embrittle the bolt. The result is often a delayed and total mechanical failure, at tensile levels far below the theoretical strength, high-hardness structural parts are particularly susceptible to this condition. The problem can be controlled by careful selection of plating formulation, proper plating procedure, and sufficient baking to drive off any residual hydrogen.

Another form of hydrogen embrittlement, which is more difficult to control, may occur after installation. Since electrolytic cell action liberates hydrogen at the cathode, it is possible for either galvanic or concentration-cell corrosion to lead to embrittling of the bolt material.

Joint Design

Certain precautions and design procedures can be followed to prevent, or at least minimize, each of the various types of corrosion likely to attack a threaded joint. The most important of these are:

For Direct Attack: Choose the right corrosion-resistant material. Usually a material can be found that will provide the needed corrosion resistance without sacrifice of other important design requirements. Be sure that the fastener material is compatible with the materials being joined.

Corrosion resistance can be increased by using a conversion coating such as black oxide or a phosphate-base treatment. Alternatively, a sacrificial coating such as zinc plating is effective.

For an inexpensive protective coating, lacquer or paint can be used where conditions permit.

For Galvanic Corrosion: If the condition is severe, electrically insulate the bolt and joint from each other..

The fastener may be painted with zinc chromate primer prior to installation, or the entire joint can be coated with lacquer or paint.

Another protective measure is to use a bolt that is cathodic to the joint material and close to it in the galvanic series. When the joint material is anodic, corrosion will spread over the greater area of the fastened materials. Conversely, if the bolt is anodic, galvanic action is most severe.

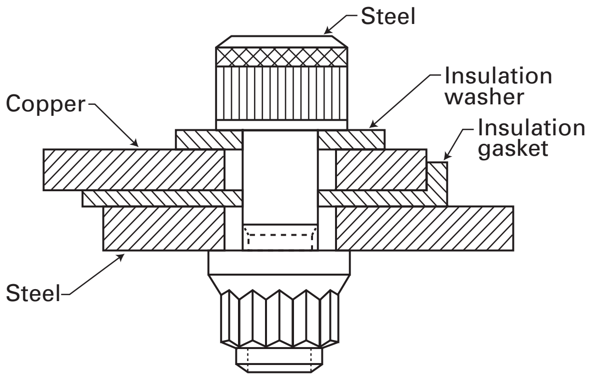


FIG. 1.1 – A method of electrically insulating a bolted joint to prevent galvanic corrosion.

For Concentration-Cell Corrosion: Keep surfaces smooth and minimize or eliminate lap joints, crevices, and seams. Surfaces should be clean and free of organic material and dirt. Air trapped under a speck of dirt on the surface of the metal may form an oxygen concentration cell and start pitting.

For maximum protection, bolts and nuts should have smooth surfaces, especially in the seating areas. Flush-head bolts should be used where possible. Further, joints can be sealed with paint or other sealant material.

For Fretting Corrosion: Apply a lubricant (usually oil) to mating surfaces. Where fretting corrosion is likely to occur: 1. Specify materials of maximum practicable hardness. 2. Use fasteners that have residual compressive stresses on the surfaces that may be under attack. 3. Specify maximum preload in the joint. A higher clamping force results in a more rigid joint with less relative movement possible between mating services.

CORROSION IN THREADED FASTENERS

For Stress Corrosion: Choose a fastener material that resists stress corrosion in the service environment. Reduce fastener hardness (if reduced strength can be tolerated), since this seems to be a factor in stress corrosion.

Minimize crevices and stress risers in the bolted joint and compensate for thermal stresses. Residual stresses resulting from sudden changes in temperature accelerate stress corrosion.

If possible, induce residual compressive stresses into the surface of the fastener by shot-peening or pressure rolling.

For Corrosion Fatigue: In general, design the joint for high fatigue life, since the principal effect of this form of corrosion is reduced fatigue performance. Factors extending fatigue performance are: 1. Application and maintenance of a high preload. 2. Proper alignment to avoid bending stresses.

If the environment is severe, periodic inspection is recommended so that partial failures may be detected before the structure is endangered.

As with stress and fretting corrosion, compressive stresses induced on the fastener surfaces by thread rolling, fillet rolling, or shot peening will reduce corrosion fatigue. Further protection is provided by surface coating.

TYPES OF CORROSION

Direct Attack…most common form of corrosion affecting all metals and structural forms. It is a direct and general chemical reaction of the metal with a corrosive medium-liquid, gas, or even a solid.

Galvanic Corrosion…occurs with dissimilar metals contact. Presence of an electrolyte, which may be nothing more than an individual atmosphere, causes corrosive action in the galvanic couple. The anodic, or less noble material, is the sacrificial element. Hence, in a joint of stainless steel and titanium, the stainless steel corrodes. One of the worst galvanic joints would consist of magnesium and titanium in contact.

Concentration Cell Corrosion…takes place with metals in close proximity and, unlike galvanic corrosion, does not require dissimilar metals. When two or more areas on the surface of a metal are exposed to different concentrations of the same solution, a difference in electrical potential results, and corrosion takes place.

If the solution consists of salts of the metal itself, a metal-ion cell is formed, and corrosion takes place on the surfaces in close contact. The corrosive solution between the two surfaces is relatively more stagnant (and thus has a higher concentration of metal ions in solution) than the corrosive solution immediately outside the crevice.

A variation of the concentration cell is the oxygen cell in which a corrosive medium, such as moist air, contains different amounts of dissolved oxygen at different points. Accelerated corrosion takes place between hidden surfaces (either under the bolt head or nut, or between bolted materials) and is likely to advance without detection.

Fretting…corrosive attack or deterioration occurring between containing, highly-loaded metal surfaces subjected to very slight (vibratory) motion. Although the mechanism is not completely understood, it is probably a highly accelerated form of oxidation under heat and stress. In threaded joints, fretting can occur between mating threads, at the bearing surfaces under the head of the screw, or under the nut. It is most likely to occur in high tensile, high-frequency, dynamic-load applications. There need be no special environment to induce this form of corrosion...merely the presence of air plus vibratory rubbing. It can even occur when only one of the materials in contact is metal.

Stress Corrosion Cracking…occurs over a period of time in high-stressed, high-strength joints. Although not fully understood, stress corrosion cracking is believed to be caused by the combined and mutually accelerating effects of static tensile stress and corrosive environment. Initial pitting somehow tales place which, in turn, further increases stress build-up. The effect is cumulative and, in a highly stressed joint, can result in sudden failure.

Corrosion Fatigue…accelerated fatigue failure occurring in the presence of a corrosive medium. It differs from stress corrosion cracking in that dynamic alternating stress, rather than static tensile stress, is the contributing agent.

Corrosion fatigue affects the normal endurance limit of the bolt. The conventional fatigue curve of a normal bolt joint levels off at its endurance limit, or maximum dynamic load that can be sustained indefinitely without fatigue failure. Under conditions of corrosion fatigue, however, the curve does not level off but continues downward to a point of failure at a finite number of stress cycles.

GALVANIC CORROSION

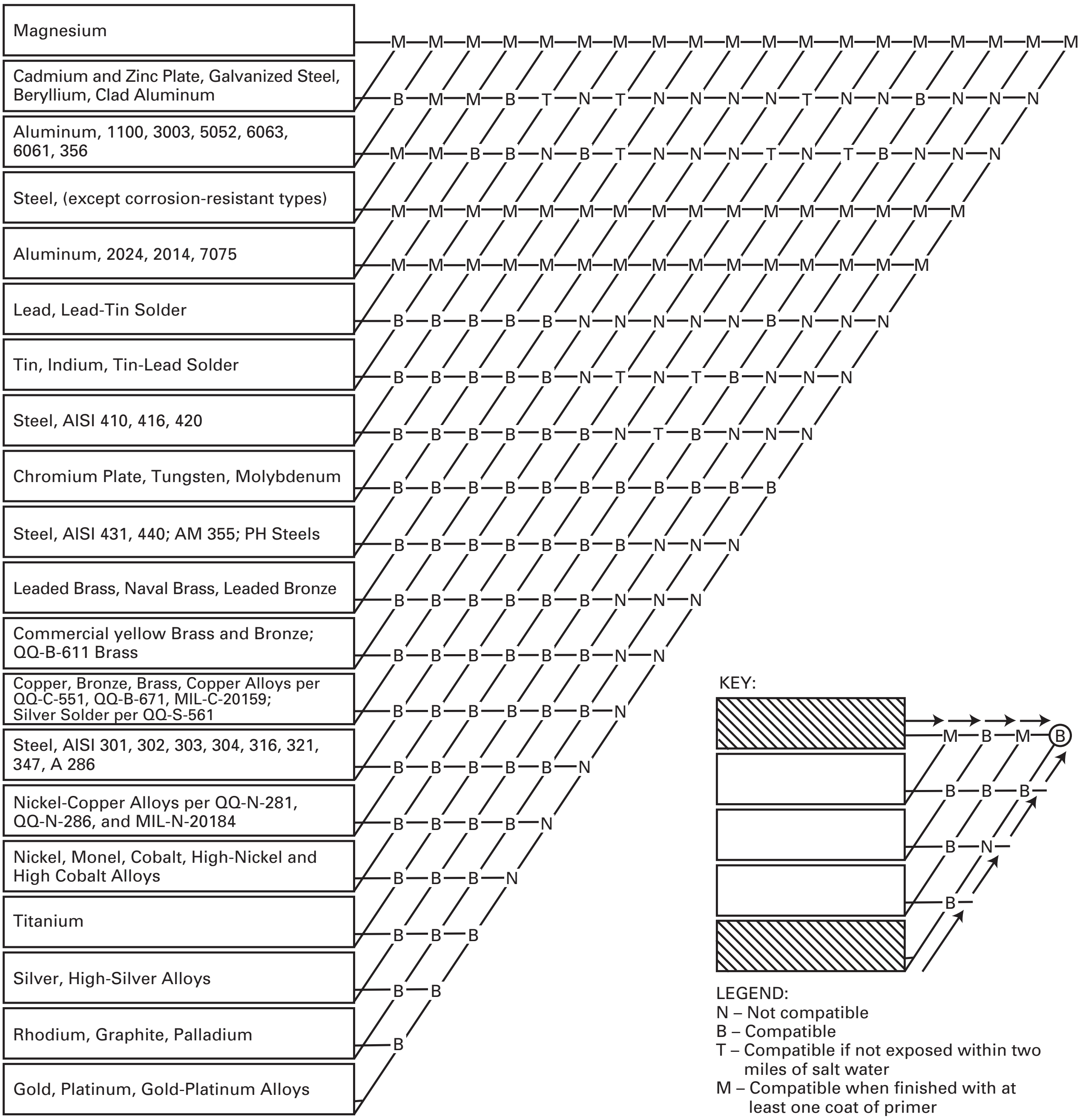


FIG. 19 – Metals compatibility chart

THE IMPACT PERFORMANCE OF THREADED FASTENERS

Much has been written regarding the significance of the notched bar impact testing of steels and other metallic materials. The Charpy and Izod type test relate notch behavior (brittleness versus ductility) by applying a single overload of stress. The results of these tests provide quantitive comparisons but are not convertible to energy values useful for engineering design calculations. The results of an individual test are related to that particular specimen size, notch geometry and testing conditions and cannot be generalized to other sizes of specimens and conditions.

The results of these tests are useful in determining the susceptibility of a material to brittle behavior when the applied stress is perpendicular to the major stress.

In externally threaded fasteners, however, the loading usually is applied in a longitudinal direction. The impact test, therefore, which should be applicable would be one where the applied impact stress supplements the major stress. Only in shear loading on fasteners is the major stress in the transverse direction.

Considerable testing has been conducted in an effort to determine if a relationship exists between the Charpy V notch properties of a material and the tension properties of an externally threaded fastener manufactured from the same material.

Some conclusions which can be drawn from the extensive impact testing are as follows:

1. The tension impact properties of externally threaded fasteners do not follow the Charpy V notch impact pattern.
2. Some of the variables which effect the tension impact properties are:
   A. The number of exposed threads
   B. The length of the fastener
   C. The relationship of the fastener shank diameter to the thread area.
   D. The hardness or fastener ultimate tensile strength

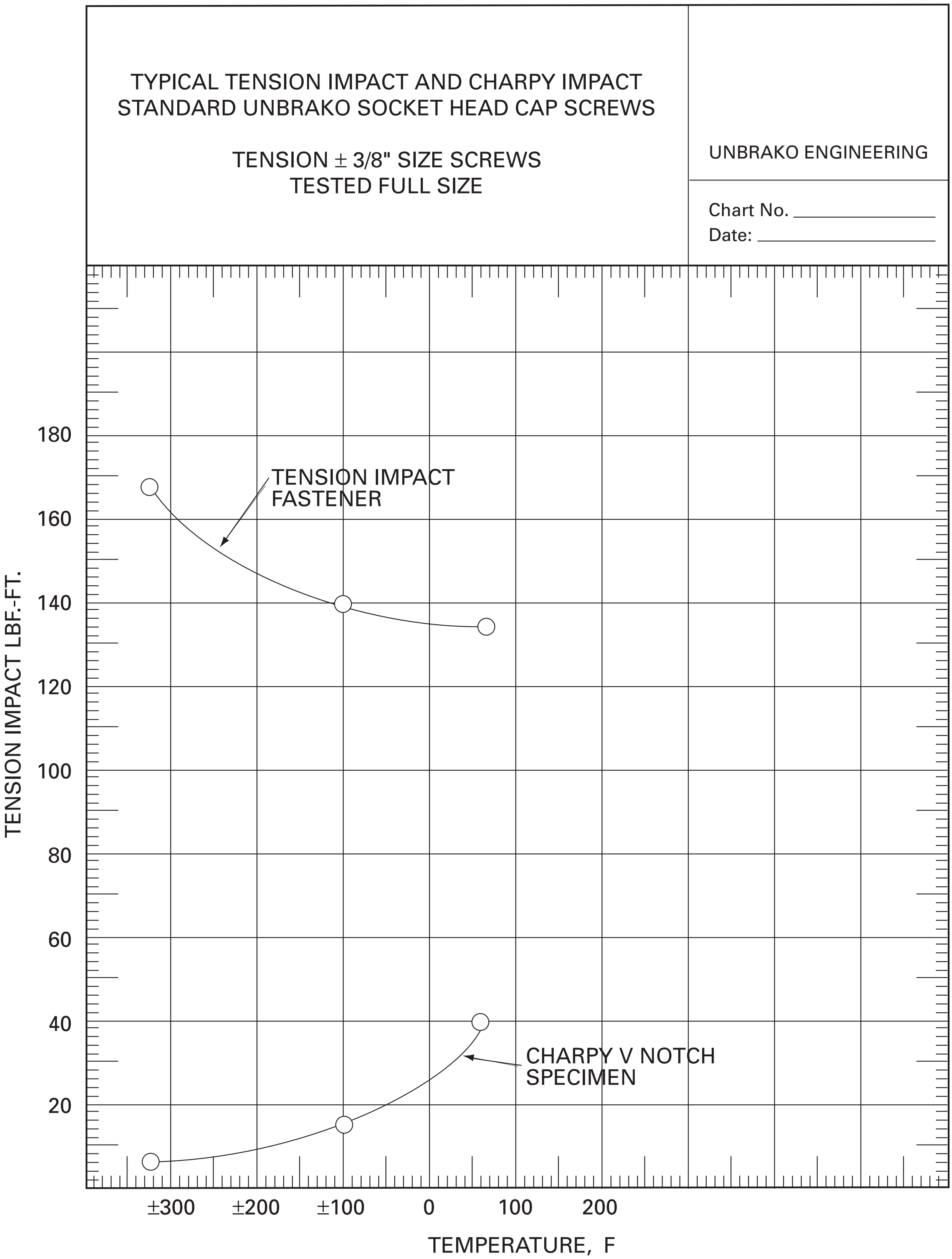
Following are charts showing tension impact versus Charpy impact properties, the effect of strength and diameter on tension impact properties and the effect of test temperature.

Please note from figure 21 that while the Charpy impact strength of socket head cap screw materials are decreasing at sub-zero temperatures, the tension impact strength of the same screws is increasing. This compares favorable with the effect of cryogenic temperatures on the tensile strength of the screws. Note the similar increase in tensile strength shown in figure 22.

It is recommended, therefore, that less importance be attached to Charpy impact properties of materials which are intended to be given to impact properties for threaded fasteners. If any consideration is to be given to impact properties of bolts or screws, it is advisable to investigate the tension impact properties of full size fasteners since this more closely approximates the actual application.

TABLE 20
LOW-TEMPERATURE IMPACT PROPERTIES OF SELECTED ALLOY STEELS

AISI no.	composition, %					heat temperature*		Hardness Rc	impact energy, ft.-lb					transition temp. (50% brittle) °F
	C	Mn	Ni	Cr	Mo	quenching temp. F+	tempering temp. F		–300°F	–200°F	–100°F	0°F	100°F	
4340	0.38	0.77	1.65	0.93	0.21	1550	400	52	11	15	20	21	21	–
							600	48	10	14	15	15	16	–
							800	44	9	13	16	21	25	–
							1000	38	15	18	28	36	36	–130
							1200	30	15	28	55	55	55	–185
4360	0.57	0.87	1.62	1.08	0.22	1475	800	48	5	6	10	11	14	–
							1000	40	9	10	13	18	23	–10
							1200	30	12	15	25	42	43	–110
4380	0.76	0.91	1.67	1.11	0.21	1450	800	49	4	5	8	9	10	–
							1000	42	8	8	10	12	15	60
							1200	31	5	11	19	33	38	–50
4620	0.20	0.67	1.85	0.30	0.18	1650	300	42	14	20	28	35	35	–
							800	34	11	16	33	55	55	–
							1000	29	16	34	55	78	78	–
							1200	19	17	48	103	115	117	–
4640	0.43	0.69	1.78	0.29	0.20	1550	800	42	16	17	20	25	27	–
							1000	37	17	22	35	39	69	–190
							1200	29	17	30	55	97	67	–180
4680	0.74	0.77	1.81	0.30	0.21	1450	800	46	5	8	13	15	16	–
							1000	41	11	12	15	19	22	–
							1200	31	11	13	17	39	43	–
8620	0.20	0.89	0.60	0.68	0.20	1650	300	43	11	16	23	35	35	–
							800	36	8	13	20	35	45	–20
							1000	29	25	33	65	76	76	–150
							1200	21	10	85	107	115	117	–195
8630	0.34	0.77	0.66	0.62	0.22	1575	800	41	7	12	17	25	31	0
							1000	34	11	20	43	53	54	–155
							1200	27	18	28	74	80	82	–165
8640	0.45	0.78	0.65	0.61	0.20	1550	800	46	5	10	14	20	23	–
							1000	38	11	15	24	40	40	–110
							1200	30	18	22	49	63	66	–140
8660	0.56	0.81	0.70	0.56	0.25	1475	800	47	4	6	10	13	16	–
							1000	41	10	12	15	20	30	–10
							1200	30	16	18	25	54	60	–90



TYPICAL TENSION IMPACT AND CHARPY IMPACT
STANDARD UNBRAKO SOCKET HEAD CAP SCREWS
TENSION ± 3/8" SIZE SCREWS
TESTED FULL SIZE
UNBRAKO ENGINEERING
Chart No.
Date:
TENSION IMPACT FASTENER
CHARPY V NOTCH SPECIMEN
TENSION IMPACT LBF.-FT.
180
160
140
120
100
80
60
40
20
±300
±200
±100
0
100
200
TEMPERATURE, F

FIG. 21

Unbrako

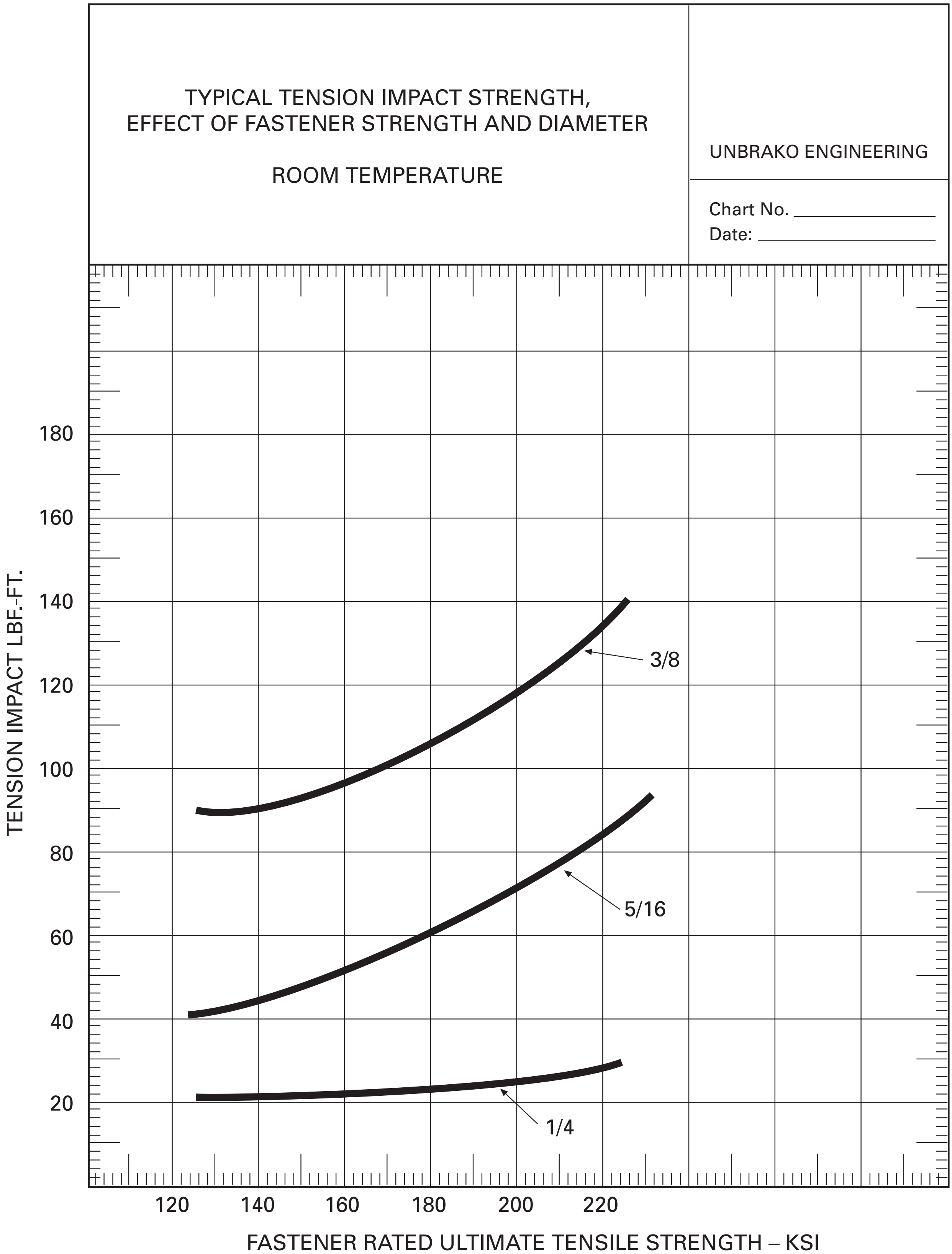


FIG. 22

UNBRAKO PRODUCT ENGINEERING BULLETIN

Standard Inch Socket Head Cap Screws Are Not Grade 8 Fasteners

There is a common, yet reasonable, misconception that standard, inch, alloy steel socket head cap screws are "Grade 8". This is not true. The misconception is reasonable because "Grade 8" is a term generally associated with "high strength" fasteners. A person desiring a "high strength" SHCS may request a "Grade 8 SHCS". This is technically incorrect for standard SHCSs. The term Grade 8 defines specific fastener characteristics which must be met to be called "Grade 8". Three of the most important characteristics are not consistent with requirements for industry standard SHCSs: tensile strength, hardness, and head marking. Some basic differences between several fastener classifications are listed below. The list is not comprehensive but intended to provide a general understanding. SHCSs can be manufactured to meet Grade 8 requirements on a special order basis.

Fastener Designation	Grade 2	Grade 5	Grade 8	Industry SHCS	Unbrako SHCS
Applicable Standard	SAE J429	SAE J429	SAE J429	ASTM A574	ASTM A574 UNB-B-271
Strength Level, UTS KSI, min.	74 (1/4-3/4) 60 (7/8 - 1 1/2)	120 (1/4 - 1) 105 (1 1/8 - 1 1/2)	150 (1/4 - 11/2)	180 (≤1/2) 170 (> 1/2)	190 (≤ 1/2) 180 (> 1/2)
Hardness, Rockwell	B80-B100 B70-B100	C25-C34 C19-C30	C33-C39	C39-C45 C37-C45	C39-C43 C38-C43
General Material Type	Low or Medium Carbon Steel	Medium Carbon Steel	Medium Carbon Alloy Steel	Medium Carbon Alloy Steel	Medium Carbon Alloy Steel
Identification Requirement	None	Three Radial Lines	Six Radial Lines	SHCS Configuration	Mfr's ID
Typical Fasteners	Bolts Screws Studs Hex Heads	Bolts Screws Studs Hex Heads	Bolts Screws Studs Hex Heads	Socket Head Cap Screws	Socket Head Cap Screws

Unbrako

THREADS IN BOTH SYSTEMS

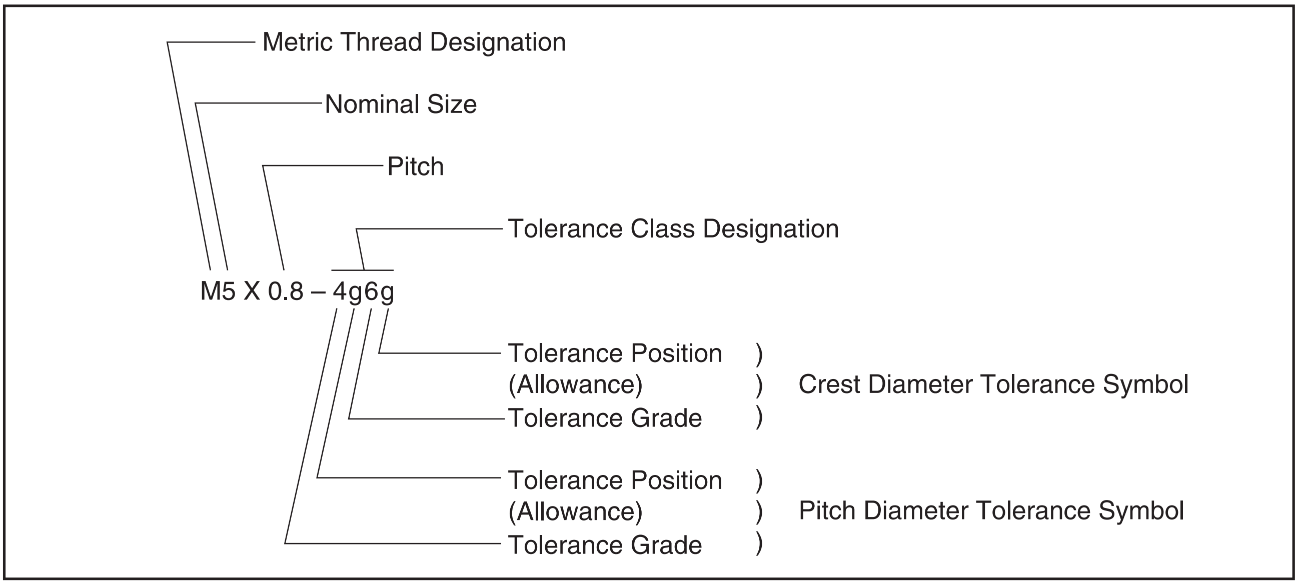
Thread forms and designations have been the subject of many long and arduous battles through the years. Standardization in the inch series has come through many channels, but the present unified thread form could be considered to be the standard for many threaded products, particularly high strength ones such as socket head cap screws, etc. In common usage in U.S.A., Canada and United Kingdom are the Unified National Radius Coarse series, designated UNRC, Unified National Radius Fine series, designated UNRF, and several special series of various types, designated UNS. This thread, UNRC or UNRF, is designated by specifying the diameter and threads per inch along with the suffix indicating the thread series, such as 1/4 - 28 UNRF. For threads in Metric units, a similar approach is used, but with some slight variations. A diameter and pitch are used to designate the series, as in the Inch system, with modifications as follows: For coarse threads, only the prefix M and the diameter are necessary, but for fine threads, the pitch is shown as a suffix. For example, M16 is a coarse thread designation representing a diameter of 16 mm with a pitch of 2 mm understood. A similar fine thread part would be M16 x 1.5 or 16 mm diameter with a pitch of 1.5 mm.

For someone who has been using the Inch system, there are a couple of differences that can be a little confusing. In the Inch series, while we refer to threads per inch as pitch; actually the number of threads is 1/pitch. Fine threads are referenced by a larger number than coarse threads because they “fit” more threads per inch.

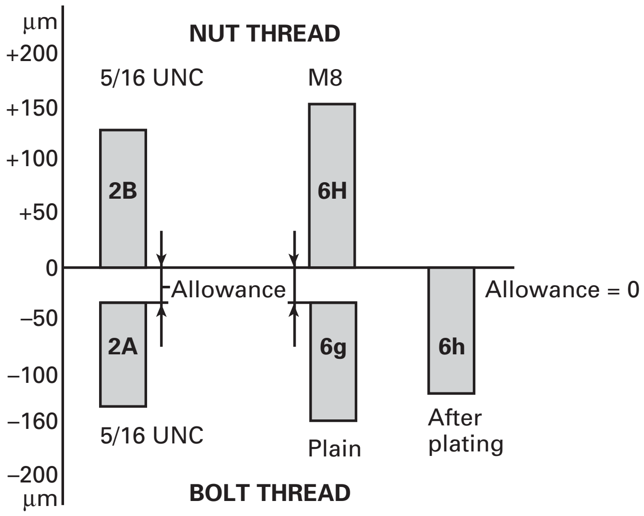
In Metric series, the diameters are in millimeters, but the pitch is really the pitch. Consequently the coarse thread has the large number. The most common metric thread is the coarse thread and falls generally between the inch coarse and fine series for a comparable diameter.

Also to be considered in defining threads is the tolerance and class of fit to which they are made. The International Standards Organization (ISO) metric system provides for this designation by adding letters and numbers in a certain sequence to the callout. For instance, a thread designated as M5 x 0.8 4g6g would define a thread of 5 mm diameter, 0.8 mm pitch, with a pitch diameter tolerance grade 6 and allowance “g”. These tolerances and fields are defined as shown below, similar to the Federal Standard H28 handbook, which defines all of the dimensions and tolerances for a thread in the inch series. The callout above is similar to a designation class 3A fit, and has a like connotation.

COMPLETE DESIGNATIONS



Example of thread tolerance positions and magnitudes.
Comparision 5/16 UNC and M8. Medium tolerance grades – Pitch diameter.



DEVIATIONS

external	internal	basic clearance
h g e	H G	none small large

NOTE:
Lower case letters = external threads
Capital letters = internal threads

THROUGH-HOLE PREPARATION

DRILL AND COUNTERBORE SIZES FOR INCH SOCKET HEAD CAP SCREWS

Note 1

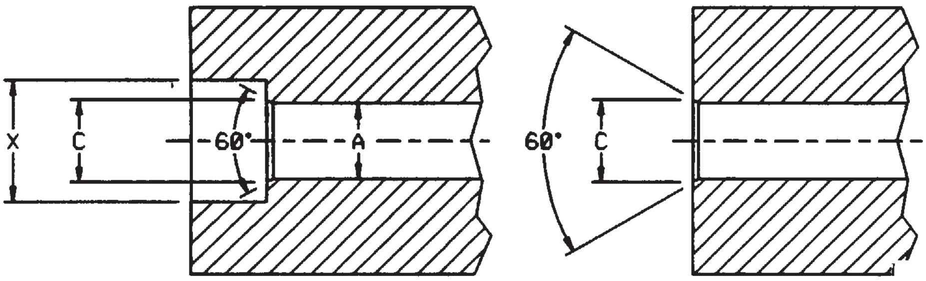
Close Fit: Normally limited to holes for those lengths of screws threaded to the head in assemblies in which: (1) only one screw is used; or (2) two or more screws are used and the mating holes are produced at assembly or by matched and coordinated tooling.

Note 2

Normal Fit: Intended for: (1) screws of relatively long length; or (2) assemblies that involve two or more screws and where the mating holes are produced by conventional tolerancing methods. It provides for the maximum allowable eccentricty of the longest standard screws and for certain deviations in the parts being fastened, such as deviations in hole straightness; angularity between the axis of the tapped hole and that of the hole for the shank; differneces in center distances of the mating holes and other deviations.

Note 3

Chamfering: It is considered good practice to chamfer or break the edges of holes that are smaller than "F" maximum in parts in which hardness approaches, equals or exceeds the screw hardness. If holes are not chamfered, the heads may not seat properly or the sharp edges may deform the fillets on the screws, making them susceptible to fatigue in applications that involve dynamic loading. The chamfers, however, should not be larger than needed to ensure that the heads seat properly or that the fillet on the screw is not deformed. Normally, the chamfers do not need to exceed "F" maximum. Chamfers exceeding these values reduce the effective bearing area and introduce the possibility of indentation when the parts fastened are softer than screws, or the possiblity of brinnelling of the heads of the screws when the parts are harder than the screws. (See "F" page 6).



		A				X	C	hole dimensions			
		drill size for hole A				counter-bore diameter	countersink diameter D Max. + 2F(Max.)	tap drill size		**body drill size	counter-bore size
		close fit		normal fit							
nominal size	basic screw diameter	nom.	dec.	nom.	dec.			UNRC	UNRF		
0	0.0600	51*	0.0670	49*	0.0730	1/8	0.074	–	3/64	#51	1/8
1	0.0730	46*	0.0810	43*	0.0890	5/32	0.087	1.5mm	#53	#46	5/32
2	0.0860	3/32	0.0937	36*	0.1065	3/16	0.102	#50	#50	3/32	3/16
3	0.0990	36*	0.1065	31*	0.1200	7/32	0.115	#47	#45	#36	7/32
4	0.1120	1/8	0.1250	29*	0.1360	7/32	0.130	#43	#42	1/8	7/32
5	0.1250	9/64	0.1406	23*	0.1540	1/4	0.145	#38	#38	9/64	1/4
6	0.1380	23*	0.1540	18*	0.1695	9/32	0.158	#36	#33	#23	9/32
8	0.1640	15*	0.1800	10	0.1935	5/16	0.188	#29	#29	#15	5/16
10	0.1900	5*	0.2055	2*	0.2210	3/8	0.218	#25	#21	#5	3/8
1/4	0.2500	17/64	0.2656	9/32	0.2812	7/16	0.278	#7	#3	17/64	7/16
5/16	0.3125	21/64	0.3281	11/32	0.3437	17/32	0.346	F	I	21/64	17/32
3/8	0.0375	25/64	0.3906	13/32	0.4062	5/8	0.415	5/16	Q	25/64	5/8
7/16	0.4375	29/64	0.4531	15/32	0.4687	23/32	0.483	U	25/64	29/64	23/32
1/2	0.5000	33/64	0.5156	17/32	0.5312	13/16	0.552	27/64	29/64	33/64	13/16
5/8	0.6250	41/64	0.6406	21/32	0.6562	1	0.689	35/64	14.5mm	41/64	1
3/4	0.7500	49/64	0.7656	25/32	0.7812	1-3/16	0.828	21/32	11/16	49/64	1-3/16
7/8	0.8750	57/64	0.8906	29/32	0.9062	1-3/8	0.963	49/64	20.5mm	57/64	1-3/8
1	1.0000	1-1/64	1.0156	1-1/32	1.0312	1-5/8	1.100	7/8	59/64	1-1/64	1-5/8
1-1/4	1.2500	1-9/32	1.2812	1-5/16	1.3125	2	1.370	1-7/64	1-11/64	1-9/32	2
1-1/2	1.5000	1-17/32	1.5312	1-9/16	1.5625	2-3/8	1.640	34mm	36mm	1-17/32	2-3/8

** Break edge of body drill hole to clear screw fillet.

DRILL AND COUNTERBORE SIZES

DRILL AND COUNTERBORE SIZES FOR METRIC SOCKET HEAD CAP SCREWS

Note 1

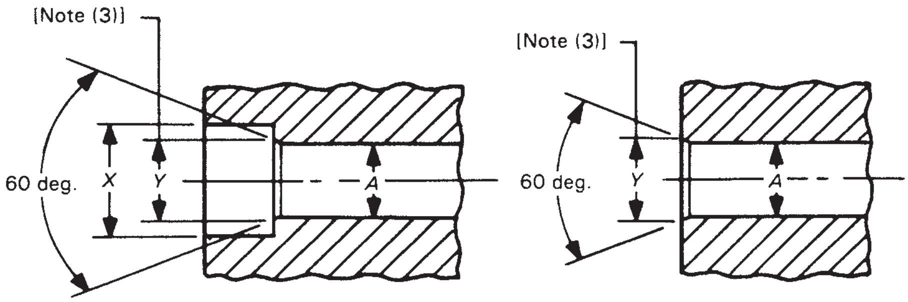
Close Fit: Normally limited to holes for those lengths of screws threaded to the head in assemblies in which: (1) only one screw is used; or (2) two or more screws are used and the mating holes are produced at assembly or by matched and coordinated tooling.

Note 2

Normal Fit: Intended for: (1) screws of relatively long length; or (2) assemblies that involve two or more screws and where the mating holes are produced by conventional tolerancing methods. It provides for the maximum allowable eccentricty of the longest standard screws and for certain deviations in the parts being fastened, such as deviations in hole straightness; angularity between the axis of the tapped hole and that of the hole for the shank; differneces in center distances of the mating holes and other deviations.

Note 3

Chamfering: It is considered good practice to chamfer or break the edges of holes that are smaller than “B” maximum in parts in which hardness approaches, equals or exceeds the screw hardness. If holes are not chamfered, the heads may not seat properly or the sharp edges may deform the fillets on the screws, making them susceptible to fatigue in applications that involve dynamic loading. The chamfers, however, should not be larger than needed to ensure that the heads seat properly or that the fillet on the screw is not deformed. Normally, the chamfers do not need to exceed “B” maximum. Chamfers exceeding these values reduce the effective bearing area and introduce the possibility of indentation when the parts fastened are softer than screws, or the possiblity of brinnelling of the heads of the screws when the parts are harder than the screws.



Nominal Size or Basic Screw Diameter	*A* Nominal Drill Size Close Fit [Note (1)]	*A* Nominal Drill Size Normal Fit [Note (2)]	*X* Counterbore Diameter	*Y* Countersink Diameter [Note (3)]	*B* Transition Diameter, Max.
M1.6	1.80	1.95	3.50	2.0	2.0
M2	2.20	2.40	4.40	2.6	2.6
M2.5	2.70	3.00	5.40	3.1	3.1
M3	3.40	3.70	6.50	3.6	3.6
M4	4.40	4.80	8.25	4.7	4.7
M5	5.40	5.80	9.75	5.7	5.7
M6	6.40	6.80	11.25	6.8	6.8
M8	8.40	8.80	14.25	9.2	9.2
M10	10.50	10.80	17.25	11.2	11.2
M12	12.50	12.80	19.25	14.2	14.2
M14	14.50	14.75	22.25	16.2	16.2
M16	16.50	16.75	25.50	18.2	18.2
M20	20.50	20.75	31.50	22.4	22.4
M24	24.50	24.75	37.50	26.4	26.4
M30	30.75	31.75	47.50	33.4	33.4
M36	37.00	37.50	56.50	39.4	39.4
M42	43.00	44.0	66.00	45.6	45.6
M48	49.00	50.00	75.00	52.6	52.6

HARDNESS – TENSILE CONVERSION

INCH
ROCKWELL – BRINELL – TENSILE CONVERSION

Rockwell "C" scale	Brinell hardness number	tensile strength approx. 1000 psi
60	654	336
59	634	328
58	615	319
57	595	310
56	577	301
55	560	292
54	543	283
53	524	274
52	512	265
51	500	257
50	488	249
49	476	241
48	464	233
47	453	225
46	442	219
45	430	212
44	419	206

Rockwell "C" scale	Brinell hardness number	tensile strength approx. 1000 psi
43	408	200
42	398	194
41	387	188
40	377	181
39	367	176
38	357	170
37	347	165
36	337	160
35	327	155
34	318	150
33	309	147
32	301	142
31	294	139
30	285	136
29	279	132
28	272	129
27	265	126

Rockwell		Brinell hardness number	tensile strength approx. 1000 psi
"C" scale	"B" scale		
26		259	123
25		253	120
24		247	118
23		241	115
22	100	235	112
21	99	230	110
20	98	225	107
(19)		220	104
(18)	97	215	103
(17)		210	102
(16)	96	206	100
(15)		201	99
(14)	95	197	97
(13)	94	193	96
(12)	93	190	93
(11)		186	91
(10)	92	183	90

METRIC
ROCKWELL – BRINELL – TENSILE CONVERSION

Rockwell "C" scale	Brinell hardness number	tensile strength approx. MPa
60	654	2,317
59	634	2,261
58	615	2,199
57	595	2,137
56	577	2,075
55	560	2,013
54	543	1,951
53	524	1,889
52	512	1,827
51	500	1,772
50	488	1,717
49	476	1,662
48	464	1,606
47	453	1,551
46	442	1,510
45	430	1,462
44	419	1,420

Rockwell "C" scale	Brinell hardness number	tensile strength approx. MPa
43	408	1,379
42	398	1,338
41	387	1,296
40	377	1,248
39	367	1,213
38	357	1,172
37	347	1,138
36	337	1,103
35	327	1,069
34	318	1,034
33	309	1,014
32	301	979
31	294	958
30	285	938
29	279	910
28	272	889
27	265	869

Rockwell		Brinell hardness number	tensile strength approx. MPa
"C" scale	"B" scale		
26		259	848
25		253	827
24		247	814
23		241	793
22	100	235	772
21	99	230	758
20	98	225	738
(19)		220	717
(18)	97	215	710
(17)		210	703
(16)	96	206	690
(15)		201	683
(14)	95	197	669
(13)	94	193	662
(12)	93	190	641
(11)		186	627
(10)	92	183	621

THREAD STRESS AREAS

Inch and Metric

STRESS AREAS FOR THREADED FASTENERS – INCH

Diameter (in.)		Diameter (mm)	Threads Per in.		Square Inches		
					Tensile Stress Area Per H-28		
			UNRC	UNRF	UNRC	UNRF	Nominal Shank
#0	0.06	1.52	–	80	–	0.00180	0.002827
#1	0.07	1.85	64	72	0.00263	0.00278	0.004185
#2	0.09	2.18	56	64	0.00370	0.00394	0.005809
#3	0.10	2.51	48	56	0.00487	0.00523	0.007698
#4	0.11	2.84	40	48	0.00604	0.00661	0.009852
#5	0.13	3.18	40	44	0.00796	0.00830	0.012272
#6	0.14	3.51	32	40	0.00909	0.01015	0.014957
#8	0.16	4.17	32	36	0.0140	0.01474	0.021124
#10	0.19	4.83	24	32	0.0175	0.0200	0.028353
1/4	0.25	6.35	20	28	0.0318	0.0364	0.049087
5/16	0.31	7.94	18	24	0.0524	0.0580	0.076699
3/8	0.38	9.53	16	24	0.0775	0.0878	0.11045
7/16	0.44	11.11	14	20	0.1063	0.1187	0.15033
1/2	0.50	12.70	13	20	0.1419	0.1599	0.19635
9/16	0.56	14.29	12	18	0.182	0.203	0.25
5/8	0.63	15.88	11	18	0.226	0.256	0.31
3/4	0.75	19.05	10	16	0.334	0.373	0.44179
7/8	0.88	22.23	9	14	0.462	0.509	0.60132
1	1.00	25.40	8	12	0.606	0.663	0.79
1-1/8	1.13	28.58	7	12	0.763	0.856	0.99402
1-1/4	1.25	31.75	7	12	0.969	1.073	1.2272
1-3/8	1.38	34.93	6	12	1.155	1.315	1.4849
1-1/2	1.50	38.10	6	12	1.405	1.581	1.7671
1-3/4	1.75	44.45	5	12	1.90	2.19	2.4053
2	2.00	50.80	4-1/2	12	2.50	2.89	3.1416
2-1/4	2.25	57.15	4-1/2	12	3.25	3.69	3.9761
2-1/2	2.50	63.50	4	12	4.00	4.60	4.9088
2-3/4	2.75	69.85	4	12	4.93	5.59	5.9396
3	3.00	76.20	4	12	5.97	6.69	7.0686

STRESS AREAS FOR THREADED FASTENERS – METRIC

Nominal Dia. Thread and Pitch (mm)	Thread Tensile Stress Area (mm^2)	Nominal Shank Area (mm^2)
1.6 x 0.35	1.27	2.01
2.0 x 0.4	2.07	3.14
2.5 x 0.45	3.39	4.91
3.0 x 0.5	5.03	7.07
4.0 x 0.7	8.78	12.6
5.0 x 0.8	14.2	19.6
6.0 x 1	20.1	28.3
8.0 x 1.25	36.6	50.3
10 x 1.5	58.00	78.5
12 x 1.75	84.3	113
14 x 2	115	154
16 x 2	157	201

Nominal Dia. Thread and Pitch (mm)	Thread Tensile Stress Area (mm^2)	Nominal Shank Area (mm^2)
18 x 2.5	192	254
20 x 2.5	245	314
22 x 2.5	303	380
24 x 3	353	452
27 x 3	459	573
30 x 3.5	561	707
33 x 3.5	694	855
36 x 4	817	1018
42 x 4.5	1120	1385
48 x 5	1470	1810

Advantages built into every detail.

THE UNBRAKO DIFFERENCE
Your application demands a fastener which outperforms all others. We build our products for life, to help you build your products for life.

What's holding your product together?

CALL FOR A SAMPLE AND EXPERIENCE THE UNBRAKO DIFFERENCE FOR YOURSELF.

HIGHER MIN ULT TENSILE
10,000 PSI stronger than industry standard

COMPOUND FILLET RADIUS
Doubles fatigue life at critical head-shank juncture

WIDE RADIUS THREADS
Maximizes fatigue resistance where it's needed most

3R (RADIUSED ROOT RUNOUT) THREAD
Increases fatigue life up to 300%

E CODE "LOT CODE" MARKINGS
The ultimate in fastener traceability

Unbrako®
BUILT FOR LIFE.

FFT
Fasteners, Fixings & Tools

www.fastenersfixingsandtools.co.uk
Email : info@fasteners-ft.co.uk
Tel.01234 333949
Fax.01234 211069