Order from Chaos

BUFAB

YOUR SMALL PART SUPPLIER

Disclaimer
This book is meant to be advisory and is not intended for specific applications. This book is not intended to address all the possible contexts and situations involving mechanical strength in fasteners.

This book is designed for generic study or research use only and is not a substitute for specific training or experience. Nor is this book intended to serve as a substitute for any proper training and/or certification needed for choosing the correct fastener.

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BULTEN'S TECHNICAL HANDBOOK

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CONTENTS

FOREWORD

1.1 IN GENERAL

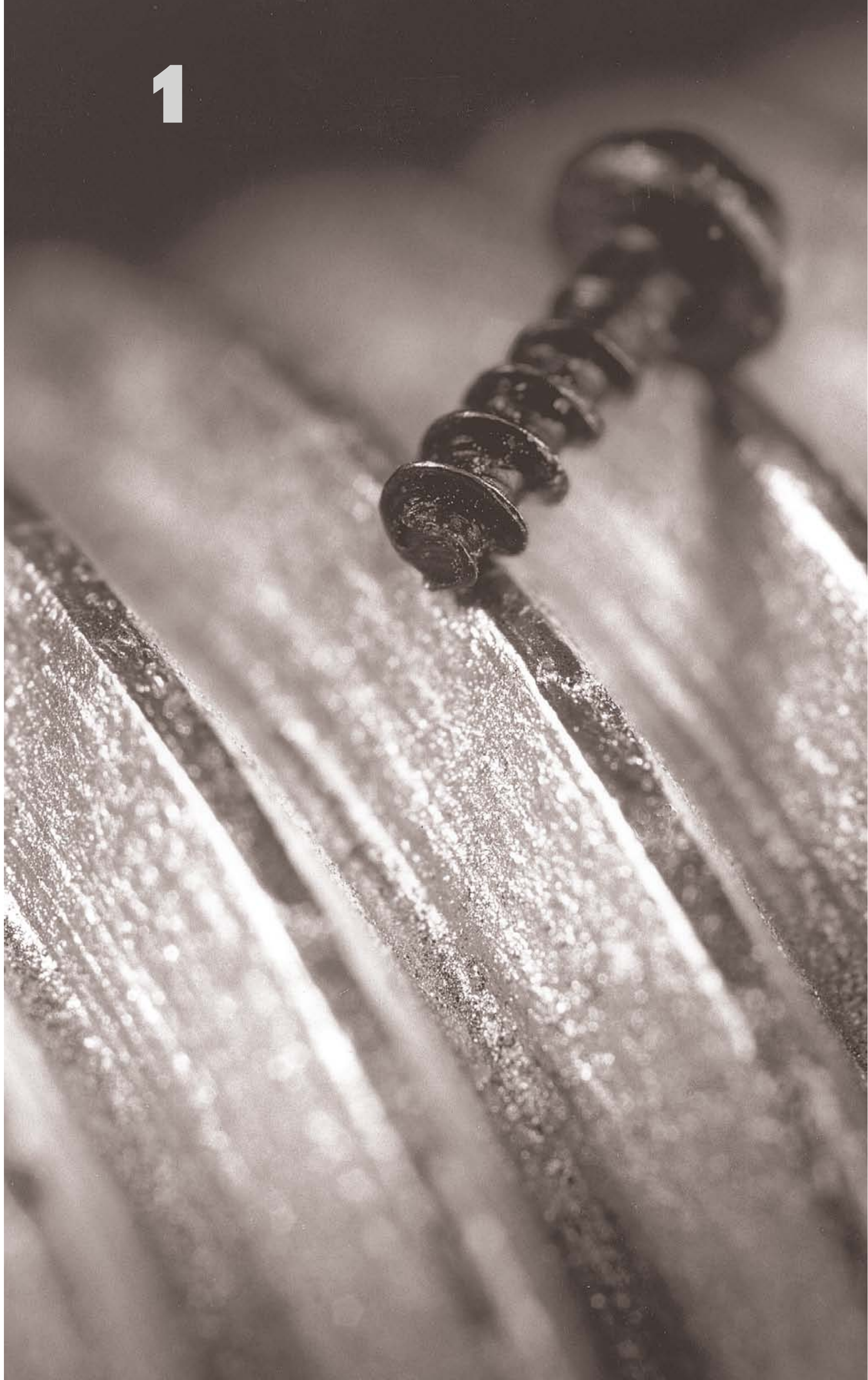
Our new handbook is intended to serve as a practical source of knowledge for everyone who requires information on the correct use of mechanical fasteners.

In order to make a bolted joint correctly, the right choices must be made in the list below:
• Type of bolt
• Strength
• Surface treatment
• Pretension
• In-place cost

We will deal with these factors and associated important data. The content of this publication is intended to cover most things that one needs to know in order to choose the correct product for a specific purpose. If that should not be sufficient, there are specialists in BULTEN who will willingly help you to solve any difficult problems in special cases.

Each chapter has been flagged in colour to make it easier to find one's way through the manual.

It is our hope that the reader, after having read all this wisdom, will not need to agree with the comment once made by a well-known purchasing manager, after he had inspected a screw and bolt factory. "I did not know much about nuts and bolts when I arrived today. And now I don't know much more about them either, but my ignorance is, anyway, on a higher level!"



1.2 DEFINITIONS

If we do not use the correct designations of objects and events in our surroundings, we can easily be misunderstood. A bolt is one such concept which even nowadays can lead to long discussions. Is it a pin, a coarse-threaded screw with nuts, a half-thread screw or ….?

Thanks to the efforts of the Swedish Centre for Terminology, TNC, since the 1950s, the following unambiguous definitions of screw and bolt are available to us, so everyone knows what we are talking about.

1.2.1 DEFINITION OF SCREW

A cylindrical or conical body with an external thread, intended for fixing, joining, transmission of force or transmission of movement in its longitudinal direction.

1.2.2 DEFINITION OF BOLT

A fastener consisting of a threaded pin or rod with a head at one end, designed to be inserted through holes in assembled parts and secured by a mated nut that is tightened by applying torque.

1.3 THE SI SYSTEM

In the early 1970s, a general transfer to the SI system was effected throughout the world. SI is an abbreviation of *Système International d'Unités*, i.e., the international unit measurement system – an internationally established system of measurement units built on older metric systems.
The transition was implemented in Sweden on 1 January 1976.

In a careful comparison between the SI system and the old technical system, the following apply:

1 N = 0.10197 kp
1 Nm = 0.101972 kpm
1 N/mm^2 = 0.101972 kp/mm^2

When the strength requirements for screws, bolts and nuts were reworked, in order to obtain even numbers, the relationship 1 N = 0.1 kp and 1 N/mm^2 = 0.1 kp/mm^2 was chosen, which means that the strength values increased by 2% compared with previous requirements.

Therefore, it is also easy to apply older tables of tightening torque and to use torque wrenches graded in kpm. The relationship is 1 kpm = 10 Nm.

Bolts and Nuts

A young schoolgirl once gave the following definitions of what she considered a bolt and a nut to be:

"A bolt is a kind of stick of hard metal, for example, iron, with an angular lump at one end and a long scratch wound around the other end. A nut is like a bolt, but it is the opposite, because it is a hole in an angular lump which is sawn off so that it is short, with wrinkles inside the hole".

ORDER FROM CHAOS

2

2.1 IN GENERAL

One of the fundamental laws of thermodynamics states that in open systems, everything strives towards maximum entropy, i.e., towards the highest level of disorder. Standardisation can therefore be perceived as an enclosed system, because within this area of activity one strives to achieve a maximum level of order.

A former CEO of GM once said, "For each dollar I spend on standardisation, I get back at least double". Something to think about for anyone who wants to find his own variations of a product.

2.2 SWEDISH AND INTERNATIONAL STANDARDISATION

In Sweden, the search for order began early. Back in the 1920s we had several national standards, including standards for threads, screws, bolts, nuts and similar machine elements.

"Harmony between nations is impaired if bolts and nuts have different threads. But harmony between nations is brighter with the standardisation of nuts and bolts."

Alf Henriksson

Sveriges Maskinindustriförenings Standardcentral – SMS (the Swedish Machine Industry Association's Standard Centre), the abbreviation of which lives on but now means *Sveriges Mekan- och Materialstandardisering* (the Swedish Mechanical and Material Standardisation) was the first national organ solely intended to produce order.

The organisation played a considerable role, and still does, within international standardisation, now expressed as ISO (the International Organisation for Standardisation) and CEN (*Comité Européen de Normalisation*).

2.3 SMS - SS - ISO - EN

These designations reflect the ongoing globalisation of standardisation work. As long as we worked out standards and set them at national level, the designations were national. Within the mechanical area, we used the designation SMS plus a number. Subsequently this was changed to SS = Swedish Standard.

Western European standardisation takes place within CEN – *Comité Européen de Normalisation* – which was founded in 1961. Work was intensified in 1985 through the New Approach work programme.

A committee for fasteners with the designation CEN/TC 185 started work in 1989 with a meeting in Berlin. Sweden has been a full member ever since CEN was founded. The EN standard is obligatory for the member states.

On the international level, work is conducted within ISO – the International Organisation for Standardisation –it was founded after the end of the Second World War and replaced the former organisation ISA – International Standards Association, founded in 1926.

The Swedish delegations and secretariat groups have often had to form mental bridges between major powers which are often at odds with one another and motivated by prestige. Among other things, this occurred during the so-called OMFS dispute (Optimum Metric Fastener System) during the 1970s, when there was a risk of two global but incompatible metric thread and screw systems materialising. Canada and the USA wanted to introduce major changes into the metric thread system and fasteners in general.

One of the companies which actively participated in founding the SMS during the 1920s was the Bultfabriks company as it then was. Ever since then, BULTEN has continued its involvement in achieving order from chaos by making experts and time available to the SMS.

Standards within EN are in most cases identical with the equivalent ISO standards. Such a standard is then designated EN-ISO and is published in Sweden as SS-EN ISO. If the standard is designated SS-EN, it conforms only with EN, if it is designated SS-ISO, it conforms only with ISO, and if it is only designated SS with a number, it is a purely Swedish standard without any international equivalent, which is quite unusual nowadays.

When an individual company wishes to review and revise its fastener standards, it should conform to the ISO or EN standard. If these are not included, one should conform to the German DIN standard, because that country is the secretariat country of CEN/TC 185, and thus has considerable influence as regards the proposals which are worked out.

The standardisation work within our product area is ongoing all the time. Greater understanding of the forces and stresses which arise in the use of threaded machine elements and new or changed types of material require continual monitoring and modification of standards and regulations. This is especially important within the internationalisation and development of trade within CEN and ISO. In this handbook, reference will often be made to such standards, and through easy contact with the staff of SMS, one can obtain full information on any current standard and up-to-date handbooks on various subjects.

2.4 TWO PROBLEMS

Within ISO, there were two particular problems which faced countries for a long time, and these were:

- what widths across flats should one have for a hexagonal head bolt or screw and hexagonal nut?
- how should regulations concerning strength be formulated?

2.4.1 THE WIDTHS ACROSS FLATS PROBLEM

During the OMFS dispute mentioned above, the USA and Canada wished to have smaller widths across flats in some bolt and screw sizes. The proposed widths across flats would harmonise better with the inch-threaded bolts and screws in use, and save much material and space. For example, it can be mentioned that for M10, on the European continent 17 mm had been used, in Sweden 15 mm, and in the USA 14.3 mm for a 3/8 inch-threaded screw

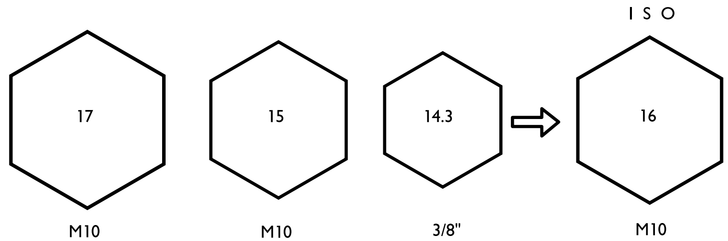


Figure 2.1

After almost endless discussions and reports, it was agreed that there would be three series, one "large", one "normal" and a series for flange bolts and screws and flange nuts.

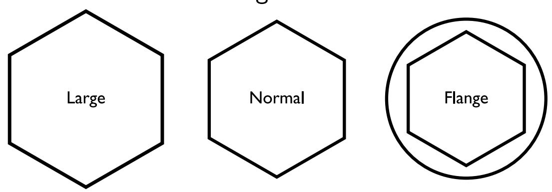


Figure 2.2

Changes in widths across flats of hexagonal head bolts and screws and hexagonal nuts:

Thread	Width across flats in mm	
d	Old	New
M10	17	16
M12	19	18
M14	22	21
M22	32	34

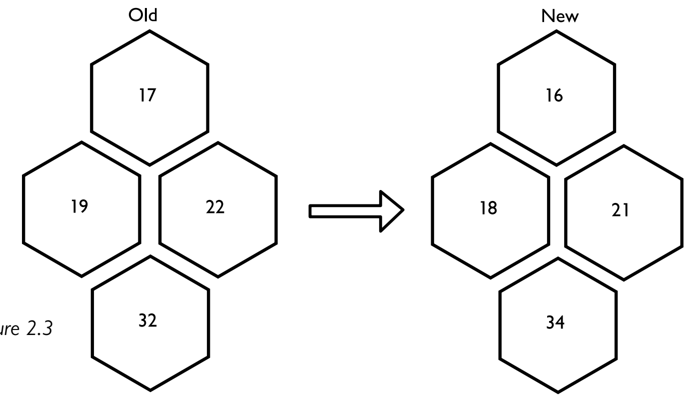


Figure 2.3

2.4.2 THE STRENGTH PROBLEM

The investigation work was carried out within a special working group, ISO/TC2/SC1 – Mechanical properties, with participation from most of the trend-setting industrial countries. In due course, international regulations for strength classes were arrived at, together with marking and testing. The extensive investigations which were carried out gave improved knowledge of tensile stress relationships in a bolt/nut joint as new and more effective tightening methods were employed. This led to an extensive revision of nut heights, as it became evident that the old rule that a nut height should be 0.8 times the thread diameter was no longer sufficient to guarantee the nut being stronger than the bolt. The new investigation results led to a change in the strength and height of nuts, which is explained in Section 5.

Unfortunately, it does happen that some suppliers refer to old nut standards, for example DIN 934, which are both out-of-date and incorrect, and also withdrawn. The same market phenomena occur if one receives an order for a bolt or screw in accordance with DIN 931 or DIN 933, as these have been withdrawn and discontinued.

A good piece of advice is therefore always to refer to international standards in accordance with EN or ISO when ordering, and also to check that you get what you have ordered when it is delivered.

It can be pointed out that the ability of the bolt to resist axial tensile traction is reduced by 15% during tightening due to the torque caused by thread friction. When the bolted joint was subjected to the maximum yield stress of the bolt, it became evident that thread chafing could occur both in the bolt and in the nut. Because for practical reasons, it would be preferable for the bolt to break rather than have a stripped nut thread, which is less easy to see; this required that the nut height should be increased. The strength, test and marking regulations which now apply in Swedish Standard are in full accord with standards according to EN or ISO. Compared with old standards, it is now evident that the property classes 6.6, 6.9, and 14.9 have ceased to exist, while one class, 9.8, has been added for thread sizes up to M16, but this class is not normally used in Europe. This was a concession for the USA.

As regards the new nut standard, there is also now international conformity. Two types have been introduced, in which Type 1 has a height of roughly 0.9 x d and Type 2 has a nut height which is about 10% higher than Type 1. The first of these is intended for nut strength classes 4, 5, 6, 8, 10 and 12 (up to M16) in combination with specific mechanical properties. Type 2 is used for classes 8, 9 (which is new) and 12, but then in combination with other mechanical properties. Class 9 is not normally used in Europe, and class 14 has been discontinued. For lower nuts, two new classes, 04 and 05, have been added. As with bolts and screws, regulations up to M39 apply, and are also recommended for larger dimensions.

For normal hexagonal head screws, bolts and nuts, strength requirements apply in accordance with the following standards:
SS-EN 20898-1 and
SS-EN 20898-2 respectively

2.5 HEXAGONAL HEAD BOLTS AND SCREWS AND HEXAGONAL NUTS

When the basic standard was ready, work began on working out the actual product standards.
Three product grades were introduced, designated A, B and C.
A = tolerance “fine” (Tolerance Class 1)
B = tolerance “medium” (Tolerance Class 1G)
C = tolerance “coarse” (Tolerance Class 2)

Five international standards have been issued for hexagonal head bolts and screws with metric coarse threads: ISO 4014 up to and including ISO 4018. Five international standards have also been issued for hexagonal nuts with metric coarse threads: ISO 4032 up to and including ISO 4036.

2.6 IMPORTANT CHANGES

Some of the most important changes in relation to earlier standards are:

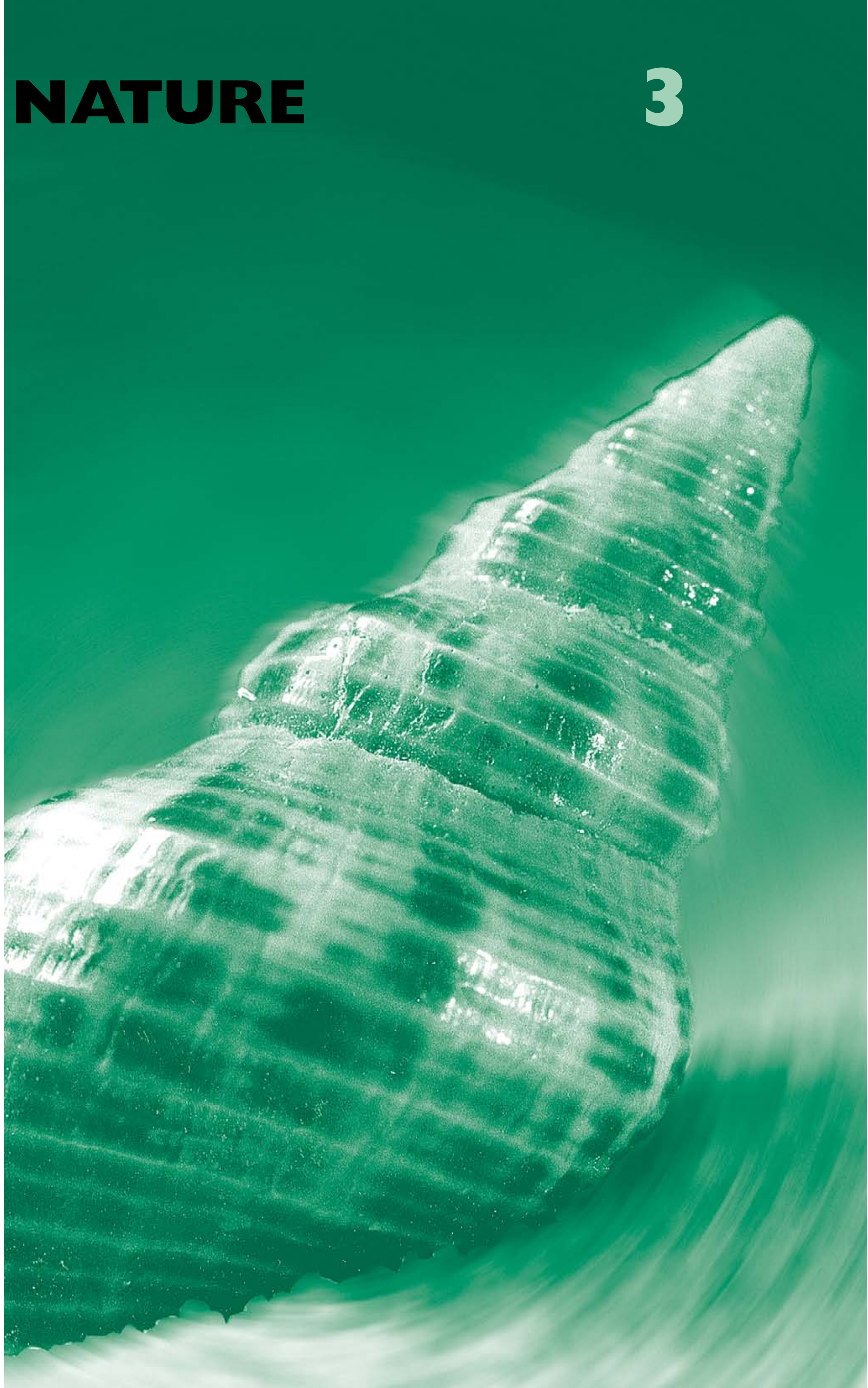
- changed limit values and materials as regards the property classes of bolts and screws, see Section 4.2
- new widths across flats for thread sizes M10, M12, M14 and M22, see Section 2.4.1
- the division of hexagonal nuts into two types, with new nut heights and revised property class requirements. See Section 5.2
- change of the angle of countersinking for bolts and screws with countersunk heads from 82 degrees to 90 degrees
- new standards for cross recesses, see Section 12.8.6
- revised standards for self-tapping screws with ST threads, see Sections 3.8 and 4.3.

2.7 OTHER FASTENER STANDARDS

Up to the present, over 200 ISO standards have been worked out for threaded fasteners. In most cases, they have been transferred to the EN ISO standard or the Swedish SS ISO standard. More detailed information can be obtained from SMS in Stockholm. As regards self-tapping screws with countersunk heads and ST threads, the countersink angle of 90 degrees has applied from 1 January 1989.

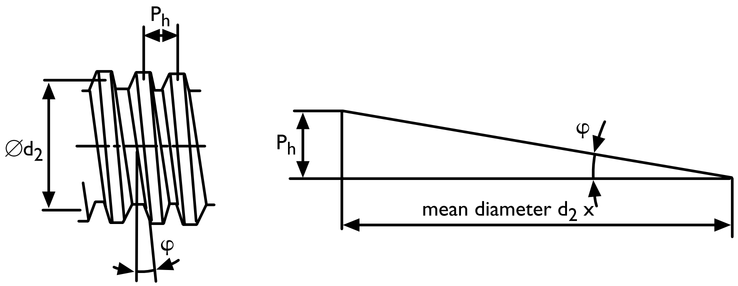
Figure 2.4

THE LINE GIVEN BY NATURE 3



3.1 IN GENERAL

If you wind a right-angled, wedge-shaped triangle around a cylinder with a circumference roughly equal to the long side of the triangle, you get an idea of the thread spiral and its pitch.



The angle of pitch j is calculated by trigonometry using the pitch Ph as one of the short sides of the triangle and the circumference of the thread as the other short side of a right-angled triangle.

Figure 3.1

This was the form of spiral which constituted the basis of a method of transmission of force through the "invention" of threaded machine elements.

The history of the development of the thread from about the 4th century BC will not be related here, there is a great deal of literature about it. It will be sufficient to mention that nowadays we have managed to unite many conflicting desires into a single international system of metric threads and thread tolerances, including how they shall be controlled. This became the ISO standard, which has been described as the most important standard of our time.

In order to be able to guarantee the correct fit of thread between an internal thread and an external thread, their dimensions must remain within certain set limits. It is necessary to agree on a system of tolerances and fits, which are described below.

It should also be pointed out that in the construction of this international thread tolerance system, Sweden has had a decisive influence on its development ever since the beginning, and has succeeded in uniting German, Russian and American interests into a completely new system, acceptable without loss of prestige.

In Table 3.1, you can see the ISO's general plan of diameters and pitch (previously called lead), both coarse and fine. Combinations in accordance with preference 1 should be selected first of all, preference 2 can be used in the case of doubt, and preference 3 should be avoided completely.

TABLE 3.1
METRIC ISO THREADS - ISO GENERAL PLAN

Thread major diameter			Pitch (P)										
Preference			coarse	fine									
1	2	3		3	2	1.5	1.25	1	0.75	0.5	0.35	0.25	0.2
1			0.25										0.2
	1.1		0.25										0.2
1.2			0.25										0.2
	1.4		0.3										0.2
1.6			0.35										0.2
	1.8		0.35										0.2
2			0.4									0.25	
	2.2		0.45									0.25	
2.5			0.45								0.35		
3			0.5								0.35		
	3.5		0.6								0.35		
4			0.7							0.5			
	4.5		0.75							0.5			
5			0.8							0.5			
		5.5	–							0.5			
6			1						0.75				
		7	1						0.75				
8			1.25					1	0.75				
		9	1.25					1	0.75				
10			1.5				1.25	1	0.75				
		11	1.5				–	1	0.75				
12			1.75			1.5	1.25	1					
	14		2			1.5	1.25 [1)]	1					
		15	–			1.5		1					
16			2			1.5		1					
		17	–			1.5		1					
	18		2.5		2	1.5		1					
20			2.5		2	1.5		1					
	22		2.5		2	1.5		1					
24			3		2	1.5		1					
		25	–		2	1.5		1					
		26	–		–	1.5		–					
	27		3		2	1.5		1					
		28	–		2	1.5		1					
30			3.5	(3)	2	1.5		1					
		32	–	–	2	1.5							
	33		3.5	(3)	2	1.5							
		35 [2)]	–	–	–	1.5							
36			4	3	2	1.5							
		38	–	–	–	1.5							
	39		4	3	2	1.5							

[1)] Only for spark plugs.
[2)] Only for roller-bearing structures.

Pitch 3 for diameters 30 and 33 should be avoided.

CONTINUATION OF TABLE 3.1

Thread major diameter			Pitch (P)					
Preference			coarse	fine				
1	2	3		6	4	3	2	1.5
		40	–		–	3	2	1.5
42			4.5		4	3	2	1.5
	45		4.5		4	3	2	1.5
48			5		4	3	2	1.5
		50	–		–	3	2	1.5
	52		5		4	3	2	1.5
		55	–		4	3	2	1.5
56			5.5		4	3	2	1.5
		58	–		4	3	2	1.5
	60		5.5		4	3	2	1.5
		62	–		4	3	2	1.5
64			6		4	3	2	1.5
		65	–		4	3	2	1.5
	68		6		4	3	2	1.5
		70		6	4	3	2	1.5
72				6	4	3	2	1.5
		75		–	4	3	2	1.5
	76			6	4	3	2	1.5
		78		–	–	–	2	–
80				6	4	3	2	1.5
		82		–	–	–	2	
	85			6	4	3	2	
90				6	4	3	2	
	95			6	4	3	2	
100				6	4	3	2	
	105			6	4	3	2	
110				6	4	3	2	
	115			6	4	3	2	
	120			6	4	3	2	
125				6	4	3	2	
	130			6	4	3	2	
		135		6	4	3	2	
140				6	4	3	2	
		145		6	4	3	2	
	150			6	4	3	2	
		155		6	4	3		
160				6	4	3		
		165		6	4	3		
	170			6	4	3		
		175		6	4	3		
180				6	4	3		
		185		6	4	3		
	190			6	4	3		
		195		6	4	3		
200				6	4	3		
		205		6	4	3		
	210			6	4	3		
		215		6	4	3		
220				6	4	3		
		225		6	4	3		
		230		6	4	3		
		235		6	4	3		
	240			6	4	3		
		245		6	4	3		
250				6	4	3		
		255		6	4			
	260			6	4			
		265		6	4			
		270		6	4			
		275		6	4			
280				6	4			
		285		6	4			
		290		6	4			
		295		6	4			
	300			6	4			

3.2 THREAD TOLERANCE SYSTEMS AND RECOMMENDED TOLERANCE CLASSES FOR METRIC ISO THREADS

3.2.1 BASIC PROFILE

The basic profile – for external and internal threads – is a joint theoretical profile to which a tolerance is attributed.

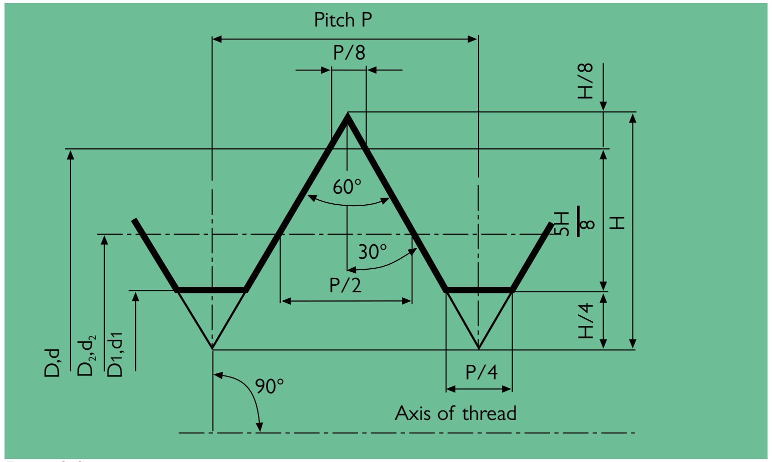


Figure 3.2

D1	= D –1.0825 P	H	=0.86603 P
d1	= d –1.0825 P	H/4	=0.21651 P
D_2	= D –0.6495 P	H/8	=0.10825 P
d_2	= d –0.6495 P	$\frac{5H}{8}$	= 0.54127 P

D = Major diameter of internal thread, basic dimension.
D1 = Minor diameter of internal thread, basic dimension.
D_2 = Pitch diameter of internal thread, basic dimension.
P = Pitch (lead).
d = Major diameter of external thread, basic dimension.
d1 = Minor diameter of external thread, basic dimension.
d_2 = Pitch diameter of external thread, basic dimension.
H = Fundamental triangle height.

3.2.2 LEAD BECOMES PITCH

Previously in Sweden, one always called the distance between two thread peaks of the same thread groove *stigning*, designated by s. The designation pitch was then used for the distance between two adjacent thread tops that had more than one start.

In connection with the international standardisation, this had to be changed, and *stigning* was called pitch, and the designation P (pitch) was introduced instead of s. The SS 1702 and ISO 5408 standards define most concepts of threads. Below are given the definitions of pitch P, lead P_h and angle of pitch φ.

Pitch P
The distance along the pitch line between two successive identically directed flanks, Fig. 3.3.

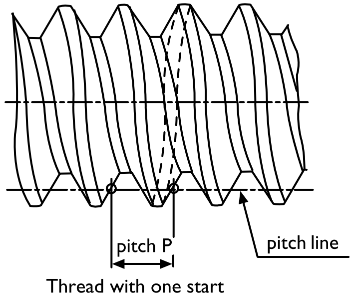


Figure 3.3

Lead P_h
The distance along the pitch line between two successive identical flanks of the same thread groove, Fig. 3.4.

The lead is a multiple of the pitch, i.e., P_h = z x P where z = the number of starts.

In the case of threads with only one start, z = 1 the lead and pitch are the same. In Swedish Standard, the word "*stigning*" (lead) is used most often in this case, and is then designated P.

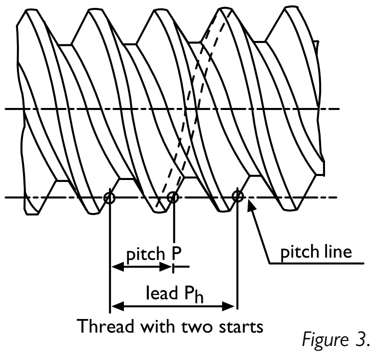


Figure 3.4

Angle of pitch φ
The angle between the tangent of a screw line and an axis at right-angles to the plane of the screw line, Figure 3.1.

The angle of pitch of an external thread grows from the thread top towards the thread root. If the position of the angle of pitch is not given exactly, the angle of pitch is gauged at the pitch line.

3.2.3 TOLERANCE SYSTEM

The system described in SS-ISO 965 1-5 consists of a combination of degrees of tolerance, denoting the extent of the tolerance range (the length of stem which increases with higher figures in Figure 3.5) and the position of tolerance, which denotes the placing of the tolerance area in relation to the nominal thread profile (the shortest distance between a stem and basic dimension in Figure 3.5).

a) *Degrees of tolerance:*

Diameters	Degrees of tolerance
Minor diameter (top diameter) internal threads (D_1)	4, 5, 6, 7, 8
Major diameter (top diameter) external threads (d)	4, 6, 8
Pitch diameter internal threads (D_2)	4, 5, 6, 7, 8
Pitch diameter external threads (d_2)	3, 4, 5, 6, 7, 8, 9

b) *Tolerance:*
G and H for internal threads; e, f, g and h for external threads. Tolerance is adapted to current coating thicknesses and to whatever is required for simple mounting.

c) A selection of combinations of degrees and levels (tolerance classes) which give the tolerance qualities normally used of Fine, Medium and Coarse for the depths of engagement of Short, Normal and Long. There is no connection between the tolerance classes of the thread tolerance system and ISO's tolerance system for smooth axles and holes.

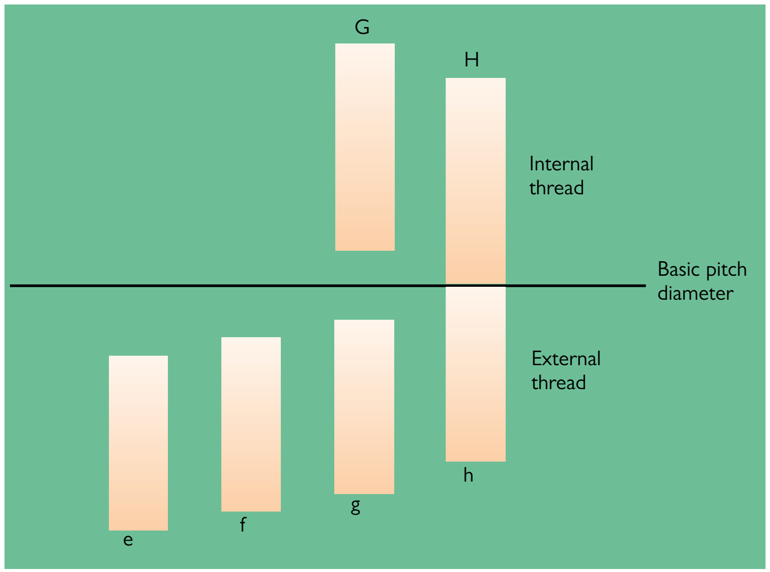


Figure 3.5

3.3 COATED THREADS

For coated threads, the tolerances apply prior to coating unless otherwise stated. After coating, the actual thread profile should not be less or more respectively at any point than the maximum material limit for H and h respectively.

The tolerances are adapted to current coating thicknesses and to whatever is required for simple mounting.

3.3.1 TOLERANCES FOR COATED THREADS

Below is shown a section through the top of a screw's thread.

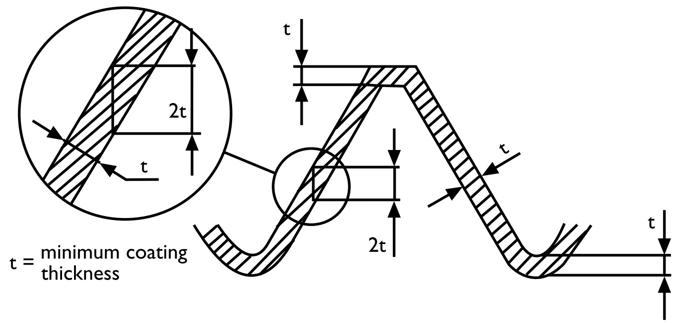
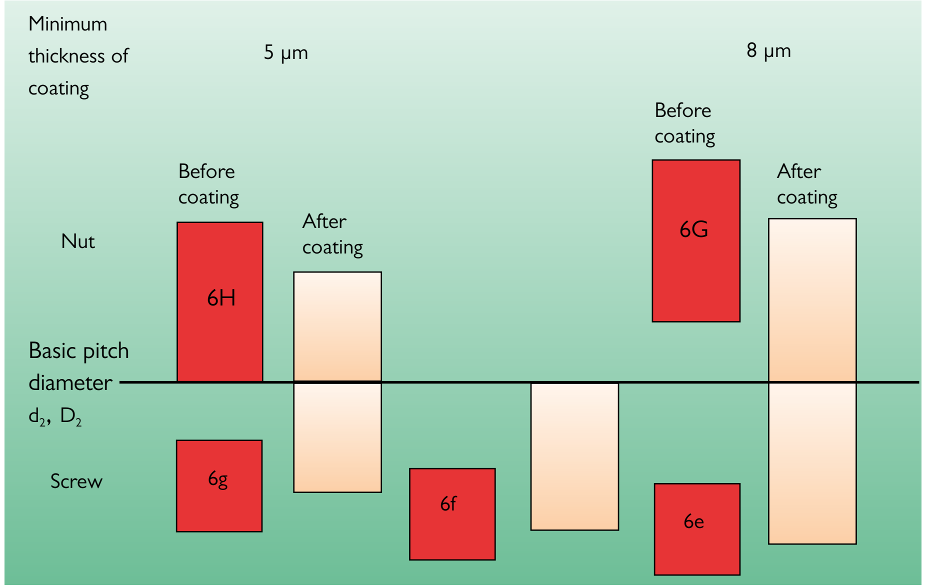


Figure 3.6

The hatched area is the coating. When threads are coated, space for the coating must be created. This is effected through the basic profile of the thread being displaced. In the case of zinc electroplating, the layer of zinc is not evenly distributed on the parts of an item. The basic profile of the thread must therefore be adjusted so that even the most unfavourable case is accommodated. If the minimum thickness of coating is t and the maximum thickness of coating is 2t, the required adjustment of diameter for the inner and major diameters is 4t and for the pitch diameter, 8t.

Appropriate thread tolerance classes for coating thicknesses of 5 µm and 8 µm are shown in Figure 3.7. The tolerance class required for any particular thickness of coating depends on the diameter of the thread in such a way that coarse diameters automatically give greater basic play. ISO 4042 provides more information about threads and zinc electroplating.



Appropriate thread tolerance classes for coating thicknesses of 5 µm and 8 µm.
Figure 3.7

For thicker coatings, for example, hot dip galvanizing, considerably greater thread play is required, which is created through having the external thread under-dimensioned.

According to SS 3192, the minimum layers for hot dip galvanizing are:
25 mm for M6 – M8
45 mm for M10 – M22
65 mm for M24 – M68

The dimensions of threads before hot dip galvanizing are given in SS 3193. Under-dimensioning is 220 µm, 300 µm and 380 µm for each respective thickness as set out above.

Despite the fact that we have standards SS 3192 and SS 3193, it is common that DIN 267 T10 is applied, giving roughly the same under-dimension.

Since 1998-12-15 ISO 965 replaced the above.

3.4 THREAD DESIGNATIONS

A full thread designation consists of a designation of the thread system and size, and designations of the thread's tolerance class at pitch diameter and top diameter.

Each class designation consists of:
- a figure indicating the degree of tolerance.
- a letter indicating the position of tolerance, capital letter for internal threads, lower case letter for external threads.

If the two class designations for one and the same thread are the same, it is not necessary to repeat the designations.

3.4.1 EXTERNAL THREADS

Examples of designations of external threads

M6 - 5g 6g

Thread designation
Class designation for pitch diameter tolerance
Class designation for top diameter tolerance

M6 - 6g

Thread designation
Class designation for median and top diameter tolerance

3.4.2 INTERNAL THREADS

Example of designations of external threads

For internal threads, as a rule, the same tolerance class is used for pitch diameter and top diameter.

M6 - 6H

Thread designation
Class designation for median and top diameter tolerance

3.4.3 THREAD FITS

Examples of designations of fits

A fit between threaded parts is denoted with the tolerance class of the internal thread followed by the tolerance class of the external thread, separated by a forward-slash.

M6 - 6H/6g M20x2 - 6H/5g6g

3.5 LENGTH OF ENGAGEMENT

The length of engagement is the axial measurement within which the external and internal threads have theoretical contact.
The depths of engagement are classed in three groups: S (Short), N (Normal) and L (Long). Group N is recommended when the actual length of engagement is unknown.
For group N, the following empirical formulae apply:

$N_{min} = 2.24P \times d^{0.2}$ $N_{max} = 6.7P \times d^{0.2}$

P is the pitch and d is the minimum standardised thread diameter of this pitch according to SS-ISO 965/1.

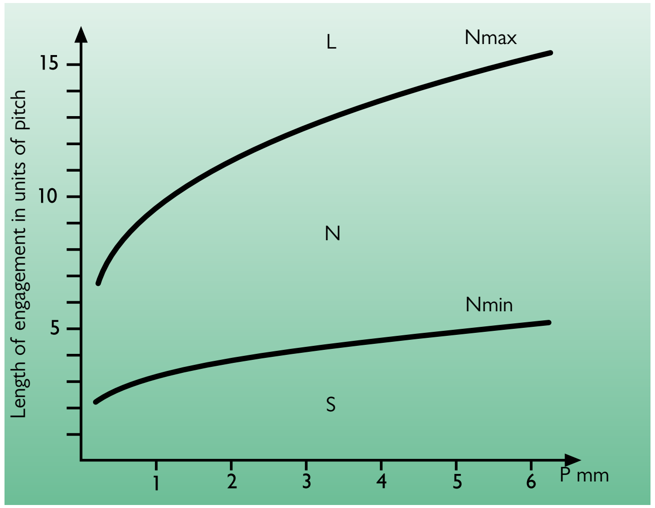


Figure 3.8

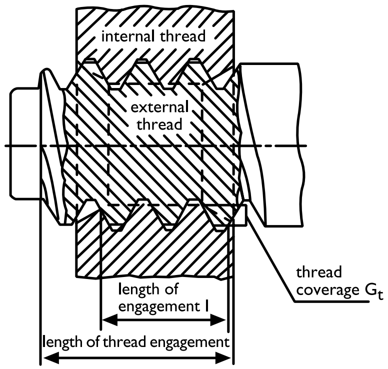


Figure 3.9

3.6 RECOMMENDED TOLERANCE CLASSES

A tolerance class should preferably be chosen from tables 3.2 and 3.3.

- **First of all, tolerance classes denoted in bold are selected.**
- in the second place, tolerance classes denoted in normal typeface are selected
- in the third place, tolerance classes denoted in normal typeface in brackets are selected; they should be avoided.

The tolerance classes in a box have been selected for threads for commercial screws, bolts and nuts. The following general rules apply for selection of tolerance quality:

- **Fine**: for precision threads when only a small variation in the character of fit is required.
- **Medium:** for general use.
- **Coarse:** for cases in which manufacturing difficulties can arise, for example, in threading hot-rolled rods and long holes

Any of the tolerance classes for internal threads can be combined with any of the tolerance classes for external threads. In order to guarantee sufficient thread coverage, the internal and external threads should form the fits H/g, H/h or G/h.

TABLE 3.2
RECOMMENDED TOLERANCES CLASSES FOR INTERNAL THREADS

Tolerance quality	Tolerance level G			Tolerance level H		
	S	N	L	S	N	L
Fine				4H	5H	6H
Medium	(5G)	**6G**	(7G)	**5H**	$\boxed{\textbf{6H}}$	**7H**
Coarse		(7G)	(8G)		7H	8H

TABLE 3.3
RECOMMENDED TOLERANCE CLASSES FOR EXTERNAL THREADS

Tolerance quality	Tolerance level e			Tolerance level f			Tolerance level g			Tolerance level h		
	S	N	L	S	N	L	S	N	L	S	N	L
Fine								(4g)	(5g4g)	(3h4h)	**4h**	(5h4h)
Medium		**6e**	(7e6e)		**6f**		(5g6g)	$\boxed{\textbf{6g}}$	(7g6g)	(5h6h)	6h	(7h6h)
Coarse		(8e)	(9e8e)					8g	(9g8g)			

3.7 SELECTION OF THREAD TOLERANCE QUALITY

3.7.1 STRENGTH

Tests carried out so far have not been able to show any substantial difference in strength between threads produced in the various thread tolerance qualities that are currently available. For threads subjected to fatigue, exactitude of form and evenness of surface are of greater importance for strength than the pure variations in diameter.

Threads produced in the most coarse thread tolerance classes, for example, for hot dip galvanizing, end up with a reduced thread coverage, which one should take into account. This has been done in DIN 267 T10 by reducing the failure loads by 8% compared with SS-EN 20898-1.

3.7.2 MANUFACTURE

Manufacturing costs increase with reduced tolerance range, as shown in Table 3.4.

TABLE 3.4 THREAD TOLERANCE QUALITY COARSE, MEDIUM AND FINE MANUFACTURING ACCURACY AND NECESSITY

	Thread tolerance quality		
	coarse	medium	fine
Required manufacturing accuracy	Good. Controlled manufacture necessary	Considerable. Great control of manufacture necessary	Very great. Especially careful manufacturing and control methods necessary
Necessity for tolerance quality	Items with long depths of engagement. Items manufactured from substances of considerable variation of dimension.	Play in tolerance class 6g is considered advantageous.	Technically well motivated reasons for this tolerance quality are lacking in most cases. The tolerance quality should be avoided.
Notes	Usual tolerance quality for commercial elements produced from hot rolled initial material. Difficulties can arise in maintaining the major diameter tolerance of the screw thread.	The most common tolerance quality for commercial elements.	The tolerance quality is only applied to a limited extent for commercial elements.

3.7.3 ASSEMBLY

Bolt and nut threads manufactured in different tolerance qualities can be assembled together, in accordance with Table 3.5.

TABLE 3.5 THREAD TOLERANCE QUALITY COARSE, MEDIUM AND FINE PLAY AND THREAD FIT CHARACTER

	Thread tolerance quality coarse	medium	fine
Nominal play	Yes	Yes	No
Variation in fit character	Considerable (1.6)	Normal (1)	Very minor (0.63)
Notes	Burrs, handling damage (dents), etc., can be taken up by play. Rapid series assembly is possible. The nominal play is advantageous when a special lubricant is used in assembly, for example, for high-pressure vessels, tubular steel flanges, fitting and assembly parts and valves for high temperatures and high pressures.		Can make series assembly difficult.

The figures in brackets give the relative relationship in fit character. The coarse tolerance class has a 60% greater thread play than the normal, while the fine class is 37% "tighter" than the normal, which is given as 1.

3.8 TAPPING SCREWS WITH ST-THREADS

The dimensions of threads and screw tips are given in SS-EN ISO 1478. Previously, the designation B-threads was used for these threads. Translation between the two systems is given in Table 3.6 in order to facilitate comparison between old and new.

TABLE 3.6
TRANSLATION BETWEEN SYSTEMS ST AND B

ST2.2	ST2.9	ST3.5	ST4.2	ST4.8	ST5.5	ST6.3
B2	B4	B6	B8	B10	B12	B14

Type C screw tip
(previously Type AB)

Type F flat front end
(previously Type B)

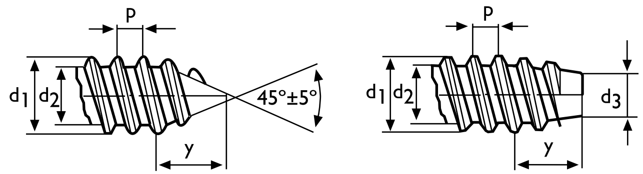


Figure 3.10

Type R rounded end

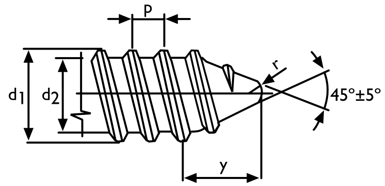


Figure 3.11

TABLE 3.7
DIMENSIONS OF ST-THREADS, MM

Thread	ST 2.2	ST 2.9	ST 3.5	ST 4.2	ST 4.8	ST 5.5	ST 6.3
P ≈	0.8	1.1	1.3	1.4	1.6	1.8	1.8
d_1 max	2.24	2.9	3.53	4.22	4.8	5.46	6.25
min	2.1	2.76	3.35	4.04	4.62	5.28	6.03
d_2 max	1.63	2.18	2.64	3.10	3.58	4.17	4.88
min	1.52	2.08	2.51	2.95	3.43	3.99	4.70
d_3 max	1.47	2.01	2.41	2.84	3.30	3.86	4.55
min	1.37	1.88	2.26	2.69	3.12	3.68	4.34
y ref [1)]							
Type C	2	2.6	3.2	3.7	4.3	5	6
Type F	1.6	2.1	2.5	2.8	3.2	3.6	3.6
Type R	–	–	2.7	3.2	3.6	4.3	5

1) Length of incomplete thread.

A particular screw dimension with thread ST 3.5 and length 19 mm is designated: ST 3.5 x 19.

3.9 ADDITIONAL INFORMATION

Kuturgeschichte der Schraube, by Wilhelm Treue.
SMS handbook 520, latest edition.
Swedish Standard SS-ISO 724.
SS-ISO 965-1 to 5.
SS-EN ISO 1478

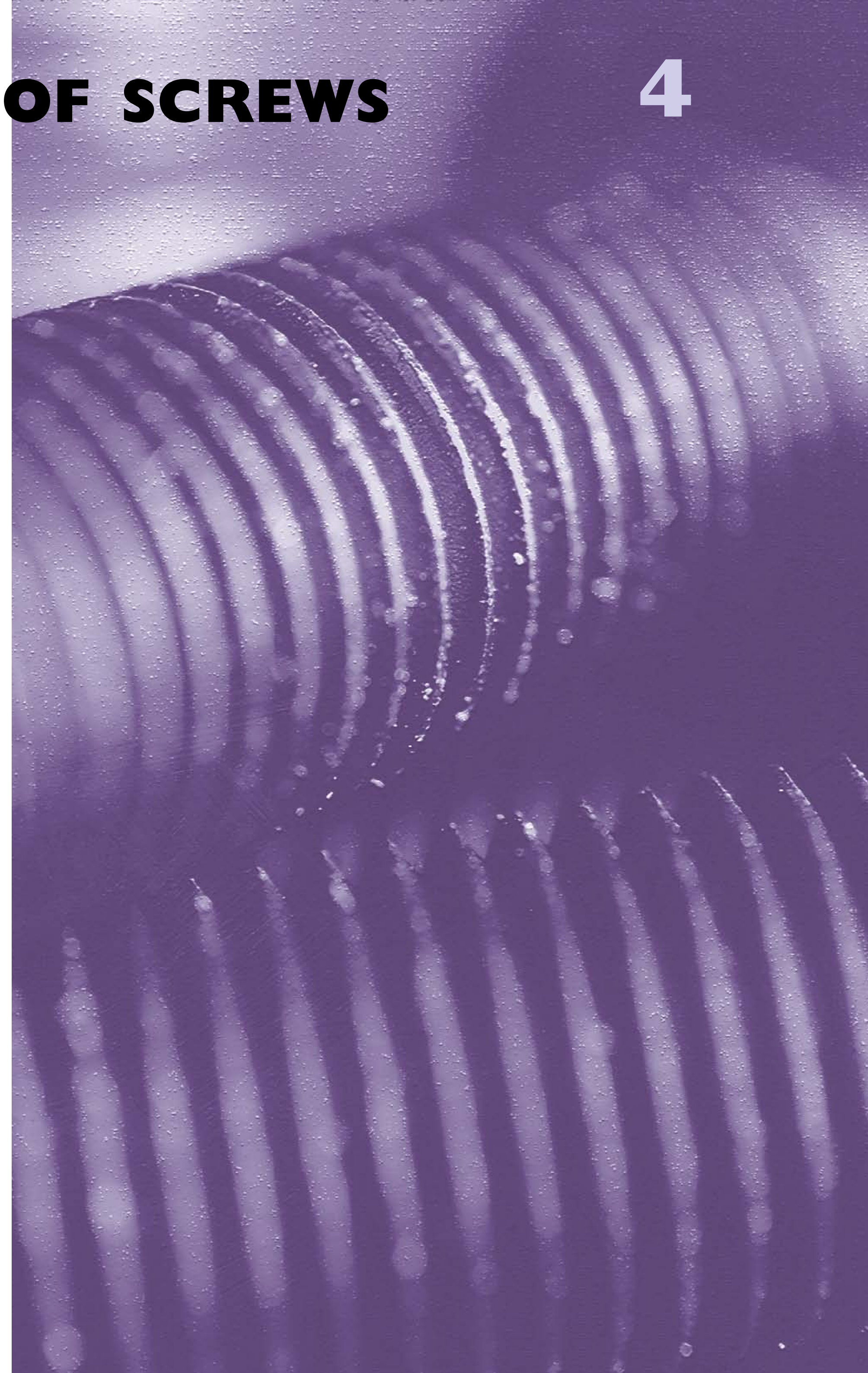
ON THE STRENGTH OF SCREWS

4

4.1 IN GENERAL

Knowledge of the strength of a fastener and the forces it can transfer is decisive in the selection of the right product. Previously, each country had its own regulations as regards choice of material, strength requirements, markings and testing methods. So it was not possible to know whether a screw manufactured in England was as strong as its German counterpart.

This began a long time ago during the time of the so-called farmer's screw. That was the time when screws were made of "farmers' iron", when they were as soft as chewing gum and were so weak that they could be pulled out by hand, at least up to M10, but at that time thread W 3/8" was used. At that time, screws were called D40, D60, D80, etc., in which the D denoted tensile traction and the figure indicated the fracture load, expressed in kp/mm^2, calculated on the so-called core area of the thread, where the inner or core diameter of the thread was used in working out the figure. Then the cold-heading procedure was introduced, and through strain hardening it was possible to obtain a simple form of carbon steel St 37, from which could be produced screws with a breaking strength of 60 kp/mm^2 at the price of a very slight elongation, but rather greater propensity for brittle fracture. At that time, all countries had their own strength and testing regulations, all more-or-less different from one another. So you had to know the rules when buying screws from different places. An 8G screw from Germany was not the same as a D80 in Sweden or a grade 5 from the USA.



After a time, it became obvious that it would be practical to agree on common strength standards, common classification and identification of fasteners which could be standardised and sold from one country to another. To begin with, agreement was reached on the area over which one should calculate stress. After a great number of reports and even more meetings, agreement was reached on the concept of nominal stress area and a formula for calculating it.

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

where d_2* is the basic dimension of the median diameter of the external thread;

d_3 is the inner diameter of the external thread

$$d_3 = d_1 - \frac{H}{6}$$

where d_1* is the basic dimension of the inner diameter of the external thread;

H is the fundamental triangle height of the thread.

* See ISO 724

The next step was to determine what the physical property classes should be called and what data should apply for them. Therefore it was necessary to discuss and agree on the types of material appropriate for the various classes and thread sizes. The international committee which had the responsibility for investigating all these rather intricate and often infected problems was called, and is still called, ISO/TC 2-Fasteners. It is not necessary here to describe the uproar which occasionally occurred during these international meetings, but sufficient to say that in 1967, an internationally accepted system was agreed on. After regular revisions were carried out, there is now a system of fasteners of metric threads based on ISO's standards, confirmed within CEN and adopted by most European countries.

4.2 PHYSICAL PROPERTY REQUIREMENTS FOR FASTENERS WITH METRIC ISO THREADS

Extract from the international standard prEN ISO 898-1 which applies to physical property requirements for bolts, screws and studs. It is valid since April 1999.

4.2.1 EXTENT AND APPLICATION

The prEN ISO 898-1 standard applies to mechanical properties of bolts, screws and studs in testing in the temperature range of 10°C – 35°C. The properties vary at higher and lower temperatures.

The prEN ISO 898-1 standard applies to bolts, screws and studs:
- with coarse pitch threads M1.6 up to M39 and fine pitch threads M8x1 up to M39x3
- with triangular ISO threads in accordance with ISO 68
- with diameter/pitch combinations in accordance with ISO 261 and ISO 262
- with thread tolerances in accordance with ISO 965-1 and 965-2
- of non-alloy steel or alloy steel

This does not apply to stop screws and similar fasteners (see SS-ISO 898-5).

It does not specify requirements in respect of properties such as
- weldability
- corrosion resistance (see ISO 3506)
- ability to resist temperatures higher than +300°C (+250°C for 10.9) or lower than -50°C
- strength to resist shearing
- fatigue strength.

4.2.2 DESIGNATION SYSTEM

The designation system for property classes of bolts, screws and studs is given in Table 4.1.

The x axis gives the value of nominal ultimate tensile strength R_m in N/mm^2, and the y axis gives the value of elongation after fracture A_{min} in percentage.

The property class designation consists of two figures:
- the first gives 1/100 of the nominal ultimate tensile strength in N/mm^2 (see R_m in Table 4.3).

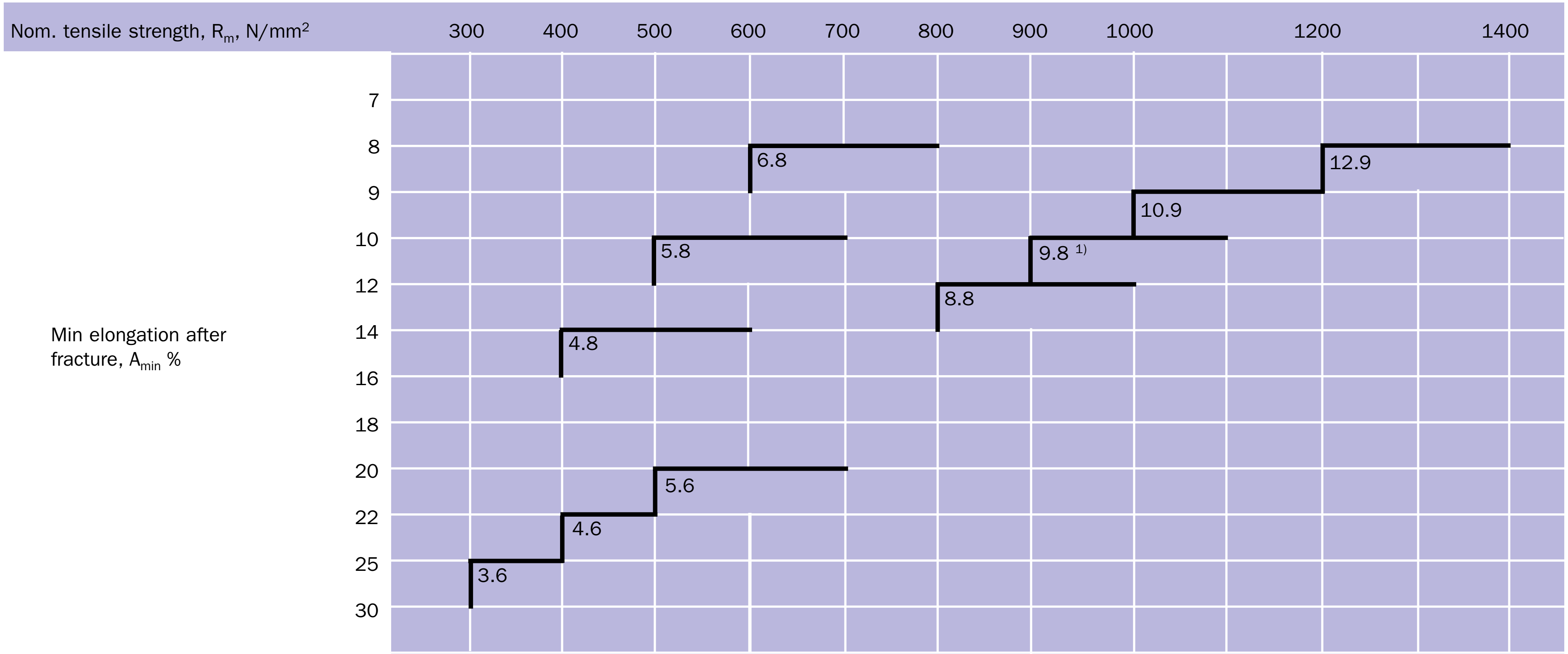
- the second figure gives the relationship in tenths between the lower yield point R_{eL} (or the stretch yield point $R_{p0.2}$) and nominal ultimate tensile strength R_m (the yield point relationship).

Multiplication of the two figures gives the yield point in N/mm^2.

The lower yield point R_{eL} (or the stretch yield point $R_{p0.2}$) and minimum ultimate tensile strength R_m are the same as, or more than, the nominal values (see Table 4.3).

Recommendation:
Always refer to international ISO or EN standards when purchasing, and check that these regulations have been conformed to by the supplier. Unfortunately, there are many examples of cheating with test certificates and quality certificates.
Therefore ensure that the fasteners you receive conform to what was promised!

TABLE 4.1
COORDINATE SYSTEM OF PROPERTY CLASSES



Relationship between yield stress and tensile strength				
The second figure of the designation		.6	.8	.9
$\dfrac{\text{Lower yield stress } R_{eL}{}^{2)} \text{ or proof stress } R_{p0.2}}{\text{Nominal tensile strength } R_m} \times 100$	%	60	80	90

1) Applies only to d ≤ 16 mm thread diameter.

2) Nominal values according to Table 4.3 apply.

Note. Despite a large number of property classes being specified in prEN ISO 898-1, this does not mean that all classes are suitable for all products. Further information concerning the application of specific property classes is given in the respective product standards. For products which are not standardised, it is advisable to follow what has been already done for similar standardised products as closely as possible.

4.2.3 **MATERIAL**

Table 4.2 specifies steel and tempering temperatures for the various property classes of bolts, screws and studs. The chemical composition shall be set in conformity with applicable ISO standards.

TABLE 4.2 STEEL

Property class	Material and thermal treatment	Chemical composition (control analysis) % C min	C max	P max	S max	B [1] max	Tempering temperature °C min
3.6 [2]	Carbon steel	–	0.20	0.05	0.06	0.003	–
4.6 [2] 4.8 [2]		–	0.55	0.05	0.06	0.003	–
5.6		0.13	0.55	0.05	0.06	0.003	–
5.8 [4] 6.8 [4]		–	0.55	0.05	0.06		
8.8 [3]	Carbon steel with additives (e.g. B or Mn or Cr), hardened and tempered	0.15 [4]	0.40	0.035	0.035	0.003	425
	Carbon steel, hardened and tempered	0.25	0.55	0.035	0.035		
9.8	Carbon steel with additives (e.g. B or Mn or Cr), hardened and tempered	0.15 [4]	0.35	0.035	0.035	0.003	425
	Carbon steel, hardened and tempered	0.25	0.55	0.035	0.035		
10.9 [5),6]	Carbon steel with additives (e.g. B or Mn or Cr), hardened and tempered	0.15 [4]	0.35	0.035	0.035	0.003	340
10.9 [6]	Carbon steel, hardened and tempered	0.25	0.55	0.035	0.035	0.003	425
	Carbon steel with additives (e.g. B or Mn or Cr), hardened and tempered	0.20 [4]	0.55	0.035	0.035		
	Alloy steel hardened and tempered [7]	0.20	0.55	0.035	0.035	0.003	
12.9 [8),9]	Alloy steel hardened and tempered [7]	0.28	0.50	0.035	0.035	0.003	380

1) The boron content can be 0.005% provided that the active boron is controlled through the addition of titanium and/or aluminium.
2) Bolts and screws in these property classes may be manufactured from free cutting steel with the following maximum content of sulphur, phosphorous and lead: sulphur 0.34%, phosphorous 0.11%, lead 0.35%.
3) For nominal diameters in excess of 20 mm, it can be necessary to use steel prescribed for class 10.9 in order to achieve sufficient hardening ability.
4) For boron-alloyed carbon steel with a carbon content of less than 0.25% (charge analysis), the minimum manganese content shall be 0.6% for property class 8.8 and 0.7% for 9.8 and 10.9.
5) Products shall also be marked with the property class symbol being underlined. All properties for 10.9 given in Table 4.3 shall be included in 10.9 but the lower tempering temperature gives other shrinkage properties at increased temperatures.
6) The material in these classes shall have sufficient hardening ability to guarantee the occurrence of about 90% martensite in the core of the threaded part after hardening, but before tempering.
7) The alloy shall contain at least one of the following substances with the stipulated minimum quantity: chrome 0.3%, nickel 0.3%, molybdenum 0.2%, vanadium 0.1%. When the substances are stated in combinations of two, three or four and have alloy quantities which are lower than those given above, the applicable limit for determining class is 70% of the total of the individual limit values above for applicable combinations of two, three or four elements.
8) A metallographically traceable, white phosphorous-enriched surface layer is not permitted for property class 12.9 on surfaces which are subject to tensile stress.
9) Investigations are ongoing concerning the composition and tempering temperature.

4.2.4 PHYSICAL PROPERTY REQUIREMENTS

When testing in accordance with the section "Testing methods" 4.2.5, bolts and screws at room temperature shall satisfy the physical property requirements given in Table 4.3.

TABLE 4.3
PHYSICAL PROPERTY REQUIREMENTS FOR BOLTS, SCREWS AND STUDS

Physical property requirements		Property class 3.6	4.6	4.8	5.6	5.8	6.8	8.8 [1] d ≤ 16 mm	8.8 [1] d > 16 mm [3]	9.8 [2]	10.9	12.9
Tensile strength R_m [4),5)] N/mm^2	nom	300	400		500		600	800	800	900	1000	1200
	min	330	400	420	500	520	600	800	830	900	1040	1220
Vickers hardness number, HV, F ≥ 98 N	min	95	120	130	155	160	190	250	255	290	320	385
	max	220 [6]						320	335	360	380	435
Brinell hardness number, HB, F = 30 D^2	min	90	114	124	147	152	181	238	242	276	304	366
	max	209 [6]						304	318	342	361	414
Rockwell hardness number, HR	min HRB	52	67	71	79	82	89	–	–	–	–	–
	min HRC	–	–	–	–	–	–	22	23	28	32	39
	max HRB	95 [6]					99.5	–	–	–	–	–
	max HRC	–					–	32	34	37	39	44
Surface hardness, HV 0.3	max	–						7)				
Lower yield stress R_{eL} [8], N/mm^2	nom	180	240	320	300	400	480	–	–	–	–	–
	min	190	240	340	300	420	480	–	–	–	–	–
Proof stress, $R_{p0.2}$ [9], N/mm^2	nom	–						640	640	720	900	1080
	min	–						640	660	720	940	1100
Stress in test load, S_p	S_p/R_{eL} or $S_p/R_{p0.2}$	0.94	0.94	0.91	0.93	0.90	0.92	0.91	0.91	0.90	0.88	0.88
	N/mm^2	180	225	310	280	380	440	580	600	650	830	970
Elongation after fracture, A%	min	25	22		20			12	12	10	9	8
Stress in angled tension [5]	Values for bolts and screws with full shank (not studs) shall not be less than the minimum values for tensile strength as given above.											
Impact strength, KU	J min	–			25	–		30	30	25	20	15
Angled impact on head		No fracture										
Min height of non-decarburised zone on the thread, E		–						1/2H_1			2/3H_1	3/4H_1
Max depth of full decarburisation, G, mm		–						0.015				

1) For bolts of property class 8.8 with diameter ≤16 mm, there is an increased risk of the threads of the nut stripping in the event of unintentional excess tensile stress giving a load greater than the test load. Reference to ISO 898-2 is recommended.
2) Only applies for nominal thread diameters d ≤ 16 mm.
3) For structural bolting, the limit is 12 mm.
4) Minimum tensile strength applies to products of nominal length 1 ≥ 2.5d. Minimum hardness applies to products of length 1 < 2.5d and other products which cannot be subjected to tensile test (for example, on account of the shape of the head).
5) For testing coarse bolts, screws and studs, the breaking loads given in Tables 4.4 to 4.7 apply for calculation of R_m.
6) In determining the hardness of the end of a bolt, screw or stud, the respective values shall be maximum 250 HV, 238 HB or 99.5 HRB.
7) The surface hardness shall not be more than 30 Vickers units greater than the measured core hardness of the product when both the surface hardness and core hardness are determined by HV 0.3. For property class 10.9, no increase in hardness of the surface is permitted if it indicates that the surface hardness exceeds 390 HV.
8) If the lower yield stress, R_{eL}, cannot be determined, it is permitted to measure the proof stress, $R_{p0.2}$. For property classes 4.8, 5.8 and 6.8, $R_{eL\ nom}$ is only given as part of the designation system and $R_{eL\ min}$ only as a basis for Sp.
9) The yield stress relationship according to the designation of property class and minimum value for proof stress 0.2% $R_{p0.2}$ applies for turned test bodies. These values can vary when they arise from test bolts and screws with full shank or test screws with full thread, depending on manufacturing method and the effect of size.

TABLE 4.4
MINIMUM BREAKING LOADS, METRIC ISO THREAD WITH COARSE PITCH

Thread	Pitch mm	Nominal stress area A_s nom mm^2	Property class 3.6	4.6	4.8	5.6	5.8	6.8	8.8	9.8	10.9	12.9
			Min failure load (A_s x $R_{m\ min}$), N									
M3	0.5	5.03	1 660	2 010	2 110	2 510	2 620	3 020	4 020	4 530	5 230	6 140
M3.5	0.6	6.78	2 240	2 710	2 850	3 390	3 530	4 070	5 420	6 100	7 050	8 270
M4	0.7	8.78	2 900	3 510	3 690	4 390	4 570	5 270	7 020	7 900	9 130	10 700
M5	0.8	14.2	4 690	5 680	5 960	7 100	7 380	8 520	11 350	12 800	14 800	17 300
M6	1	20.1	6 630	8 040	8 440	10 000	10 400	12 100	16 100	18 100	20 900	24 500
M7	1	28.9	9 540	11 600	12 100	14 400	15 000	17 300	23 100	26 000	30 100	35 300
M8	1.25	36.6	12 100	14 600	15 400	18 300	19 000	22 000	29 200	32 900	38 100	44 600
M10	1.5	58.0	19 100	23 200	24 400	29 000	30 200	34 800	46 400	52 200	60 300	70 800
M12	1.75	84.3	27 800	33 700	35 400	42 200	43 800	50 600	67 400[1)]	75 900	87 700	103 000
M14	2	115	38 000	46 000	48 300	57 500	59 800	69 000	92 000[1)]	104 000	120 000	140 000
M16	2	157	51 800	62 800	65 900	78 500	81 600	94 000	125 000[1)]	141 000	163 000	192 000
M18	2.5	192	63 400	76 800	80 600	96 000	99 800	115 000	159 000	–	200 000	234 000
M20	2.5	245	80 800	98 000	103 000	122 000	127 000	147 000	203 000	–	255 000	299 000
M22	2.5	303	100 000	121 000	127 000	152 000	158 000	182 000	252 000	–	315 000	370 000
M24	3	353	116 000	141 000	148 000	176 000	184 000	212 000	293 000	–	367 000	431 000
M27	3	459	152 000	184 000	193 000	230 000	239 000	275 000	381 000	–	477 000	560 000
M30	3.5	561	185 000	224 000	236 000	280 000	292 000	337 000	466 000	–	583 000	684 000
M33	3.5	694	229 000	278 000	292 000	347 000	361 000	416 000	576 000	–	722 000	847 000
M36	4	817	270 000	327 000	343 000	408 000	425 000	490 000	678 000	–	850 000	997 000
M39	4	976	322 000	390 000	410 000	488 000	508 000	586 000	810 000	–	1 020 000	1 200 000

1) For structural bolting, the values of 70 000, 95 500 and 130 000 N respectively apply.

TABLE 4.5
NOMINAL YIELD LOAD IN KN, METRIC ISO THREAD WITH COARSE PITCH

Thread	Pitch	Nominal stress area	Yield stress R_{eL} N/mm^2						$R_{p0.2}$ N/mm^2		
			180	240	320	300	400	480	640	900	1080
			Property class								
	mm	mm^2	3.6	4.6	4.8	5.6	5.8	6.8	8.8	10.9	12.9
M3	0.5	5.03	0.905	1.21	1.61	1.51	2.01	2.41	3.22	4.53	5.43
M3.5	0.6	6.78	1.22	1.63	2.17	2.03	2.71	3.25	4.34	6.10	7.32
M4	0.7	8.78	1.58	2.11	2.81	2.63	3.51	4.21	5.62	7.90	9.48
M5	0.8	14.2	2.56	3.41	4.54	4.26	5.68	6.82	9.09	12.8	15.3
M6	1	20.1	3.62	4.82	6.43	6.03	8.04	9.65	12.9	18.1	21.7
M7	1	28.9	5.20	6.94	9.25	8.67	11.6	13.9	18.5	26.0	31.2
M8	1.25	36.6	6.59	8.78	11.7	11.0	14.6	17.6	23.4	32.9	39.5
M10	1.5	58	10.4	13.9	18.6	17.4	23.2	27.8	37.1	52.2	62.6
M12	1.75	84.3	15.2	20.2	27.0	25.3	33.7	40.5	54.0	75.9	91.0
M14	2	115	20.7	27 .6	36.8	34.5	46.0	55.2	73.6	104	124
M16	2	157	28.3	37.7	50.2	47.1	62.8	75.4	100	141	170
M18	2.5	192	34.6	46.1	61.4	57.6	76.8	92.2	123	173	207
M20	2.5	245	44.1	58.8	78.4	73.5	98.0	118	157	220	265
M22	2.5	303	54.5	72.7	97.0	90.9	121	145	194	273	327
M24	3	353	63.5	84.7	113	106	141	169	226	318	381
M27	3	459	82.6	110	147	138	184	220	294	413	496
M30	3.5	561	101	135	180	168	224	269	359	505	606
M33	3.5	694	125	167	222	208	278	333	444	625	750
M36	4	817	147	196	261	245	327	392	523	735	882
M39	4	976	176	234	312	293	390	468	625	878	1050

TABLE 4.6
MINIMUM FAILURE LOAD, METRIC ISO THREAD WITH FINE PITCH

Thread	Nominal stress area A_s nom mm^2	Property class 3.6	4.6	4.8	5.6	5.8	6.8	8.8	9.8	10.9	12.9
		Min failure load (A_s x $R_{m\ min}$), N									
M8 x 1	39.2	12 900	15 700	16 500	19 600	20 400	23 500	31 360	35 300	40 800	47 800
M10 x 1	64.5	21 300	25 800	27 100	32 300	33 500	38 700	51 600	58 100	67 100	78 700
M10 x 1.25	61.2	20 200	24 500	25 700	30 600	31 800	36 700	49 000	55 100	63 600	74 700
M12 x 1.25	92.1	30 400	36 800	38 700	46 100	47 900	55 300	73 700	82 900	95 800	112 400
M12 x 1.5	88.1	29 100	35 200	37 000	44 100	45 800	52 900	70 500	79 300	91 600	107 500
M14 x 1.5	125	41 200	50 000	52 500	62 500	65 000	75 000	100 000	112-000	130 000	152 000
M16 x 1.5	167	55 100	66 800	70 100	83 500	86 800	100 000	134 000	150 000	174 000	204 000
M18 x 1.5	216	71 300	86 400	90 700	108 000	112 000	130 000	179 000	–	225 000	264 000
M20 x 1.5	272	89 800	109 000	114 000	136 000	141 000	163 000	226 000	–	283 000	332 000
M22 x 1.5	333	110 000	133 000	140 000	166 000	173 000	200 000	276 000	–	346 000	406 000
M24 x 2	384	127 000	154 000	161 000	192 000	200 000	230 000	319 000	–	399 000	469 000
M27 x 2	496	164 000	198 000	208 000	248 000	258 000	298 000	412 000	–	516 000	605 000
M30 x 2	621	205 000	248 000	261 000	310 000	323 000	373 000	515 000	–	646 000	758 000
M33 x 2	761	251 000	304 000	320 000	380 000	396 000	457 000	632 000	–	791 000	928 000
M36 x 3	865	285 000	346 000	363 000	432 000	450 000	519 000	718 000	–	900 000	1055 000
M39 x 3	1030	340 000	412 000	433 000	515 000	536 000	618 000	855 000	–	1070 000	1260 000

TABLE 4.7
NOMINAL YIELD LOAD IN kN, METRIC ISO THREAD WITH FINE PITCH

Thread	Nominal stress area	Yield stress R_{eL} N/mm^2 180	240	320	300	400	480	$R_{p0.2}$ N/mm^2 640	900	1080
	mm^2	Property class 3.6	4.6	4.8	5.6	5.8	6.8	8.8	10.9	12.9
M8 x 1	39.2	7.06	9.41	12.5	11.8	15.7	18.8	25.1	35.3	42.3
M10 x 1	64.5	11.6	15.5	20.6	19.3	25.8	31.0	41.3	58.0	69.7
M10 x 1.25	61.2	11.0	14.7	19.6	18.4	24.5	29.4	39.2	55.1	66.1
M12 x 1.25	92.1	16.6	22.1	29.5	27.6	36.8	44.2	58.9	82.9	99.5
M12 x 1.5	88.1	15.9	21.1	28.2	26.4	35.2	42.3	56.4	79.3	95.1
M14 x 1.5	125	22.5	30.0	40.0	37.5	50.0	60.0	80.0	113	135
M16 x 1.5	167	30.1	40.1	53.4	50.1	66.8	80.2	107	150	180
M18 x 1.5	216	38.9	51.8	69.1	64.8	86.4	104	138	194	233
M20 x 1.5	272	49.0	65.3	87.0	81.6	109	131	174	245	294
M22 x 1.5	333	59.9	79.9	107	100	133	160	213	300	360
M24 x 2	384	69.1	92.2	123	115	154	184	246	346	415
M27 x 2	496	89.3	119	159	149	198	238	317	446	536
M30 x 2	621	112	149	199	186	248	298	397	559	671
M33 x 2	761	137	183	244	228	304	365	487	685	822
M36 x 3	865	156	208	277	260	346	415	554	779	934
M39 x 3	1030	185	247	330	309	412	494	659	927	1112

4.2.5 TEST METHODS

4.2.5.1 *Tensile Testing of Test Rods*

The following properties shall be checked on test rods through tensile testing in accordance with ISO 6892:

a) tensile strength, R_m
b) lower yield stress, R_{eL}, or proof stress 0.2%, $R_{p0.2}$
c) percentage elongation after fracture

$$A = \frac{L_u - L_o}{L_o} \times 100\%$$

d) percentage area reduction after fracture

$$Z = \frac{S_o - S_u}{S_o} \times 100\%$$

Turned test rods in accordance with Figure 4.1 shall be used in the tensile test. If it is not possible to determine stretch yield after fracture, due to the length of the bolt, the area reduction after fracture shall be measured, provided that L_o is at least $3d_o$.

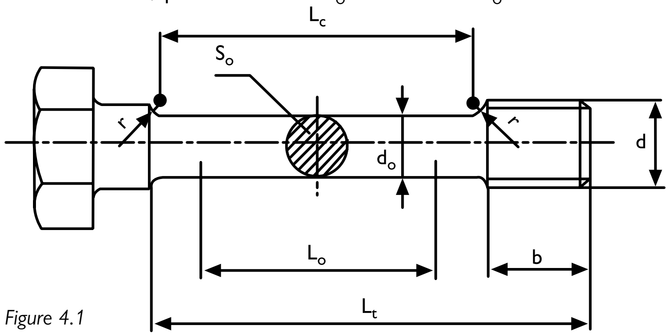


Figure 4.1

d	= nominal thread diameter
d_o	= diameter of the test rod (d_o< the minor diameter of the thread)
b	= thread length ($b \geq d$)
L_o	= $5d_o$ or ($5.65\sqrt{S_o}$): original measured length for determining stretch yield
L_o	$\geq 3d_o$: original measured length for determining area reduction
L_c	= length of straight part ($L_o + d_o$)
L_t	= total length of the test rod (L_c + 2r + b)
L_u	= measured length after fracture
S_o	= cross section area prior to tensile test
S_u	= cross section area after fracture
r	= radius ($r \geq 4$ mm)

When turning test rods of heat-treated bolts of d > 16 mm, the reduction of shank shall not exceed 25% of the original diameter (ca 44% of the original cross section area).

Products in property classes 4.8, 5.8 and 6.8 (cold headed products) shall be tensile tested coarse.

4.2.5.2 *Tensile Testing of Coarse (Shank Cross Section ≥ Nominal Stress Area) Bolts, Screws and Studs*

Tensile testing shall be carried out on coarse bolts and screws in accordance with the section above, "Tensile testing of test rods". This shall be carried out in order to determine the tensile strength R_m which is calculated on the nominal stress area A_s:

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

where

d_2	= basic dimension of the pitch diameter of the external thread (see ISO 724)
d_3	= the minor diameter of the external thread = $d_1 - \frac{H}{6}$

where

d_1	= basic dimension of the minor diameter of the external thread (see ISO 724)
H	= the fundamental triangle height of the thread (see ISO 68-1)

When testing coarse bolts, screws and studs, the forces given in Tables 4.4 and 4.6 apply.

When testing, a free thread length equal to a thread diameter (1d) shall be subjected to load. In order to meet the requirements of this test, the fracture shall occur in the shank or the thread and not in the junction of head and shank. The form of the test nut must conform to this.

The test speed, determined with an unloaded machine, shall not exceed 25 mm/min. The tensile test machine's tensile arrangement shall be self centring so that the test piece is not subject to side thrust.

4.2.5.3 *Hardness Testing*

For routine control of bolts, screws and studs, hardness can be measured at the head, at the screw tip or the shank after any coating has been removed and after suitable pre-treatment of the test piece. For property classes 4.8, 5.8 and 6.8, hardness shall only be measured at the screw tip.

If the maximum hardness is exceeded, a new test shall be carried out. The test shall be made one diameter from the tip of the test piece and opposite the centre of the test piece and its major diameter. In this position, the specified maximum hardness shall not be exceeded. In the case of uncertainty, hardness testing in accordance with Vickers is decisive for approval.

4.2.5.4 *Wedge Loading Test of Coarse Bolts (not studs)*

The wedge loading test shall be carried out in accordance with Figure 4.2.

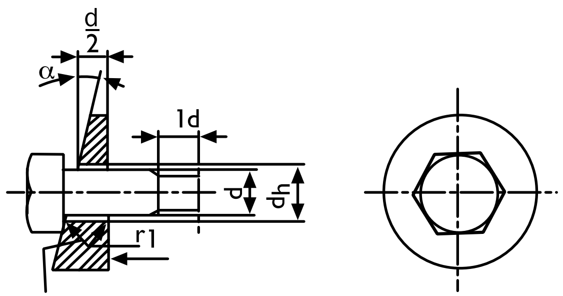


Figure 4.2

Radius or 45° chamfer

d_h in accordance with ISO 273, series average

The minimum distance from the end of the bolt to the contact surface of the nut in the fastening device shall be d. A hardened, angled washer with dimensions and angles in accordance with Tables 4.8 and 4.9 shall be placed under the head of the bolt. The tensile test shall be carried out until a fracture occurs.

In order to fulfil the requirements of this test, a fracture shall arise in the shank or the thread of a full shank bolt and not in the junction between head and shank. The bolt shall satisfy the requirements regarding minimum fracture limit, either through wedge loading testing or through extra tensile testing without an angled washer, in accordance with the values which apply for the applicable property class prior to fracture occurring.

Bolts threaded up to the head shall be considered to fulfil the requirements of this test if a breakage which causes fracture arises in the free thread length, and has also expanded or spread to the junction between shank and head, or to the head, prior to breaking.

For bolts of product grade C, a radius r_1 shall be used, as:

$$r_1 = r\ \text{max} + 0.2$$

$$\text{where } r\ \text{max} = \frac{d_a\ \text{max} - d_s\ \text{min}}{2}$$

NOTE The symbols r, d_a and d_s are defined in ISO 225.

TABLE 4.8
HOLE DIAMETERS FOR WEDGE LOADING TEST

Nominal thread diameter d	3	3.5	4	5	6	7	8	10	12	14
d_h	3.4	3.9	4.5	5.5	6.6	7.6	9	11	13.5	15.5
r_1	0.7	0.7	0.7	0.7	0.7	0.8	0.8	0.8	0.8	1.3

Nominal thread diameter d	16	18	20	22	24	27	30	33	36	39
d_h	17.5	20	22	24	26	30	33	36	39	42
r_1	1.3	1.3	1.3	1.6	1.6	1.6	1.6	1.6	1.6	1.6

TABLE 4.9
MEASURES OF OBLIQUENESS

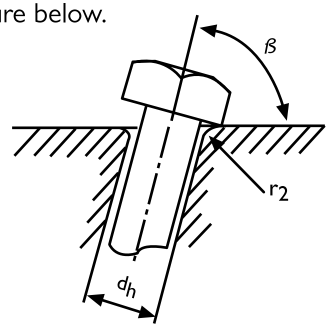
Nominal bolt diameter d	Property class of:			
	Bolts with smooth (non-threaded) shank length $l_s \geq 2d$		Fully threaded bolts or bolts with plain shank length $l_s < 2d$	
	3.6 4.6 4.8 5.6 5.8 8.8 9.8 10.9	6.8 12.9	3.6 4.6 4.8 5.6 5.8 8.8 9.8 10.9	6.8 12.9
mm	α ±30’			
d ≤ 20	10°	6°	6°	4°
20 < d ≤ 39	6°	4°	4°	4°

For products with contact diameters in excess of 1.7d which do not fulfil the requirements of the wedge loading test, the head may be worked to 1.7d and re-testing carried out measuring obliqueness as given in Table 4.9.

Additionally, for products with contact diameters in excess of 1.9d, the obliqueness shall be reduced from 10° to 6°.

4.2.5.5 *Oblique Impact Angle Test of Coarse Bolts of d ≤ 10 mm, Too Short for Wedge Loading Test to be Carried Out*

The oblique impact angle test shall be carried out in accordance with the Figure below.



For d_h and r_2 ($r_2 = r_1$) see Table 4.8.
The thickness of the test plate should be greater than 2d.

Figure 4.3

The head of the bolt should turn to an angle of 90° - ß by means of repeated hammer blows, without fractures being visible in the junction between shank and head on inspection with magnification not less than x8 and not more than x10.

Full-threaded bolts shall be approved, even if fractures occur in the first thread, however, provided that the head does not come loose.

TABLE 4.10
VALUES FOR THE OBLIQUE IMPACT ANGLE TEST

Prop. class	3.6 4.6 5.6	4.8 5.8 6.8 8.8 9.8 10.9 12.9
Angle ß	60°	80°

4.2.6 DECARBURISATION TEST

By using an appropriate means of measurement, an axial section through the thread shall be examined to determine whether the height of the zone of the base material (E) and the depth of the completely decarburised zone (G) are within the set limits, see Figure 4.4.

The maximum value for G and the formulae for minimum value for E are given in Table 4.3.

H_1 = the section height of external thread at maximum dimension.

Decarburisation outside the permitted limits can put at risk the strength of a bolted joint.

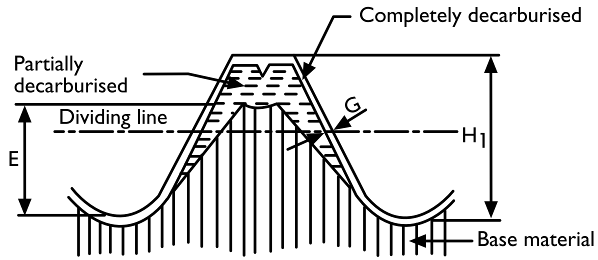


Figure 4.4

4.3 PHYSICAL PROPERTY REQUIREMENTS FOR SCREWS WITH ST THREADS MADE FROM CARBURISED STEEL

Extract from the Swedish and international standard SS-EN ISO 2702 for screws with ST threads.

4.3.1 EXTENT

In this international standard, properties and associated test methods are given for carburised, self-tapping steel screws with thread ST2.2-ST8 in accordance with SS-EN ISO 1478.

4.3.2 MATERIAL

Self-tapping screws shall be manufactured from cold-heading steel for carburising.

4.3.3 METALLURGICAL REQUIREMENTS

4.3.3.1 *Surface Hardness*

After thermal treatment, surface hardness shall be minimum 450 HV 0.3. (See ISO 6507-2).

4.3.3.2 *Carburising Depth*

Carburising depth shall be in conformity with Table 4.11.

TABLE 4.11 CARBURISING DEPTH Dim. in mm

Thread	Depth of hardening	
	min	max
ST2.2 ST2.6	0.04	0.10
ST2.9 ST3.3 ST3.5	0.05	0.18
ST3.9 ST4.2 ST4.8 ST5.5	0.10	0.23
ST6.3 ST8	0.15	0.28

4.3.3.3 *Core Hardness*

After thermal treatment, core hardness shall be 270 HV 5 – 390 HV 5 for threads ≤ ST3.9 and 270 HV 10 – 390 HV 10 for threads ≥ ST4.2.

4.3.4 TORSIONAL STRENGTH TESTING

The shank of the test screw (whether coated or not, as it is received) shall be clamped in a suitable threaded vice or similar equipment in such a way that the clamped part of the screw is not damaged, and that at least two full rows of thread project outside the clamping device, and at least two full rows of thread, apart from at the tip of the screw, are held in the clamping device. An insert with threaded blind hole may be used instead of a clamping device (see Figure 4.5) provided that the depth of the hole is such that a fracture would occur above the tip.

Using appropriate, calibrated torque equipment, torque is applied until a fracture occurs. The screw shall resist the minimum torsional strength given in Table 4.12.

For determining strength and assembly properties for self-tapping screws, please see SS 3392.

TABLE 4.12 TORSIONAL STRENGTH

Thread	Minimum torsional strength Nm
ST2.2	0.45
ST2.6	0.9
ST2.9	1.5
ST3.3	2
ST3.5	2.7
ST3.9	3.4
ST4.2	4.4
ST4.8	6.3
ST5.5	10
ST6.3	13.6
ST8	30.5

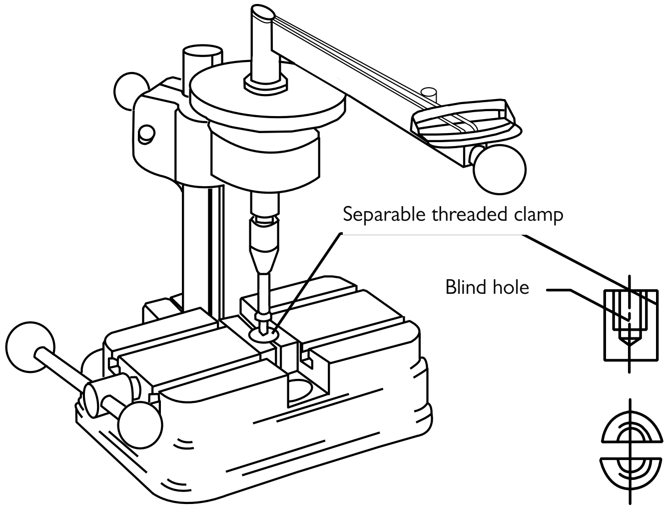


Figure 4.5

4.4 STRENGTH OF SET SCREWS

The regulations for property classes, material, testing, etc., are given SS-ISO 898-5.

4.5 ADDITIONAL INFORMATION

Swedish standard
SS-EN 20898-1 (SS-ISO 898-1)
SS-EN 20898-2 (SS-ISO 898-2)
SS-ISO 898-5
SS-EN ISO 898-6
SS-EN 20898-7
SS-1523
SS-3392

ON THE STRENGTH OF NUTS

5

5.1 IN GENERAL

"It is the tiny details that are decisive", sang Povel Ramel once upon a time, and it is perfectly true that tiny details can have big effects. A nut that came loose was sucked into the air intake of a jet engine and demolished the turbine fans. The nut cost less than one Euro. The repair of the engine cost about half a million.

Just as with bolts and screws, it is necessary to have the strength of nuts well organised. An identification system needed to be applied in the same way as for the bolts and screws. Agreeing about this took time, but it was finalised together with the identification system for bolts and screws in 1967.

If a bolted joint is overloaded, it is usually preferable that the bolt should break first. A broken bolt can be seen immediately, while a nut with stripped thread can remain on the bolt so that the fact that the joint is defective and has no tensile strength is not visible. Only the German steel construction norms permit nut threads to break in this way, as in their case a broken bolt can mean a fitter losing his foothold and falling from a power pylon or from the structure he is climbing.

Figure 5.1

If the height of the nut is too low, and if it is made of harder material than the bolt, there is a risk that the nut will shear the bolt thread.

Figure 5.2

As with the bolts, differing practices developed in different countries as regards the “right” nut height. In ISO, it was considered that the metric nut height should be 0.8 x the major diameter of the thread, and this rule of thumb worked well until more sophisticated tightening methods were developed in that it was now possible to load the screw joint closer to its yield stress. At the same time, improved materials began to be used in the manufacture of bolts, which raised the actual yield stress and thus the load on the threads of the nut in a highly tensioned bolt/nut joint.

A major international survey was started, which included Swedish involvement, to investigate the various phenomena which occurred in the loading of a bolt/nut joint. In due course, a system of calculation was developed by a Canadian – the Alexander method – which was accepted as a norm for the calculation of nut heights. This showed that it was no longer possible to use a constant relationship between the height of the nut and the diameter of the bolt. Nuts were therefore divided into two groups – Type 1 and Type 2 – depending on physical property requirements, manufacturing methods, materials used in manufacture and in some cases, thermal treatment. Life became more complicated both for the manufacturers and the buyers. In practice today, Type 1 is used, it has a nut height of about = 0.9d. Nuts from M16 and coarser are also hardened. If you would like to know more, you can read about it in the relevant Swedish standard SS-EN 20898-2 or in ISO 898-2.

Many suppliers still sell their nuts quoting DIN 934, despite the fact that they know that this standard no longer exists. This means that they deliver nuts which only have a height of 0.8d and with mechanical properties which conform to an ISO standard which was discontinued in 1978, or in accordance with the German standard DIN 267 Teil 4, which was discontinued in February 1992. Products according to these discontinued standards do not at all meet present requirements, and lead the user to have a false sense of security.

5.2 PHYSICAL PROPERTY REQUIREMENTS FOR NUTS WITH METRIC ISO THREADS, COARSE PITCH

Extract from the Swedish and international standard EN 20898-2 or ISO 989-2 for nuts with metric ISO threads.

5.2.1 EXTENT AND APPLICATION

The international standard gives the mechanical properties of nuts and the required proof load for testing at room temperature (see ISO 1). The properties differ at higher and lower temperatures respectively.
It applies for nuts:
- with nominal thread diameter up to 39 mm
- for triangular IS threads and with diameters and pitch in accordance with ISO 68 and ISO 262 (coarse pitch)
- with diameter/pitch combinations in accordance with ISO 261 (coarse pitch)
- with thread tolerance 6H in accordance with ISO 965-1 and ISO 965-2
- with width across flats in accordance with ISO 272 or equivalent
- with specified mechanical requirements
- of nominal height ≥ 0.5D
- manufactured from non-alloyed or low-alloyed steel

It does not apply for nuts for which special requirements are imposed as regards
- locking properties (see ISO 2320)
- weldability
- corrosion resistance (see ISO 3506)
- ability to resist temperatures above +300°C and below -50°C

5.2.2 DESIGNATION SYSTEM

5.2.2.1 *Nuts of Nominal Height of ≥ 0.8D*

Nuts of nominal height of ≥ 0.8D (effective thread length ≥ 0.6D) are designated with a figure which indicates the maximum property class of bolts with which the nut can be paired. The fastener can fracture through excessive tension. Such fracture can consist of a break in the bolt shank or stripping of thread in the nut or on the bolt. A fracture in the shank occurs suddenly and is therefore easy to notice. Stripping of thread is a gradual process and therefore more difficult to discover. It involves a danger of partly damaged fasteners remaining in the bolted joint. It is therefore desirable that bolted joints are designed so that fractures always occur in the bolt shank, but unfortunately shear strength depends on so many factors (strength of nut and bolt material, play in the thread, width across flats, etc.) that nuts would have to be unreasonably high to guarantee against such fractures on all occasions.

A bolt of a certain property class mounted with a nut of the equivalent property class (see Table 5.1) is intended to constitute a joint in which one can achieve tensile stress in the bolt equivalent to its proof load without risk of the thread stripping.

TABLE 5.1
DESIGNATION SYSTEM FOR NUTS OF NOMINAL HEIGHT ≥ 0.8D

Property class of nut	Equivalent bolt		Nut	
	Property class	Thread area	Type 1 Thread area	Type 2
4	3.6 4.6 4.8	> M16	> M 16	–
5	3.6 4.6 4.8 5.6 5.8	≤ M16 ≤ M39	≤ M39	–
6	6.8	≤ M39	≤ M39	–
8	8.8	≤ M39	≤ M39	> M16 ≤ M39
9	9.8	≤ M16	–	≤ M16
10	10.9	≤ M39	≤ M39	–
12	12.9	≤ M39	≤ M16	≤ M39

Note – In general, nuts of a higher property class can replace nuts of a lower property class. This is recommended for bolt-nut joints which will be subject to forces greater than the yield stress or test tensioning.

If, however, pretension is applied of a value in excess of the proof load, the intention is that the nuts shall be so designed that 10% of the excessively tensile-stressed joints shall fracture through bolt breakage, in order to warn the user that his assembly methodology is unsuitable.

5.2.2.2 *Nuts of a Nominal Height of ≥ 0.5D but < 0.8D*

Nuts with a nominal height of ≥ 0.5D but < 0.8D (effective thread length ≥ 0.4D but < 0.6D) are designated with a combination of two figures: the second figure gives the nominal stress of proof load testing with a hardened mandrel; while the first figure indicates that the loadability of a bolt-nut joint is lower in comparison with loadability with a hardened mandrel and also in comparison with a bolt-nut joint as described in 5.2.2.1. The effective loadability is determined not only on the basis of the hardness of the nut and the bolt upon which the nut is mounted. The designation system and stresses in proof load testing of the nuts are shown in Table 5.2.

TABLE 5.2 DESIGNATION SYSTEM AND STRESSES IN PROOF LOAD TESTING OF NUTS OF NOMINAL HEIGHT ≥ 0.5D BUT < 0.8D

Property class of nut	Nominal tension in proof load testing N/mm^2	Current stress in proof load testing N/mm^2
04	400	380
05	500	500

5.2.3 MATERIAL

Nuts shall be made of steel of a chemical composition in accordance with Table 5.3. Nuts of property class 05, 8 (Type 1 > M16), 10 and 12 shall be tempered.

TABLE 5.3
LIMITS OF CHEMICAL COMPOSITION

Property class		Chemical composition (control analyses), % C max	Mn min	P max	S max
4 1) 5 1) 6 1)	–	0.50	–	0.060	0.150
8 9	04 1)	0.58	0.25	0.060	0.150
10 2)	05 2)	0.58	0.30	0.048	0.058
12 2)	–	0.58	0.45	0.048	0.058

1) Nuts in these property classes may be made of free cutting steel unless otherwise agreed between manufacturer and user. In such case, the following maximum contents of sulphur, phosphorous and lead are permitted: Sulphur 0.34%, phosphorous 0.12% and lead 0.35%.

2) Alloy substances may be added if it is necessary to achieve the physical property requirements.

5.2.4 MECHANICAL PROPERTIES

When nuts are tested in accordance with methods described in the section "Test methods", 5.2.5, they shall satisfy the hardness requirements given in Tables 5.4 and 5.5.

TABLE 5.4
MECHANICAL PROPERTIES

Thread		Property class														
		04					05					4				
		Stress in proof load testing S_p	Vickers hardness HV		Nut		Stress in proof load testing S_p	Vickers hardness HV		Nut		Stress in proof load testing S_p	Vickers hardness HV		Nut	
over	up to	N/mm²	min	max	condition	type	N/mm²	min	max	condition	type	N/mm²	min	max	condition	type
–	M4	380	188	302	NQT1)	low	500	272	353	QT2)	low	–	–	–	–	–
M4	M7															
M7	M10															
M10	M16															
M16	M39											510	117	302	NQT 1)	1

Thread		Property class																			
		5 3)					6					8									
		Stress in test loading S_p	Vickers hardness HV		Nut		Stress in test loading S_p	Vickers hardness HV		Nut		Stress in test loading S_p	Vickers hardness HV		Nut		Stress in test loading S_p	Vickers hardness HV		Nut	
over	up to	N/mm²	min	max	con-dition	type	N/mm²	min	max	con-dition	type	N/mm²	min	max	con-dition	type	N/mm²	min	max	con-dition	type
–	M4	520	130	302	NQT1)	1	600	150	302	NQT1)	1	800	180	302	NQT1)	1	–	–	–	–	–
M4	M7	580					670					855	200								
M7	M10	590					680					870									
M10	M16	610					700					880									
M16	M39	630	146				720	170				920	233	353	QT2)		890	180	302	NQT1)	2

Thread		Property class																			
		9					10					12									
		Stress in proof load testing S_p	Vickers hardness HV		Nut		Stress in proof load testing S_p	Vickers hardness HV		Nut		Stress in proof load testing S_p	Vickers hardness HV		Nut		Stress in test loading S_p	Vickers hardness		Nut	
over	up to	N/mm²	min	max	con-dition	type	N/mm²	min	max	con-dition	type	N/mm²min	max	dition	con- type	N/mm²		min	max	con-dition	type
–	M4	900	170	302	NQT1)	2	1040	272	353	QT2)	1	1140	295	353	QT2)	1	1150	272	353	QT2)	2
M4	M7	915	188				1040					1140					1150				
M7	M10	940					1040					1140					1160				
M10	M16	950					1050					1170					1190				
M16	M39	920					1060					–	–	–	–	–	1200				

1) NQT = Non-Quenched and Tempered
2) QT = Quenched and Tempered
3) Maximum hardness for bolts of property classes 5.6 and 5.8 will be changed to 220 HV in the next revision of ISO 898-1:1988. This is the maximum hardness of the bolt thread's engagement area, while the bolt tip and head may have a maximum hardness of 250 HV. The values for stress in proof load testing are therefore based on a maximum hardness of the bolt of 220 HV.

Note – Minimum hardness is only prescribed for hardened nuts and for nuts which are too large for proof load testing. For all other nuts, no minimum hardness is not prescribed, but has been given only for information. For nuts which are not tempered and which satisfy the proof load testing requirements, minimum hardness shall not be a reason for rejection.

TABLE 5.5
PROOF LOADS, COARSE THREADS

Thread	Thread pitch	Nominal stress area of the mandrel A_s	Property class										
			04	05	4	5	6	8		9	10	12	
			Proof load (A_s x S_p) N										
					type 1	type 1	type 1	type 1	type 2	type 2	type 1	type 1	type 2
	mm	mm²											
M3	0.5	5.03	1 910	2 500	–	2 600	3 000	4 000	–	4 500	5 200	5 700	5 800
M3.5	0.6	6.78	2 580	3 400	–	3 550	4 050	5 400	–	6 100	7 050	7 700	7 800
M4	0.7	8.78	3 340	4 400	–	4 550	5 250	7 000	–	7 900	9 150	10 000	10 100
M5	0.8	14.2	5 400	7 100	–	8 250	9 500	12 140	–	13 000	14 800	16 200	16 300
M6	1	20.1	7 640	10 000	–	11 700	13 500	17 200	–	18 400	20 900	22 900	23 100
M7	1	28.9	11 000	14 500	–	16 800	19 400	24 700	–	26 400	30 100	32 900	33 200
M8	1.25	36.6	13 900	18 300	–	21 600	24 900	31 800	–	34 400	38 100	41 700	42 500
M10	1.5	58.0	22 000	29 000	–	34 200	39 400	50 500	–	54 500	60 300	66 100	67 300
M12	1.75	84.3	32 000	42 200	–	51 400	59 000	74 200	–	80 100	88 500	98 600	100 300
M14	2	115	43 700	57 500	–	70 200	80 500	101 200	–	109 300	120 800	134 600	136 900
M16	2	157	59 700	78 500	–	95 800	109 900	138 200	–	149 200	164 900	183 700	186 800
M18	2.5	192	73 000	96 000	97 900	121 000	138 200	176 600	170 900	176 600	203 500	–	230 400
M20	2.5	245	93 100	122 500	125 000	154 400	176 400	225 400	218 100	225 400	259 700	–	294 000
M22	2.5	303	115 100	151 500	154 500	190 900	218 200	278 800	269 700	278 800	321 200	–	363 600
M24	3	353	134 100	176 500	180 000	222 400	254 200	324 800	314 200	324 800	374 200	–	423 600
M27	3	459	174 400	229 500	234 100	289 200	330 500	422 300	408 500	422 300	486 500	–	550 800
M30	3.5	561	213 200	280 500	286 100	353 400	403 900	516 100	499 300	516 100	594 700	–	673 200
M33	3.5	694	263 700	347 000	353 900	437 200	499 700	638 500	617 700	638 500	735 600	–	832 800
M36	4	817	310 500	408 500	416 700	514 700	588 200	751 600	727 100	751 600	866 000	–	980 400
M39	4	976	370 900	488 000	497 800	614 900	702 700	897 900	686 600	897 900	1035 000	–	1171 000

5.2.5 **TEST METHODS**

5.2.5.1 ***Proof Load Testing***

Proof load testing should always be carried out when the available test equipment so permits and is a reference method for sizes ≥ M5.

The nut shall be mounted on a hardened, threaded mandrel in accordance with Figures 5.3 and 5.4 on this page. For reference purposes, the axial tension test is decisive.

The proof load shall be applied to the nut in the axial direction for 15 seconds. The nut shall resist the load without a fracture occurring in the thread or the body of the nut, and after the load is removed it shall be possible to loosen it by hand. If the threads of the mandrel are damaged during the test, the test shall be ignored. (It may be necessary to use a hand tool to make the nut move. Such turning is permitted provided that it is limited to half a turn and the nut can subsequently be loosened with fingers).

The hardness of the mandrel shall be minimum HRC 45.

The threads of the mandrel shall be of thread tolerance 5h6g with the exception that the tolerance of the major diameter shall be the last quarter of the tolerance range of tolerance class 6g.

5.2.5.2 ***Proof Loads***

The values of proof loads are given in Table 5.5. The nominal stress area A_s is calculated according to:

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

d_2 = base dimension of the pitch diameter of the external thread.

d_3 = minor diameter of external thread = $d_1 - \frac{H}{6}$

d_1 = base dimension of the minor diameter of the external thread.

H = fundamental triangle height of the thread.

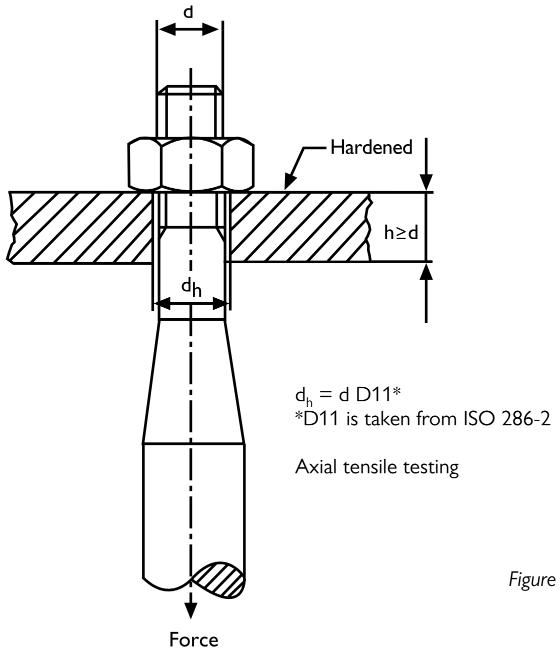


Figure 5.3

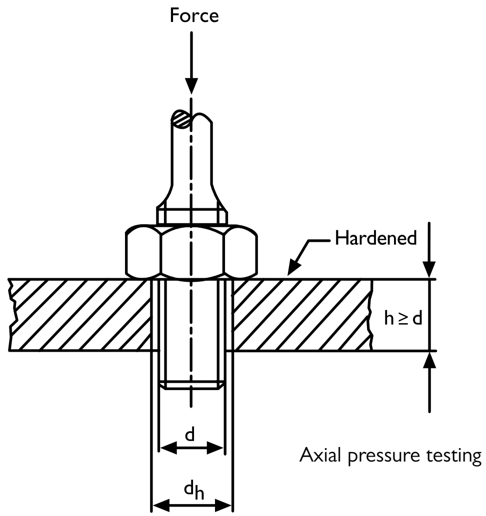


Figure 5.4

5.2.5.3 *Hardness Testing*

In a routine check, hardness shall be measured on one of the contact surfaces. The hardness value shall be the mean of three measurements made with 120° displacement. If there is a dispute, a measurement shall be in an axial direction through the nut, and the measurement shall be made as close to the major diameter of the nut threads as possible.

Hardness testing in accordance with Vickers is a reference method, and if possible, a force of HV 30 shall be applied.

If the Brinell test and Rockwell hardness test are applied, the conversion tables in accordance with ISO 4964 shall be used.

Harness testing in accordance with Vickers shall be performed in accordance with ISO 6507-1.

Harness testing in accordance with Brinell shall be performed in accordance with ISO 6506.

Recommendation:
Always quote the current international ISO or EN standard when purchasing, and check that these regulations have been complied with by the supplier.

5.3 ADDITIONAL INFORMATION

Swedish standard
SS-EN 20898-1 (SS-ISO 898-1)
SS-EN 20898-2 (SS-ISO 898-2)
SS-ISO 898-5
SS-EN ISO 898-6
SS-EN 20898-7
SS-1523
SS-3392

6 THE MARKING SYSTEM

6.1 IN GENERAL

When strength regulations and test instructions have been agreed on, it is necessary to inform the user of this in some way. That is why an international designation system for fasteners with associated marking regulations was introduced.

6.2 WHY?

Because of the movement of goods between countries and continents, it has become increasingly important to be able to identify the origin of goods, in case something happens and it is necessary to claim damages. There are examples of accidents, including with fatal outcome, due to false markings and forged test certificates.

6.3 MARKING REQUIREMENTS

For bolts and screws, marking with property classes for thread diameters d ≥ 5 mm is required in accordance with SS-E2N 0898-1, as follows:

Hexagonal-head bolts of all property classes.

Hexagonal socket countersunk-head screws of property class 8.8 or higher.

Stud bolts of property class 8.8 or higher.

Other types of bolt of property classes 4.6, 5.6, 8.8 or higher, in accordance with agreement between the parties involved.

For nuts, marking with property class in accordance with SS-EN 20898-2 is required for thread diameters d ≥ 5 mm in all property classes.

The manufacturer's trademark shall always be included on bolts, screws and nuts when marking with property class is required.

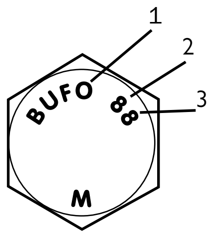


Figure 6.1

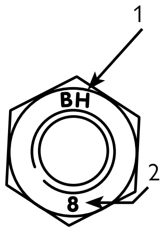


Figure 6.2

6.4 THE PURPOSE OF THE MARKING SYSTEM

6.4.1 BOLTS AND SCREWS

1. Manufacturer's designation. Registered trademark.

2. The first figure denotes one hundredth of the nominal tensile strength of the bolt or screw in N/mm^2.
   In this case, 100 x 8 = 800 N/mm^2.

3. The second figure denotes the relationship between the yield stress and tensile strength of the bolt or screw, expressed in tenths. In this case, the relationship is = 0.8. If one multiplies the two figures, the yield stress of the bolt or screw in N/mm^2 is obtained.
   In this case, 800 x 0.8 = 640 N/mm^2.

6.4.2 NUTS

1. Manufacturer's mark, for example,
   BUMAX = BUFAB BULTEN STAINLESS

2. The figure denotes one hundredth of the tensile strength of the bolt for which the nut can be used without breaking.
   In this case, 100 x 8 = 800 N/mm^2.

For stainless and acid-resistant items, special markings apply; these are given in Chapter 8.4.

6.5 THE UN THREAD

Examples of marked products show how BULTEN previously marked some of them with inch threads, they are now no longer manufactured. On UNC and UNF threaded BUFO 8.8. bolts, the three dashes indicate the American property class Grade 5 (≈8.8). The dashes are represented on the nuts by the three triangles or points.

The six dashes on UNC and UNF threaded BUFO 10.9 bolts denote property class SAE Grade 8 (≈10.9), the equivalent on the nut side are the six triangles.

6.6 EXAMPLES OF MARKINGS

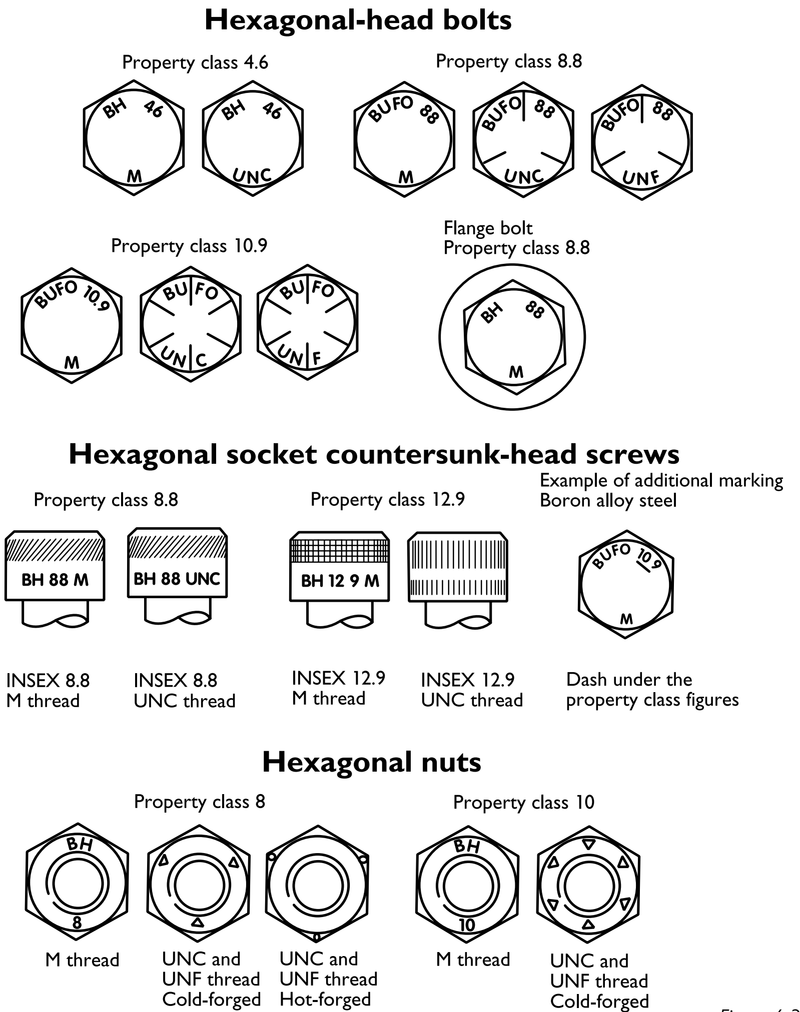
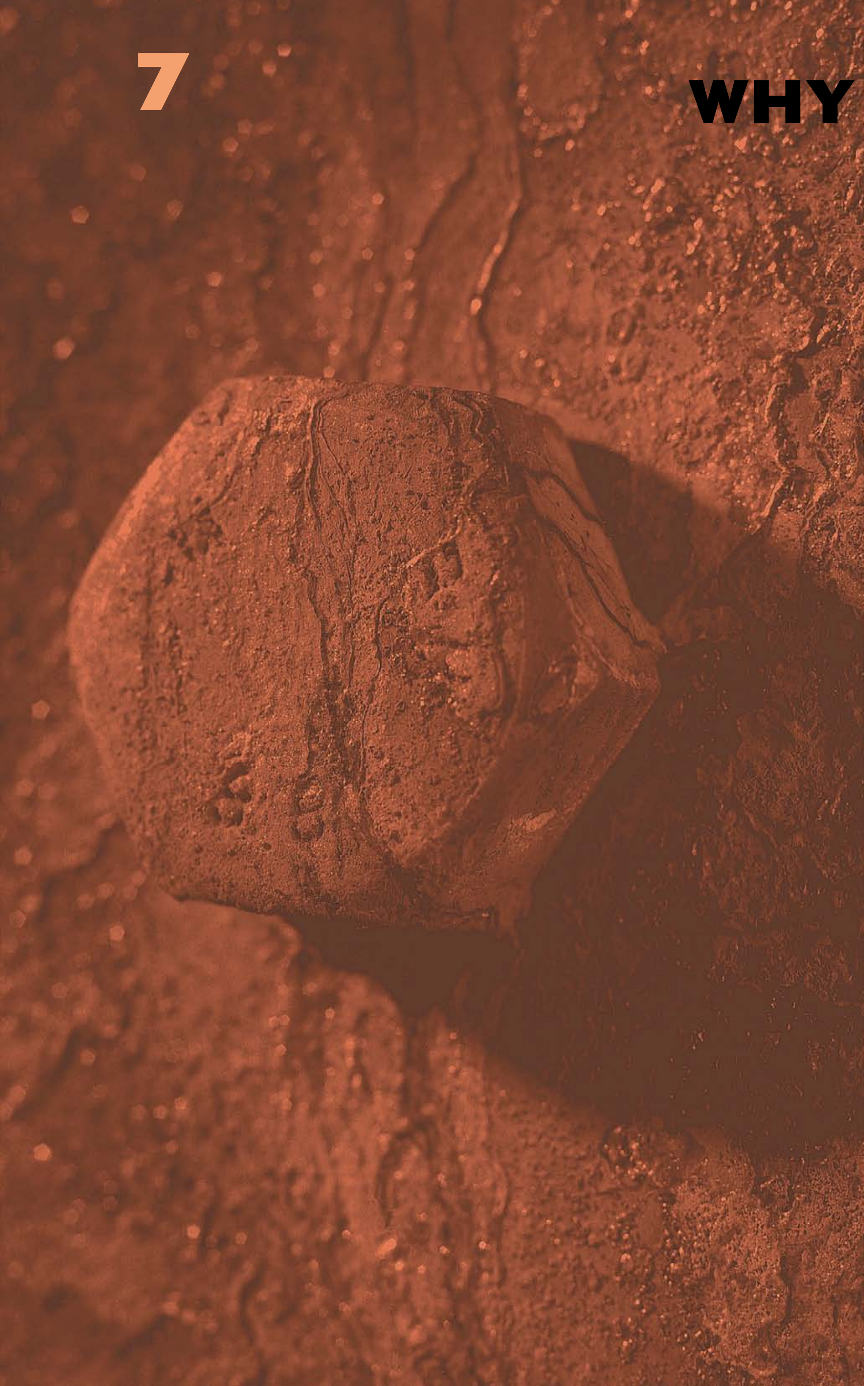


Figure 6.3

6.7 ADDITIONAL INFORMATION

Various articles from Fogningsteknik
Fastener Technology International
Fastener World
Swedish Standard
SS-EN 20898-1 (SS-ISO 898-1)
SS-EN 20898-2 (SS-ISO 898-2)
SS-ISO 898-5
SS-EN ISO 898-6
SS-EN 20898-7



7

WHY DOES RUST OCCUR?

7.1 IN GENERAL

Rust damage on products costs large sums every year, and can also cause breakdowns through various kinds of corrosion weakening supporting structures. There are therefore strong reasons to think about what type of surface treatment a fastener should be given. Below, an account is given of the mechanism of corrosion, with some general advice which can help in choosing corrosion protection for fasteners. Also, several types of surface treatment on special parts are described.

The most common surface treatment that BUFAB carries out on special parts is three variants of electrolytic zinc coating, i.e., with white chromating, yellow chromating and the Chrome VI free coatings such as GEOMET® or DELTAPROTEKT®. Other electrolytic surface treatments which can be provided are zinc coating with green or black chromating and zinc/nickel, nickel/chrome and tin. Other surface treatments which can be delivered are mechanical zinc coating, hot zinc coating,

Often, additional lubrication of surface-treated products is required for controlled friction in assembly, and therefore we can offer several combinations.

7.2 WHAT IS CORROSION?

Steel rusts, copper oxidises and other metals, except those which are most inert, are broken down in a similar way. This type of material destruction is given the general name of corrosion. It occurs when the material reacts with its surroundings and is converted into other substances – to corrosion products. Almost all corrosion which occurs in normal working environments is of electrochemical type. It occurs in galvanic cells, corrosion cells, which function roughly in the same way as a flashlight battery. The battery has a carbon rod in its centre, and a casing of zinc plate. The carbon rod is called the cathode, the zinc plate the anode. Inside the battery, there is also a paste or sludge which is called electrolyte.

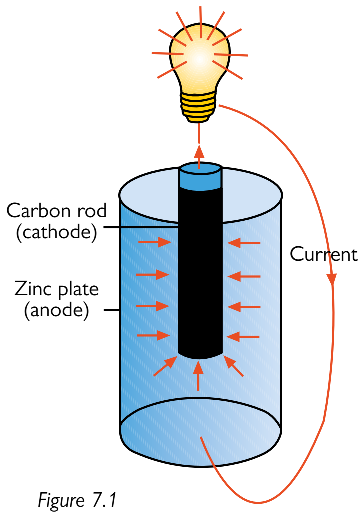


Figure 7.1

When you switch on a flashlight, current runs from the carbon rod through the bulb to the zinc plate. From there, the current goes through the electrolyte back to the carbon rod. The current takes with it zinc particles from the plate, which becomes corroded and, in due course, begins to leak. The cathode is inert (- pole), the anode is reactive (+ pole).

Galvanic cells which cause the corrosion can form when two different metals (or one metal and another substance which conducts electricity in the same way as a metal, for example, graphite) come into contact with an electrolyte. Corrosion caused by such material combinations is called by the common name of galvanic corrosion. Galvanic cells, often extremely small, also occur on individual metal surfaces. This is connected to the fact that industrial metals consist of microscopic granules of varying composition, and can also be due to various impurities on the surface, such as oxide scale, slag residue, etc. The granules and particles are, as with various metals, of varying degrees of inertness in relation to one another.

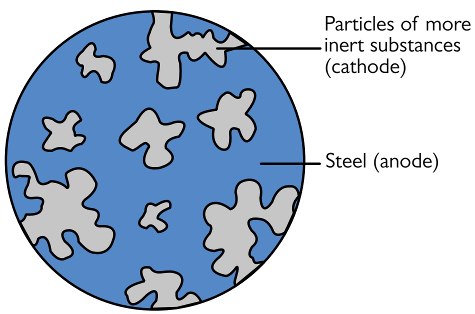


Figure 7.2

In the case of corrosion through galvanic cells, the most inert metal or particle is always a cathode, the most reactive one is always an anode.

7.3 DAMP AND OXYGEN

Without damp, no corrosion occurs, nor does it occur without oxygen. But oxygen and damp are present in the air. A certain level of damp is necessary for corrosion to occur. If the relative humidity of air exceeds 60%, a very thin film of damp forms on the metal surface, and this functions as an electrolyte.

If a steel surface has impurities on it, such as dust, salt, etc., it can corrode if the relative humidity is less than 60%, because dirt absorbs damp. Outdoors in Sweden, the humidity of the air is almost always sufficiently high for steel, for example, to rust. Indoors, air is heated, and as a rule, so dry that steel does not rust. When the temperature falls, for example in a factory closed for vacation, there is a risk of corrosion.

7.4 WHEN STEEL RUSTS

When the film of damp covers the steel surface, a very large number of galvanic cells are formed, all functioning in the same way as a flashlight battery.

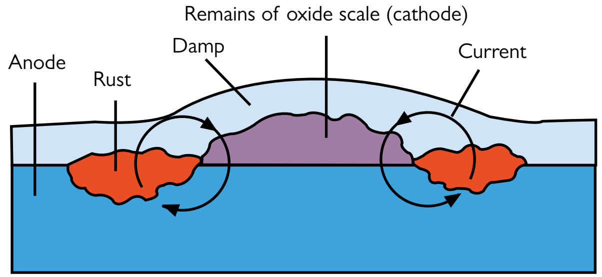


Figure 7.3

A current flows from the more inert granules or particles (cathodes) through the steel to the more reactive parts of the steel surface (anodes). The current returns from the more reactive parts out in the film of damp (electrolyte) and through it back to the more inert parts. Where the current leaves the anode, rust forms.

After a time, the entire surface is covered with rust, which also spreads downwards into the steel. As long as there is access to damp and oxygen, rust will form.

The speed of corrosion depends on the difference in inertness between the metals or metal granules, the electrical conductivity of the electrolyte, the supply of oxygen and differences in size between anode and cathode. A screw can be anode or cathode.

If a steel screw is tightened in a copper plate, the screw becomes anode, because the copper is more inert. The screw will rust more quickly because the "potential difference" is large.

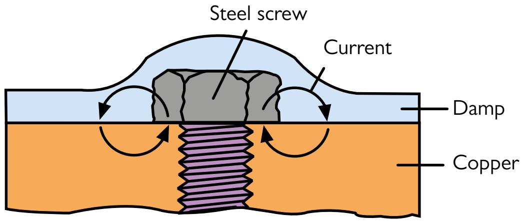


Figure 7.4

If the same steel screw is tightened in a more reactive metal, for example, zinc, the screw will be cathode and will not rust. On the hand, the plate will corrode because it is more reactive than the screw.

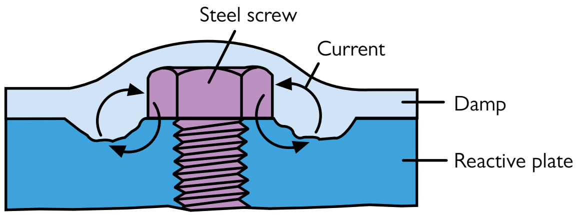


Figure 7.5

The greater the difference of inertness between the metals, the more rapid the corrosion of the more reactive/anode. In the absence of special protection measures, one should therefore only combine metals which have the same, or almost the same, inertness, if they are to be subjected to a damp environment.

In order to be able to assess the relationships of metals against one another more easily, they have been placed on a "ladder" called the galvanic stress series, see Figure 7.6. Through an electrical means of measurement, it has been possible to deduce what is called the potential of the metals.

This is an electrochemical concept which gives an idea of how inert or reactive a metal is. The most common stress series is measured with seawater as electrolyte at a temperature of +20°C. Gold is the most inert metal and is at the top of the ladder. Magnesium is the most reactive and is at the bottom.

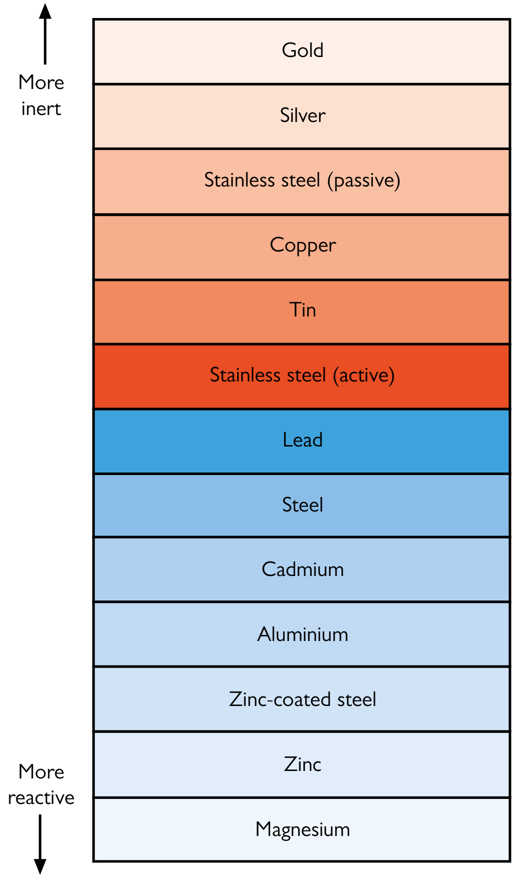


Figure 7.6

7.5 CONDUCTIVITY OF AN ELECTROLYTE

If the electrical conductivity of an electrolyte is poor, as is often the case in fresh water, the corrosion is concentrated to the area closest to the point of contact between the anode and the cathode.

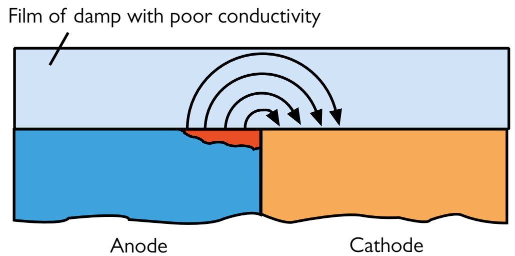


Figure 7.7

On the other hand, if conductivity is good, as in seawater, the spread of corrosion will be greater. Outdoors, the speed of corrosion is affected by impurities in the air. Rusting is worst of all in very contaminated, industrial and city atmospheres, and after that in maritime climates with salt content. Metals last better in the countryside.

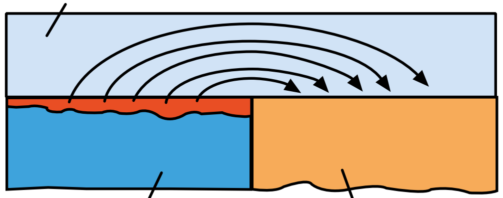


Figure 7.8

7.6 THE EFFECT OF OXYGEN

If oxygen is prevented from reaching a metal surface, the surface does not corrode. The more oxygen that arrives, the faster it corrodes within certain limits. In a hermetically sealed steel beam, for example, no rust can occur, because no oxygen gets in. Similarly, oxygen diminishes or is exhausted in closed heating systems, and corrosion decreases or ceases.

7.7 THE SIZES OF ANODES AND CATHODES

The larger the surface of the cathode, the more oxygen will be in contact with it, and the smaller the surface of the anode in relation to the cathode, the more concentrated the corrosion current will be in the anode.

The combination of small anode surface/large cathode surface therefore involves a greater risk of corrosion.

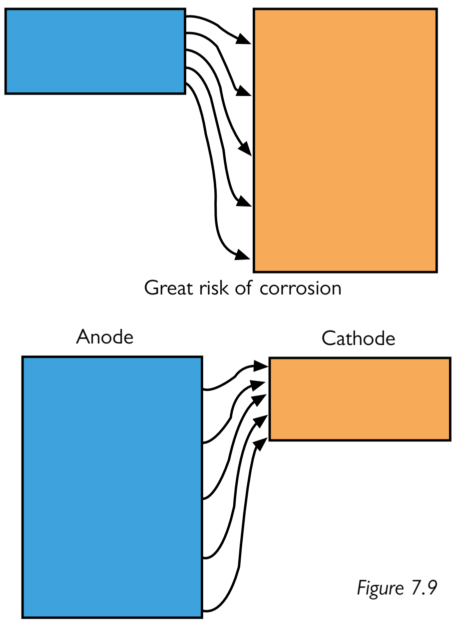


Figure 7.9

7.8 ZINC ELECTROPLATING

Zinc electroplating, or as it is more correctly called, electrolytic zinc coating, is carried out in an electrolyte with water content containing zinc ions. The item is connected as cathode to a direct current source. Metallic zinc in the form of spheres or plates can function as anodes, but with more modern methods the anode is instead a steel plate. When the current is connected, zinc is deposited on the item. New zinc is dissolved chemically in a zinc-dissolving tank and replaces the zinc ions from the electrolyte which have been used up.

In zinc electroplating of fasteners, they are enclosed in perforated plastic drums, which are rotated in the treatment bath. The course of treatment in zinc electroplating is degreasing, acid treatment, electrical degreasing, activation, zinc coating, chromating and drying. Between each stage of treatment, there is a thorough rinse with water.

The thickness of the zinc layer is controlled by the electrolyte current in relation to the surface of the item during constant treatment time in the electrolyte. The thickness of the zinc layer on the finished item is measured with a magnetic method in connection with emptying of each processing drum. In the case of more exact determination, a microscope or x-ray is used.

The thickness of the zinc layer is usually a minimum of 5 μm for fasteners with threads less than M6, and minimum 8 μm for M6 and upwards. In the case of special runs, zinc layers of up to 20-25 μm can be produced. The thickness of the zinc layer is measured on the head or tip of the screw when screws are the object of the process, and on the hexagonal side or contact surface in the case of nuts.

The highest property class that we zinc-electroplate is 10.9. Property classes in excess of 8.8 involve expulsion of hydrogen after the zinc electroplating in order to reduce the risk of hydrogen brittleness.

7.8.1 ZINC ELECTROPLATING PLUS CHROMATING

In badly ventilated spaces, white corrosion products form on zinc surfaces in a relatively short time. This corrosion is usually called white rust. By chromating zinc surfaces, the onset of zinc corrosion can be delayed. Chromating is a chemical process which converts the topmost layer of the coated zinc, and thus improves its corrosion protection. In chromating, the newly zinc-plated items are dipped into an acid chromating solution. The composition of the solution determines the type of chromate film which forms. In general, bright chromating (often called bright zinc plating) is the most common method, but for aesthetic reasons, forms of chromating other than the bright variant may be preferred. Yellow and green chromating considerably improve protection against white rust, while bright and black chromating only improve protection to a certain degree. Long-term protection, i.e., the time it takes until base metal corrosion (red rust) occurs, is only marginally prolonged by colour chromating in comparison with bright chromating. The long-term protection is almost entirely due to the thickness of the zinc layer.

7.8.1.1 *Where is bright zinc plating used?*

If long-term corrosion protection is desired, bright zinc plating with a surface layer of about 10 μm should only be used in less aggressive environments or as a surface to paint onto. If the thickness of the surface layer is increased, the corrosion protection time is increased to an equivalent degree.

Table 7.1 gives the guideline values for the life of a 10 μm-thick bright zinc plated surface in various environments.

TABLE 7.1
LENGTH OF LIFE OF BRIGHT ZINC PLATING, GUIDELINE VALUES

Environment	Life in years, approx Bright zinc plating 10 μm
Indoors	50
Countryside	8
Small towns	5
Large towns	2
Coastal atmosphere, west coast	5
Industrial atmosphere, light	4
Industrial atmosphere, normal	2
Industrial atmosphere, heavy	1

7.8.1.2 *Where is colour-chromated zinc used?*

Colour-chromated electrolytic zinc coating, especially yellow or green chromated, is used where better protection against white rust is desired than bright chromating can offer. Black chromating is primarily used for aesthetic reasons, which is also often the case in respect of yellow and green chromating. The life in various environments of a colour-chromated 10 μm zinc layer is only marginally longer than that recorded for bright zinc plating in Table 7.1.

7.8.2 THE ZINC/IRON METHOD

Continuous work on improving known methods has been carried out, and trials have been carried out using zinc in combination with other metals. Of these can be mentioned Zn + Ni, Zn + Co and Zn + Fe. The last named, which is called the zinc/iron method was invented in Japan and is much used in Sweden.

Surface treatment is carried out as with other electrolytic zinc coating of fasteners, but in an electrolyte with zinc/iron. The difference is that steel anodes are always used instead of anodes of zinc. The zinc is chemically dissolved in a zinc solution tank, and replaces the zinc ions used up in the electrolyte. The iron content of the layer is 0.3 – 0.6 % of weight in order to achieve the best possible corrosion protection, and an even black colour in the subsequent chromating. If the iron content is too low, the colour will be greenish, and if it is too high it will be grey. The zinc chromating used is silver-free and is similar to green chromating in composition.

During treatment in the drums, the items become worn to differing extents, which can give varying results in testing in a neutral salt spray as regards the time until white rust occurs. In order to achieve 200 hours until white rust occurs in a neutral salt spray, some form of subsequent sealing is required, sealer or wax with a corrosion-resisting effect. Subsequent sealing with hexavalent chromium is less suitable because it can give rise to a high leachable amount of hexavalent chromium, which produces a risk of allergy for the assembly personnel.

7.8.2.1 *Where is black zinc-iron used?*

Zinc/iron with black chromating is a good replacement for other zinc electroplating with chromating. It gives good surface treatment and considerably better corrosion protection, also it is more amenable in modern design when it is desired that fasteners shall merge into the environment and not stand out as bright patches.

7.9 NICKEL PLATING

Nickel plating with or without an underlying copper layer is carried out as a special order.

Nickel plating is an electrolytic process, and the course of treatment is in principle the same as with zinc electroplating. The electrolyte of course, has another composition and chromating does not occur.

In some cases, the steel surfaces are coated with a thin copper coating of 1 – 2 μm before the nickel is applied. The copper coating is also applied through an electrolytic process. The copper surface increases the adhesion of the nickel to the parts, and improves the corrosion protection somewhat.

The thickness of the nickel coat applied to the fastener is usually 5–10 μm. The corrosion protection ability of a nickel layer of this thickness, with or without an underlying copper layer, is very limited. The area of use is therefore most often as a decorative surface treatment for indoor environments.

7.10 TIN COATING

BUFAB can carry out electrolytic tin coating as a special order. The treatment process is generally the same as that used in zinc electroplating and electrical nickel plating. Subsequent treatment with chromating is not applicable when tin is involved. Normally, a layer of 5–10 µm is applied to the fastener.

Tin coating is primarily used to increase solderability, but is also used for corrosion protection in certain applications.

7.11 HYDROGEN BRITTLENESS AND DEHYDROGENATION

In electrolytic surface treatment and sometimes in dipping, hydrogen is released so that hydrogen atoms can enter the base metal where they can combine with hydrogen molecules. When, for example, a high-strength screw is loaded, hydrogen can collect in zones of high tension, which can lead to brittle fractures.

The harder an item is, the more sensitive it is to the uptake of hydrogen in connection with electrolytic surface treatment. Items with a hardness in excess of HRC 31, HV 320, HB 300 or an tensile strength in excess of 1000 N/mm^2 shall be subject to dehydrogenation after electrolytic surface treatment.

Dehydrogenation is a heat treatment which is carried out at a temperature of about 200°C. It is important that treatment is commenced as soon as possible after the surface treatment, and that the item is in the heat for a sufficient period. BUFAB follows ISO 4042 and ISO 15330.

7.12 HOT DIP GALVANIZING

Fasteners which are to be hot dip galvanized must first undergo pretreatment consisting of degreasing, rinsing, treatment with acid, rinsing, fluxing in zinc ammonium chloride and drying in warm air. Sometimes, sandblasting is used instead of treatment with acid. After treatment, the fasteners are loaded into a perforated steel basket and lowered into molten zinc at a temperature of about 550°C. After some minutes, the basket is raised and immediately centrifuged in order to remove excess zinc.

The average zinc thickness for fasteners with threads less than M10 is about 40 µm, and for M10 and above about 55 µm. Property classes above 10.9 cannot be hot dip galvanized on account of tempering effects and the fact that zinc can force its way in between granules and produce micro-cracks, which can weaken steel catastrophically.

7.12.1 *Where is Hot Dip Galvanizing used?*

Hot dip galvanized fasteners are especially appropriate for use in construction in outdoor environments or in demanding indoor environments. Table 72 gives the approximate lives of 55 µm-thick hot dip galvanized layers in various environments.

TABLE 7.2
LIFE OF HOT DIP GALVANIZED LAYERS, GUIDELINE VALUES

Environment	Approximate life in years Hot dip galvanized surface 55 µm
Countryside Small towns Large towns	40 28 11
Coastal atmosphere Industrial atmosphere, light Industrial atmosphere, normal	28 20 11
Industrial atmosphere, heavy	5

7.13 MECHANICAL METAL COATING

Mechanical metal coating is a process in which fasteners and other items are coated mechanically with zinc, tin or a combination of these metals. The big advantages of this method are that no hydrogenation occurs with metal coating, and that it is carried out at room temperature. Mechanical metal coating with zinc should be seen as a complement to electrolytic and hot dip galvanizing. The surface treatment is applied in a rotating drum, similar to a cement mixer.

The process is as follows: The sandblasted items are emptied into a treatment drum together with glass beads of various sizes, and water. First, light oxide which may have formed on the items is dissolved, through the application of a weak oxide solvent containing an inhibitor in order to prevent bating on the steel. Subsequently, copper salt is applied and the items are copper-coated chemically. The copper coating is very thin, less than 1 µm, and functions as a good binder between the steel and the subsequent zinc coating.

The next step is to apply the zinc powder. This is carried out two or more times, in accordance with the zinc thickness required. Certain special chemicals are also applied in connection with the zinc-coating step.

The zinc powder fastens onto the items, in principle, by the glass beads hammering the zinc powder onto them. Different items require different "hammering energy", which is achieved through varying the speed of rotation of the treatment drum and the size of the glass beads. Usually, the glass beads are a mixture of various dimensions.

When the coating is complete, which takes about an hour, the contents of the drum are emptied onto a separator, and the coated items are separated from the glass beads and the water. Subsequently, the items can be bright-chromated or yellow-chromated. The glass beads are recycled and used again.

A zinc coating of 5 µm up to 30 µm-thick can be achieved with this process. The corrosion protection of a mechanically applied zinc coating is comparable to a zinc coating applied with other methods, provided that the zinc coat of the same weight per area is compared. A coat consisting of a combination of tin and zinc gives somewhat better corrosion protection than zinc alone. A coat applied mechanically has a somewhat lower density than, for example, an electrolytic or hot dip galvanizing. In order to achieve the same weight of zinc per area with mechanical application, the thickness of the coating must be increased by about 20%.

7.13.1 *Where is Mechanical Metal Coating used?*

The fact that hydrogenation does not occur in this process and that the process is carried out at room temperature, makes it especially appropriate for surface coating of high-strength or surface-hardened fasteners. Electrolytic zinc coating of such product types does involve the risk of hydrogen embrittlement. hot dip galvanizing cannot be used either, as the high temperature of the zinc bath tempers the items and reduces hardness. The risk of zinc embrittlement is also obvious.

Items which are appropriate for running in our equipment should have a minimum diameter of about 4 mm and maximum length of about 150 mm. The maximum item weight is about 0.5 kg. Items with narrow blind holes, for example, cross-recessed screws, are difficult to coat. Coating of such items must therefore be studied in each individual case.

7.14 PHOSPHATING

We can carry out zinc phosphating in two separate coating weights. Phosphating is a chemical process in which the iron in the surface of the steel reacts with the process chemicals. Through special treatment, we can achieve a fine-crystal zinc phosphate, which is preferable in many cases.

The quantity of phosphate which the item receives in phosphating is normally expressed in g/m^2. Phosphate processes produce a coating weight of 12–25 g/m^2 and 6–11 g/m^2 respectively. A diameter increase of an item which is phosphated with a coating of 12–25 g/m^2 is 4 to 8 µm. For the phosphate coat to give any appreciable corrosion protection, the coat must be given a light oil coating or be coated in some other way.

7.14.1 *Where is Phosphating used?*

Phosphating as a surface treatment is most often chosen for qualified fasteners in engines and similar structures. The choice is primarily because friction is low and scatter small in connection with mounting, which means that high, even pretension can be obtained.

The corrosion protection ability of phosphating plus the application of a light coating of oil is limited. In outdoor environments, red rust generally occurs within less than one year.

7.15 OTHER COATINGS WHICH ARE NOT PURELY METAL

There is continual development within the coating-treatment industry with attempts to develop other methods than electrolytic metal coatings, for example. This has led to a system of organic surface coatings, comparable to painting, which function as a barrier and protect the base metal from damp which promotes corrosion. In other systems, large quantities of metal flakes have been suspended in an organic binder to give a surface cathodic protection. Common to all these methods is coating through dipping, centrifuging to even out the coating, followed by heating to some hundreds of degrees Centigrade for drying or hardening. Often this procedure is called "dip-spin" (dipping – centrifuging). Some of the most common coatings that BUFAB can provide are described below.

7.15.1 GEOMET®

Treatment with GEOMET is carried out in a similar way as for POLYSEAL. The agent, which consists of 80% zinc flakes and 10% aluminium flakes, is suspended in water which also contains a binder with chromating effect. This is applied to the fasteners through dipping followed by centrifuging, drying and hardening at about 300°C.

The first coating gives a coat of 2–3 µm. The coating is often applied several times with hardening in between until the coat is about 8 µm thick, in order to obtain the desired corrosion protection.
The colour is either matt silver-grey or black. By adding topcoats, the threaded fastener acquires a low, even friction.

The method is available free from hexavalent chromium. In corrosion tests in a neutral salt spray with 8–12 µm GEOMET 500 A, no red rust occurred until after 1000 hours.

7.15.1.1 *Where is GEOMET used?*

GEOMET is well suited for high-strength screws (10.9 and 12.9) where, with some other treatment, there would be a risk of hydrogen embrittlement. Within the automotive industry, it is often used in difficult environments, for example, under a vehicle, thanks to its wear resistance. The silver-grey surface is also used for decoration.

7.15.2 DELTA-TONE/DELTA-SEAL

DELTA-Tone is not a paint, it is a zinc-rich system in an organic/non-organic base.

DELTA-Seal is a so-called top coat with organic base which seals the DELTA-Tone layer and improves corrosion protection. It is available in various colours and with various additives to reduce friction. DELTA-Tone/DELTA-Seal do not contain any heavy metals.

Surface treatment is carried out in three steps:

1. Zinc phosphating without the application of a thin layer of oil.
2. Dipping in DELTA-Tone, centrifuging and hardening at 200°C.
3. Dipping in DELTA-Seal, centrifuging and oven-hardening, which is often carried out twice.

DELTA-Tone is available in silver-grey and black. DELTA-Seal is available in a number of colours; silver, black, white, light blue, light green, dark blue, light grey and red. Other colours can be specially ordered. Phosphating, followed by DELTA-Tone (2x4 µm) + DELTA-Seal (2x5 µm) produces red rust in a neutral salt spray only after 500 hours.

7.15.2.1 *Where are DELTA-Tone/DELTA-Seal used?*

The method is especially appropriate as a surface treatment for high-strength screws, and can be carried out without the steel taking in any hydrogen to speak of. The surface layer also has a relatively high temperature resistance (over 200°C). The surface treatment is commonly used in the German automotive industry.

7.16 LUBRICATION OF FASTENERS

Different surface treatments give considerable variations in friction coefficients in pretensioning threaded fasteners.

This creates problems with, for example, automotive construction companies, as they would prefer to have the same assembly data for a certain screw or nut dimension irrespective of how the product is treated or where it sits. Things were simpler in the past when mainly untreated or bright-zinc coated products were used. Problems became especially apparent with the transition to black zinc-iron treating, which sometimes gives high friction.

During recent years, systems have been developed for lubricating (sometimes called waxing) in connection with surface treatment, so that friction can be controlled towards desired levels.

Sometimes, agents can be added to the surface treatment agent itself, be included in some agents for sealing as a last operation, or applied after all other treatment.

At BUFAB we provide several different agents, for example, products from Gleitmo® which are completely touch-dry, or agents for customers' special requirements.

Usually, treatment is requested which gives friction in the range 0.1 to 0.18 or about 0.08 depending on the area of use.

STAINLESS AND ACID-RESISTANT – THE MOST SECURE PROTECTION 8

8.1 WHAT IS "STAINLESS" STEEL?

By "stainless" is generally meant a large group of types of steel which contain at least 12% chromium (Cr) and often additions of other alloy substances, for example, nickel (Ni) and molybdenum (Mo). The types of steel that BULTEN STAINLESS generally uses can be divided into the following main groups in accordance with their inner construction or structure:

1. Ferritic steel
(partially temperable)
Cr = 12 – 30%
Mo, if used = 1.3 – 2.5%
C = max 0.08%

2. Martensitic steel
(temperable)
Cr = 12 – 18%
Mo, if used 1.3 – 2.0%
C = max 0.25%

3. Austenitic steel
(not temperable)
Cr = 16.5 – 26%
Ni = 7 – 25%
Mo, if used 1.5 – 4.5%
C = max 0.07%

4. Ferritic-austenitic steel
(partially temperable)
Cr = 17 – 27%
Ni = 4 – 6%
Mo, if used = 1.3 – 2.0%
C = max 0.10%

8.2 Stainless or Acid-resistant?

Within various supplier and customer groups, different interpretations of the concepts of stainless and acid-resistant steel are used. In its widest usage, all steel which contains over 12% chromium is designated "stainless", irrespective of what other alloy substances are included.

In daily usage however, "stainless" primarily applies to the group's austenitic 18/8 steel (18% Cr, 8% Ni) without Mo, and "acid-resistant" to the Mo-alloyed austenitic and ferritic-austenitic steel types. The designation "acid-resistant steel" is primarily derived from the cellulose industry, and is connected with good resistance to sulphite cooking acid. Although nowadays one is aware of the limitations of acid-resistant steel, it is still true to say that in general, the acid-resistant steel types are more resistant to corrosion than other stainless steel types, i.e., they have a lower dissolving speed. Compared with ordinary steel, for example, low-alloy steel types, stainless and acid-resistant steel have a lower dissolving speed, which means that their corrosion resistance is much better.

8.3 COMMON TYPES OF STEEL

In choosing stainless material, one should strive for the best weighting between corrosion resistance, strength and cost.

Stainless steel SS 14 23333 (A2) has good general strength in moderate corrosion exposure, for example in air except in a coastal climate, in fresh water, in oxidising acids[1)] (for example, nitric acid), in organic acids and in a large number of alkali and salt solutions. This steel should, however, not be used in a non-oxidising acid (for example, hydrochloric acid) and agents with a chlorine content that is of brackish water and seawater type.

Acid-resistant steel SS 14 2347 (AISI 316) is included in the A4 group, and is the commonest steel for A4 fasteners. The steel is intended for normal corrosion exposure in a maritime climate. Steel 14 2343 8AISI 316 L Hi Mo), which is used in BULTEN STAINLESS' new standard BUMAX® 88 and BUMAX® 109 programmes have improved corrosion resistance compared with normal commercial A4. The BUMAX products can therefore be used in somewhat more aggressive environments, for example, water with a chloride content and non-oxidising acids. For severe corrosion conditions in liquids or gases, one must in many cases use other materials, which are dealt with in section 8.5 "Choice of material".

[1)] **NB!** An acid can be oxidising at room temperature and non-oxidising at higher temperatures.

Our stock programme comprises fasteners within the following steel groups:

A2	(SS 14 2333-AISI 304)
A4	(SS 14 2347-AISI 316)
BUMAX 88	(SS 14 2343-AISI 316 L Hi Mo)
BUMAX 109	(SS 14 2343-AISI 316 L Hi Mo)

TABLE 8.1
OUR MOST COMMON TYPES OF MATERIAL

Swedish designation	International US/ASTM UNS	Germany Werkstoff No. or alternatively EN	France AFNOR	International or alternatively BSAB marking for bolts and nuts
SS 2303	AISI 420	1.4021	Z2 0C13	C1
SS 2326	S 44400	1.4521	Z3 CD T18-02	F 1
SS 2333	AISI 304	1.4301	Z7 CN 18-09	A 2
(302HQ)	AISI 304CU	–	–	A 2
SS 2343 1)	AISI 316L	1.4436 1)	Z3 CND 17-12-03	
SS 2347	AISI 316	1.4401	Z7 CND 17-11-02	A 4
SS 2562	AISI 904L	1.4539	Z2 NCDU 25-20	SÅ 5
SS 2378	S31254	1.4529	–	SÅ 10
SS 2324	AISI 329	1.4460	Z5 CND 27-05AZ	SÅ 8
SS 2327	S 32304	1.4362	Z2 CN 23-04A2	–
SS 2377	S 31803	1.4462	Z2 CND 22-05-03	SAF 2205
SS 2328	S 32750	–	Z3 CND 25-06AZ	SAF 2507

TABELL 8.2
MATERIAL FOR HIGH TEMPERATURES

Swedish designation	International US/ASTM UNS	Germany Werkstoff No. or alt EN	France AFNOR	International or alternatively BSAB marking
SS 2361	AISI 3105	1.4845	Z8 CN 25-20	SÅ 3
SS 2368	S 30815	EN 1.4835	–	253MA
SS 2570	AISI 660	1.4980	Z5 NCTDV 25-15B	SÅ 7
17-4PH	AISI 630	1.4542	Z7 CNU 17-04	
NIMONIC 80A 2)		2.4952	NC 20 TA	SÅ N

1) Max C 0.030 A4-316L.
2) Registered trademark of INCO.

8.4 SPECIAL MATERIAL

For special environments, other materials are required than those which are included in stock products. In our factories, we can manufacture fasteners of most types of steel and alloy available in the market. Examples of our most common types of material for use in temperatures from – 200°C ~ +400°C are shown in Table 8.1. Table 8.2 shows some materials for use at temperatures > 400°C. Table 8.3 shows designations and chemical analyses of some the most used types of steel. Examples of the areas of application of various types of steel are also given in the table.

8.5 CHOICE OF MATERIAL

The types of steel listed can be used in most structures which require stainless or acid-resistant fasteners.

If in doubt regarding choice, BULTEN STAINLESS can in most cases help and give recommendations. In order to be able to make a correct choice of material, one should know the following:

1. Application, manufacturing process in which the fastener shall be used.
2. Medium – liquid or gas that the fastener will be in contact with, chemical composition, temperature and pressure conditions. Particular attention should be given to cases where the fastener will come into contact with other parts which are also in contact with the liquid in question. It is then important to state what material is involved.
3. Requirements concerning the mechanical and physical properties, strength, magnetism, etc.
4. Information about any material previously used and its life and problems in use can also be of considerable value.

TABLE 8.3 MATERIAL DATA

Designation	Types of steel					
ISO	A4					
BULTEN STAINLESS		BUMAX A4				
(old designations)	SÅ 4	SÅ 4	SÅ 8	SÅ 3	SÅ 5	SÅ 7
SS	2347	2343	2324	2361	2562	2570
AISI	316	316LHiMo	329	310S	N 08904	660
Werkstoff No.	1.4401	1.4436	1.4460	1.4845	(1.4539)	1.4980
British norm	316S16	En58J	–	S 523	904314	–
French norm	Z7 CND 17-11-02	Z3CND 17-12-03	25CND27-05Az	Z8CN25-20	Z2NCDU25-20	Z5NCTDV25-15B
Analysis						
Carbon (C) %	max 0.07	max 0.03	max 0.10	max 0.08	max 0.02	max 0.08
Chromium (Cr) %	18	16-18.5	24-27	4-26	19-21	13.5-16.0
Nickel (Ni) %	11	10.5-14	4.5-6.0	19-22	24-26	24-27
Molybdenum (Mo) %	2.0-2.5	2.5-3.0	1.3-1.88	–	4.5-5.0	1.0-1.5
Copper (Cu) %	–	–	–	–	1.2-2.0	–
Strength of the raw material in annealed condition at 20°C						
Tensile strength R_m N/mm^2	min 490	min 490	min 590	min 780	min 590	min 960
Yield stress $R_{p0.2}$, N/mm^2	min 195	min 440	min 245	min 245	min 660	min 660
Elongation A_s %	min 45	min 45	min 20	min 35	min 50	min 14
Brinell hardness HB	max 200	max 200	max 260	max 220	max 140	approx 270
Physical data (in annealed condition)						
Magnetic	No	No	Yes	No	No	No
Scaling temperature in air ca. °C	850	850	1070	850	1150	approx 1000
Machining properties						
Hardness	No	No	No	No	No	Can be hardened separately
Weldability	Very good [1)]	Very good [1)]	Good [2)]	Very good	Very good	Bad
Notes	Is used, for example, within the sulphite, cellulose and bleaching industry. Good corrosion resistance but somewhat less good resistance against pitting than 2343.	Is used, for example, within the sulphite, cellulose and bleaching industry. Good corrosion resistance.	Is used within the sulphite industry for generally the same purposes as 2343. Should only be selected where a high yield stress cannot be achieved with quality 2343	Good heat resistance. Should not be used in gases with sulphur content at temperatures in excess of 700°C.	Good resistance against sulphuric acid and phosphoric acid. Is used, for example, within the chemical industry for batch pickling and within the cellulose industry. Also appropriate for marine environments.	Very good strength and corrosion resistance at high temperatures, up to 750°C. Appropriate for structures exposed to high temperatures, for example, internal combustion engines, and where springy, flexible qualities are desired.

[1)] Not sensitive to intergranular corrosion up to 500°C.

[2)] With annealed weld joints. Non-annealed weld joints satisfactory.

TABLE 8.3 MATERIAL DATA (CONT.)

Designation	Types of steel					
ISO BULTEN STAINLESS (old designations) SS AISI Werkstoff No. British norm French norm	C1 SÅ6 2302 410 1.4006 En 56A Z10C13	C3 SÅ9 2321 431 1.4057 En 57 Z15CN16-02	A2 SÅ2 2333 304 1.4301 En 58E	F1 SÅ1 2326 – 14521 – Z6CN 18-09	C1 2303 (420) 14021 BS 4200537 –	 SÅ10 – S31254(254SMO) – – Z20C13
Analysis Carbon (C) % Chromium (Cr) % Nickel (Ni) % Molybdenum (Mo) % Copper (Cu) % Nitrogen (N) %	 0.09-0.15 12-14 – – – –	 0.17-0.25 16-18 1.25-2.5 – – –	 max 0.07 17-19 8-11 – – –	 0.25 17-19 max 0.2 2.0-2.5 – –	 0.18-0.25 12-14 max 1.0 – – –	 max 0.020 19.5-20.5 17.5-18.5 6-6.5 0.50-1.0 0.18-0.22
Strength of the raw material in annealed condition at 20°C Tensile strength R_m N/mm^2 Yield stress $R_{p0.2}$, N/mm^2 Elongation A_s % Brinell hardness HB	min 440 min 245 min 20 max 220	– – – max 270	min 490 min 205 min 45 max 200	min 440 min 340 min 25 max 200	min 740 – – max 225	min 650 min 300 min 35 approx 210
Physical data (in annealed condition) Magnetic Scaling temperature in air ca. °C	 Yes 820	 Yes 830	 No [3] 850	 Yes 1000	 Yes 820	 No 1000
Machining properties Hardness Weldability	 Yes Bad	 Yes Bad	 No Very good	 Yes Good	 Yes Bad	 No Very good
Notes	Limited corrosion resistance in hardened condition, good strength properties for turbines, pump axles, knives.	Higher corrosion resistance than 2302. In hardened condition, good strength properties.	For normal corrosion conditions, can be used within the food and chemical industries. Household articles, draining boards.	Can be used as replacement for quality 2333 and in water with a high chloride content.	Limited corrosion resistance. In hardened condition, good strength properties. Axles, screws, turbine parts.	Excellent corrosion properties. Completely resistant in seawater. Appropriate for structures in seawater.

[3] With Ni content of less than 10% and in higher property classes, magnetism can occur.

[2] In annealed weld joints not sensitive to intergranular corrosion.

8.5.1 CHOICE OF MATERIAL IN RESPECT OF LIFE

It is most natural to test fasteners in the environments in which they are intended to be used. The disadvantage is that this takes a long time before the result is known. Another method is accelerated testing in artificial atmospheres (for example, salt spray testing). It is important to understand that such testing does not equate well with reality. It only gives comparative values in a certain set of assumptions. Table 8.4 below is based on results from testing in artificial and natural environments.

TABLE 8.4 LIFE IN YEARS

		Industrial atmosphere	Coastal atmosphere
Life on zinc layer	Fzb (Bright zinc coated) Fzv (Hot dip galvanized)	≈ 2.5 years ≈ 15 years	≈ 5 years ≈ 30 years
Probable period of time before the strength of the screw is seriously affected	Stainless A2 Acid-resistant A4	≈ 25 years ≈ 50 years	≈ 50 years ≈ 100 years

8.6 CORROSION IN STAINLESS AND ACID-RESISTANT FASTENERS

8.6.1 PASSIVITY

The corrosion resistance of stainless steel is based mainly on passivity. This means that steel which is alloyed with sufficient chromium in the presence of an oxidising agent, for example, oxygen in the air, is covered by a protective, invisible chromic oxide skin, a so-called passive layer. In very mild corrosion environments, for example, in clean air or oxygenated, fresh water, the lowest alloyed chromium steels also achieve their "stainless" properties. Passivity is improved however, through the addition of more Cr and to some extent also of Ni. Through alloying Mo, a passive layer is obtained, which also resists non-oxidising agents, for example, sulphuric acid and phosphoric acid, within certain limits. A comparison between A2 (SS 2333), A4 (SS 2347) and BUMAX (SS 2343) is given below in relation to the various corrosion types, pitting corrosion, crevice corrosion and general corrosion.

8.6.2 PITTING CORROSION

The strength of a steel in resisting pitting corrosion is determined primarily by its chromium and molybdenum content, and the nitrogen content also has a considerable effect. In order easily to be able to compare the pitting corrosion resistance of different types of steel on the basis of differences in composition, the so-called PRE number (Pitting Resistance Equivalent) is used nowadays. The PRE number is normally expressed as:

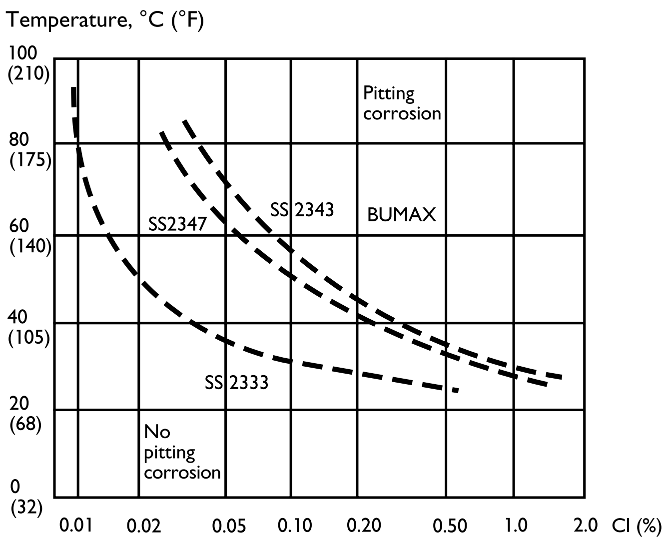
PRE = %Cr+3.3x%Mo+16x%N or %Cr+3.3x%Mo+30x%N

The PRE numbers for SS 2343, SS 2347 and SS 2333 are given in the table below.

TABLE 8.5 PRE NUMBERS

Steel	%Cr	%Mo	%N	PRE
BUMAX SS2343	17.5	2.75	–	26.6
SS 2347	17.5	2.25	–	24.9
SS 2333	18.0	–	–	18.0

On this basis of assessment, SS 2343 is somewhat better than SS 2347. How this difference actually looks in real testing is shown below in Figure 8.1. This shows how the critical pitting corrosion temperature varies with the chloride content in constant potential. The higher up the diagram a curve is situated, the better the pitting corrosion resistance of the material. SS 2343 is somewhat higher than SS 2347, especially with low chloride content.



Critical pitting corrosion temperature in neutral chloride solution.
Potentiostatic testing at +300 mV SCE.
(Saturated Calomel Electrode), reference electrode.

Figure 8.1

8.6.3 CREVICE CORROSION

Crevice corrosion is closely related to pitting corrosion, but is normally initiated at considerably lower temperatures. If, for example, an environment can produce pitting corrosion when the temperature exceeds 60°C, crevice corrosion can normally occur in the same environment at 30-40°C. The importance of Cr, Mo and N is considered equal for all of them as regards pitting corrosion. In a study of pitting and crevice corrosion in various types of stainless steel, it is shown that Mo and N are somewhat more effective in preventing crevice corrosion than pitting corrosion. In the equivalent of the PRE formula (see under Pitting Corrosion), the coefficients for Mo and N are 4.1 and 27 respectively. From of the point of view of crevice corrosion, SS 2343 is therefore even more preferable than SS 2347.

8.6.4 GENERAL CORROSION

Molybdenum improves the behaviour of a steel type in non-oxidising acids, among other things. Table 8.6 shows the result of testing in respect of general corrosion in sulphuric acid (H_2SO_4) at 50°C. The rate of corrosion is expressed in mm/years.

TABLE 8.6 RATE OF CORROSION IN mm/YEARS

Steel	$\%H_2SO_4$			
	3.0	10	15	20
SS 2333	1.08	3.0	–	–
BUMAX i SS 2343	0	0.04	0.33	0.44
SS 2347	0	0.32	–	1.30

The higher Mo content of SS 2343 contributes significantly to a slower corrosion rate. The table also shows that the corrosion properties of SS 2333 (A2) are clearly insufficient. This applies in most acids and other severe environments.

For more detailed information about corrosion and the corrosion properties of stainless and acid-resistant fasteners, please see BULTEN STAINLESS TECHNICAL INFORMATION 11:4 and data sheets from the steel manufacturer.

8.7 PERMEABILITY

Permeability is a measure of the penetrability of a material by magnetic fields, and is designated with the permeability figure k_m. This is a measure of permeability of a material in relation to vacuum, where k_m is 1.0, i.e., completely non-magnetic. In high technology sectors, such as IT, particle research and some areas of the marine industry, low permeability is very important. In Table 8.7, k_m is given for some stainless and acid-resistant fasteners. Here can be seen a clear difference between BULTEN STAINLESS products and products of the A2 steel group. The values of BUMAX 88 and BUMAX 109 are so low that they can be considered completely non-magnetic.

TABLE 8.7
RELATIVE PERMEABILITY OF SOME STEEL AND STRENGTH CLASSES

Steel grade and class	$R_{p0.2}$ N/mm^2	Maximum relative permeability k_m
A2 - 70	450	1.400
A2 - 80	600	1.800
A4 - 80	600	1.012
BUMAX 88	640	1.006
BUMAX 109	900	1.007

8.8 STRENGTH DATA FOR STAINLESS AND ACID-RESISTANT FASTENERS

For normal commercial standards, strength applies in accordance with ISO 3506-1 and 2.

TABLE 8.8 MECHANICAL PROPERTIES OF FASTENERS OF AUSTENITIC STEEL

			Bolt/screw [1]				Nut
Group	Type	Strength	Diam. area	Ultimate tensile strength R_m [2] N/mm² min	Stretch yield point $R_{p0.2}$ N/mm² min	Break elongation A_L [3] min	Test stress Sp N/mm² min
Austenit	A1, A2	50	< M39	500	210	0.6d	500
		70	< M24 [4]	700	450	0.4d	700
	A4	80	< M24 [4]	800	600	0.3d	800
	SS 2343	BUMAX 88	< M36	800	640	0.3d	800
	SS 2343	BUMAX 109	< M12	1000	900	0.2d	1000
	SS 2343	BUMAX 109	M 14-M 20	1000	800	0.2d	1000

1) Refers to finished bolts/screws (not test pieces).
2) All mechanical strength values are calculated with respect to the stress area of the thread in accordance with ISO 3506-1.
3) Break elongation is expressed in mm as part of the nominal bolt/screw diameter. The shortest length of bolt/screw = 2.5d.
4) For classes 70 and 80 over M24, there must be agreement between user and manufacturer

TABLE 8.9 MECHANICAL PROPERTIES OF FASTENERS OF MARTENSITIC AND FERRITIC STEEL

			Bolt/screw [1]			Nut			
Group	Type	Strength	Ultimate tensile strength R_m [2] N/mm² min	Stretch yield point $R_{p0.2}$ N/mm² min	Break elongation A_L [3] min	Test stress N/mm² min	HV min/max	HB min/max	HRC min/max
Martensitic	C1	50	500	250	0.2d	500	–	–	–
		70	700	410	0.2d	700	220/330	209/314	20/34
	C3	80	800	640	0.2d	800	240/340	228/323	21/35
	C4	50	500	250	0.2d	500	–	–	–
		70	700	410	0.2d	700	220/330	209/314	20/34
Ferritic	F1 [4]	45	450	250	0.2d	450	–	–	–
		60	600	410	0.2d	600	–	–	–

1) Refers to finished bolts/screws (not test pieces).
2) All mechanical strength values are calculated with respect to the stress area of the thread.
3) Break elongation is expressed in mm as part of the nominal bolt/screw diameter. The shortest length of bolt/screw = 2.5d.
4) Maximum diameter of F1 = 16 mm

BULTEN STAINLESS is a leader in the development of methods for the manufacture of high-strength fasteners of stainless and acid-resistant material. The company was the first in the world to be able to manufacture class 80 products. BULTEN STAINLESS technicians were also active as regards producing the international standard now in force, ISO 3506-1, which applies to bolts and screws of stainless and acid-resistant steel. BULTEN STAINLESS has now developed class 80 further, and produced BUMAX 88 and BUMAX 109, which are tested in the same way as described in ISO 3506-1, but achieve higher values than those which apply according to the standard. BUMAX 88 and BUMAX 109 have been developed to achieve a strength equal to the property classes 8.8 and 10.9 for low-alloy steel bolts and screws, while they have better qualities otherwise than normal A4. ISO 3506-1 also stipulates the mechanical data for some other types of stainless steel. The high strength achieved with BULTEN STAINLESS manufacturing methods makes possible a new approach on the design side.

By choosing high-strength bolts and screws, either thinner dimensions can be used or fewer bolts and screws. This means, among other things, lower weight and lower cost of the end product. In this context, it is important to point out the following:

In the unthreaded part of a roll-threaded bolt or screw, strength is lower than in the threaded part. Despite this, the bolt or screw is of equal strength through most of the unthreaded part. A finished bolt or screw which has been subsequently subjected to cutting machining, consequently loses the strength it had, which can constitute a serious risk in subsequent handling.

8.9 CHOICE OF PROPERTY CLASS

From an economic point of view, one should always select high strength, i.e., BUMAX 88 or BUMAX 109.

With stronger bolts and screws, one can choose thinner dimensions, giving the following advantages:
- The bolts and screws are cheaper.
- The structure can be made smaller, saving expensive space.
- It is not necessary to drill such large holes, which can give considerable savings in time.

A stronger bolt or screw also gives:
- A harder bolt or screw with less linear expansion, giving a stronger joint even if the temperature rises.
- Less elongation, which is often an advantage.
- Better resistance to pitting.

BUMAX 88 and 109 bolts and screws are to be considered safe fasteners, primarily from a practical point of view. By this is meant that class 50 is so soft and has such a large elongation after fracture that the fitter, at least as regards dimensions up to M16, does not notice that the pretension has stopped increasing and is replaced by plasticity of the screw. Such joints are not safe and often lead to repair costs.

The high strength of BULTEN STAINLESS bolts and screws is due to deformation hardening in the plastic process, in which the elongation limit especially is increased. One can select smaller bolts and screws when selecting BULTEN STAINLESS strong bolts and screws.

8.10 IMPACT STRENGTH

At very low temperatures, low alloy steel is normally brittle, while the austenitic steels retain their toughness and can be used at very low temperatures. The ability of a type of steel to function at these temperatures (down to -200°C) is determined by its impact strength. Table 8.10 shows the results of impact strength testing of bolts and screws of various property classes.

The higher the property class, the lower the impact strength. BUMAX 88 and BUMAX 109 have very good impact strength, despite high property class, especially compared with steel bolts and screws of property classes 8.8 and 19.9.

TABLE 8.10

Property class	Impact strength in Joules (J) at °C				
	+20	–50	–100	–140	–196
A4 -70	96.9	91.9	84.5	80.6	73.4
BUMAX 88	79.5	74.5	69.0	60.0	47.4
BUMAX 109	36.0	34.0	32.9	31.0	29.5

8.11 MARKING

According to ISO 3506-1 and –2, hexagonal bolts, screws and nuts with thread diameters ≥ 5 mm shall be marked with type of steel and property class. See Tables 8.8 and 8.9. This marking shall be followed by the manufacturer's logotype.

Other types of bolt or screw can be marked in the same way if possible. It is also permitted to add a marking if this does not confuse. Figure 8.2 is an example of how our range is marked in accordance with the requirements mentioned.

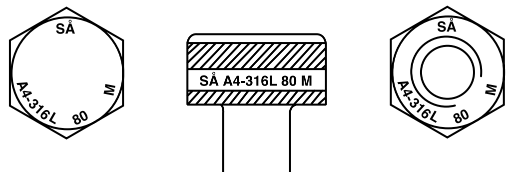


Figure 8.2

Dimensions below M5 can lack some of the marking for reasons of space. Subcontract-manufactured products can also have somewhat deviating markings, but the class of steel should always be stated.

The various symbols denote the following:

SÅ = designation of manufacturer
A4-316L = type – group of material
80 = property class
M = type of thread, metric

UNF and UNC threaded products are not covered by ISO 3506-1 and -2. BULTEN STAINLESS does, however, also manufacture these products with property class data in accordance with ISO 3506-1 and -2.

8.12 TIGHTENING TORQUE AND FORCE

For a bolted joint to be intact, it is important that in assembly the correct preload is achieved. The values in Table 8.11 are calculated to give a preload of about 65% of the proof stress ($R_{p0.2}$). The correct preload can only be achieved if one has a joint with low friction.

If no lubricant is used, the preload can vary considerably and can be as low as about 20% of the stated value. This is because the tightening torque is used to bridge friction instead of tightening the joint. BUMAX 88 and BUMAX 109 are treated with a high-class lubricant to achieve low friction. For these products, the scatter of the preload values amounts to about +/-15% in assembly in nuts with a calibrated torque wrench.

See also Chapter 9 “How hard should one tighten?”

TABLE 8.11
TIGHTENING TORQUE AND FORCES FOR A2 AND A4 AND BUMAX 88 AND 109, M THREADS, WAXED PRODUCTS

Description	Class	Screw/bolt diameter														
		M3	M4	M5	M6	M8	M10	M12	M14	M16	M18	M20	M24	M27	M30	M36
Tightening	50	0.4	1.0	1.9	3.3	7.8	15	27	43	65	91	127	220	318	434	755
torque	70	0.9	2.0	4.1	7.0	17	33	57	91	140	195	273	472	682	930	1620
Mv i Nm	80	1.2	2.7	5.4	9.3	22	44	76	121	187	261	364	629	909	1240	2160
	BUMAX 88	1.3	2.9	5.7	9.8	25	47	82	129	198	275	385	665	961	1310	2280
	BUMAX 109	1.7	4.1	8.1	14	34	66	115	161	248	344	481				
Prestressing force	50	0.8	1.4	1.9	2.7	5.0	7.8	12	16	21	27	33	48	63	77	112
average	70	1.5	2.6	4.2	5.9	11	17	25	34	47	56	72	103	134	164	239
kN	80	2.0	3.4	5.5	7.8	14	23	33	45	61	75	96	138	179	219	319
	BUMAX 88	2.1	3.6	5.9	8.4	15	24	35	48	65	80	102	181	235	287	418
	BUMAX 109	2.9	5.2	8.6	12	21	34	49	60	81	100	128				
Breaking force	50	2.5	4.4	7.1	10	18	29	42	58	79	96	123	177	230	281	409
kN	70	3.5	6.1	9.9	14	26	41	59	81	110	134	172	247	321	393	572
	80	4.0	7.0	11	16	29	46	67	92	126	154	196	282	367	449	654
	BUMAX 88	4.0	7.0	11	16	29	46	67	92	126	154	196	282	367	449	654
	BUMAX 109	5.0	8.8	14	20	37	58	84	115	157	192	245				
Yield load	50	1.3	2.2	2.9	4.2	7.7	12	18	24	33	40	51	74	96	118	172
kN	70	2.2	3.9	6.4	9	16	26	38	52	71	86	110	159	207	253	368
	80	3.0	5.3	8.5	12	22	35	51	69	94	115	147	212	275	337	490
	BUMAX 88	3.2	5.6	9.1	13	23	37	54	74	100	123	157	226	294	359	523
	BUMAX 109	4.5	8.0	13	18	33	52	76	93	125	154	196				
Pretension area	mm^2	5.03	8.78	14.2	20.1	36.6	58	84	115	157	192	245	353	459	561	817
Scatter	mm	0.6	0.7	0.8	1.0	1.25	1.5	1.8	2.0	2.0	2.5	2.5	3.0	3.0	3.5	4.0

8.13 YIELD STRESS IN TENSION AT HIGH TEMPERATURES

When using fasteners at high temperatures, it is important to know what strength can be achieved at the temperatures in question. The values available for the strength of a steel type at a raised temperature are normally those which the steel manufacturers include in their data sheets. These values thus apply only for the steel in its delivered capacity, which is normally soft-annealed condition and cannot be used for high-strength fasteners, which achieve their strength through cold deformation.

BULTEN STAINLESS has tested yield stress in tension at high temperatures for:

1. Hexagonal bolt/screw BUMAX 88 with dimension M 10 x 100 mm
2. Hexagonal bolt/screw BUMAX 109 with dimension M 10 x 100 mm.

The survey was carried out with Sandvik Steel Research Center and is shown as a graph in Figure 8.3. The result of the test shows that BUMAX bolts and screws have a very high remaining strength, even at relatively high temperatures. As it is a question of joints requiring high strength and which are exposed to high temperatures for long periods, instead of the proof stress values, one should calculate the strength using the so-called creep-rupture values. In such case, one should select more distinctive high temperature steels in order to achieve a secure joint. See BULTEN STAINLESS' brochure "Fasteners for high temperatures".

8.14 LINEAR EXPANSION COEFFICIENTS

Linear expansion coefficients in austenitic steel are affected by temperature and the condition the material happens to be in. For SS 2343 and SS 2333 in annealed condition, the values are at about the same level for both types of steel. Through cold working, the coefficient is reduced, and then somewhat more for SS 2333 A2-80 than for SS 2343 BUMAX 88. The degree for cold working is about 30%. This is shown in Figure 8.4.

8.15 PROPERTY CLASSES FOR NUTS

Nuts in steel groups A2 and A4 are cold formed in the dimension areas of M 6 – M 24 and 1/4 UNC – 1 UNC respectively. They are manufactured in various property classes equivalent to those for bolts and screws and are marked in the same way. See Section 8.11 and Tables 8.8 and 8.2. Bolts and nuts are combined so that they have the same property class, but nuts of a higher class can always replace nuts of a lower class.

8.16 HARD WASHERS

In the case of high prestressing force, when the strength of bolts and screws is utilised optimally, a hard underlay is required under the bolt head and nut. If the underlay is not hard enough, this can lead to settling and deformation. In structures with BUMAX 109, the hardness of the underlay should be at least HB 300. Equivalent hardness for BUMAX 88 is HB 250. BULTEN STAINLESS stocks washers which are adapted for the high mechanical property values of these bolts and screws.

Figure 8.3

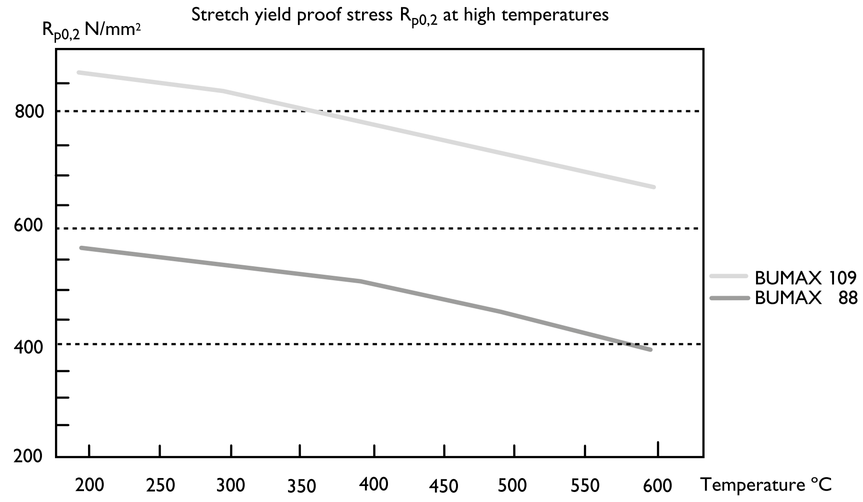
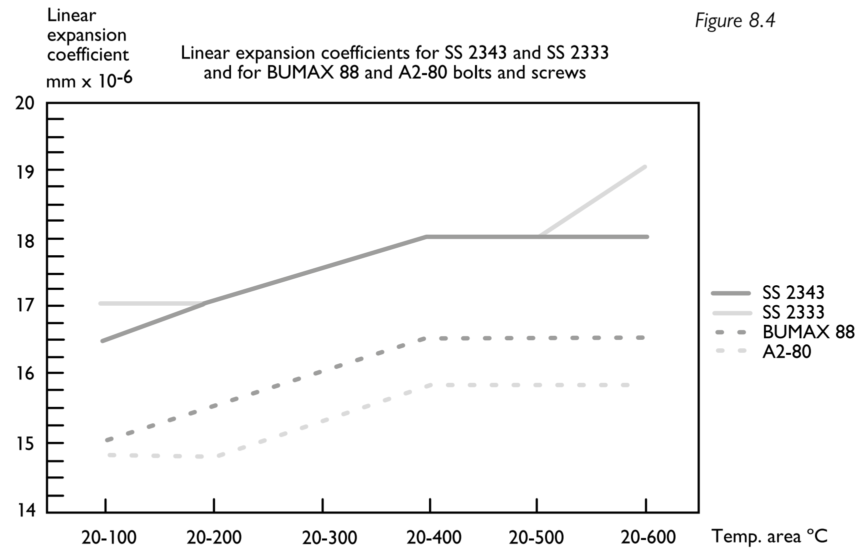
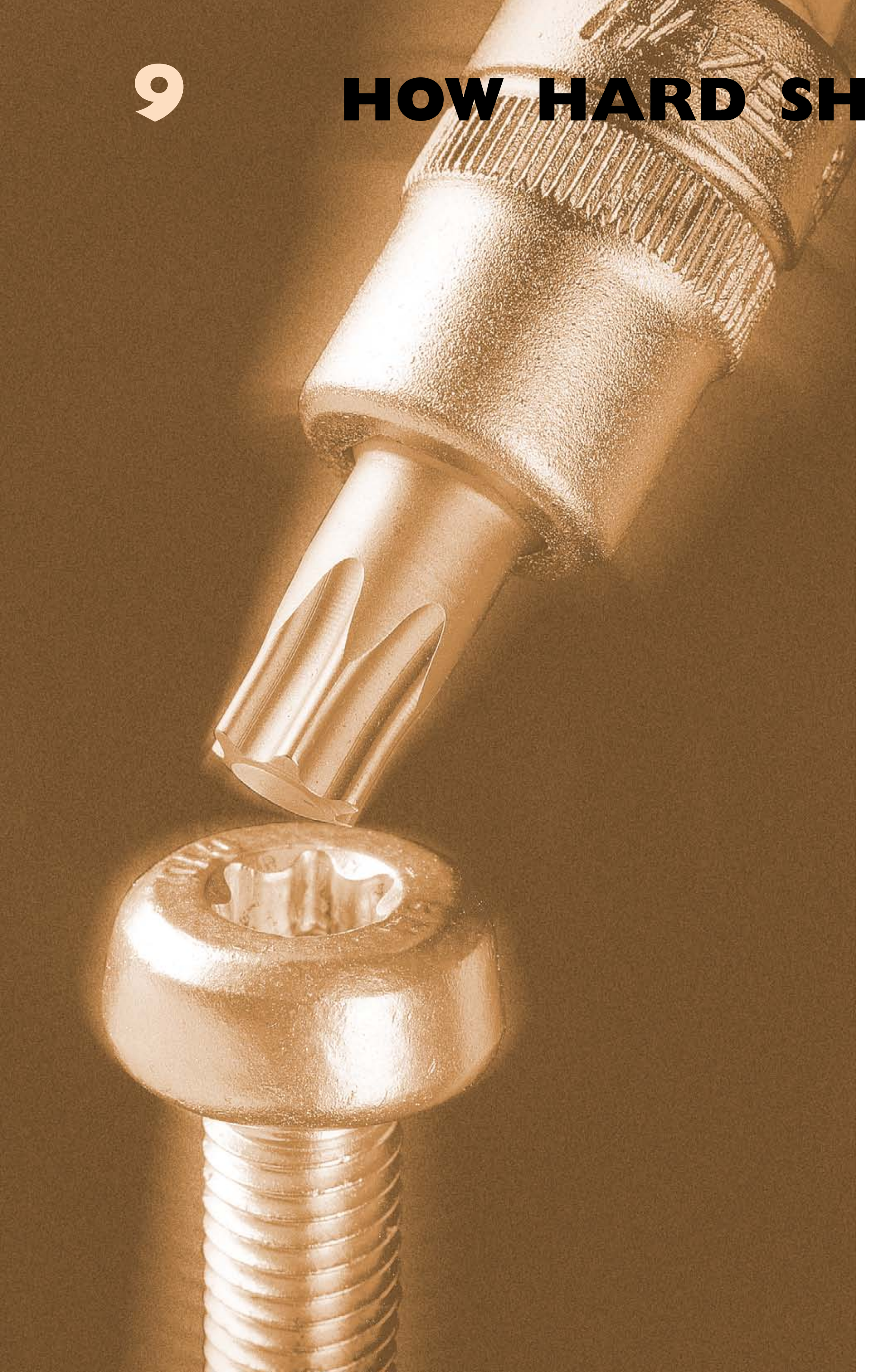


Figure 8.4





9 HOW HARD SHOULD ONE TIGHTEN?

9.1 PRETENSION AND TIGHTENING TORQUE

For a bolted joint to function correctly and resist strong static or alternating forces for a long time, the bolts must be pretensioned. Pretension shall normally be maintained at such a level that the combined tension of tensile stress and torsional stress in the bolt does not exceed the yield stress of the bolted material. It is always necessary to adapt the friction and tightening methods to the strength of the bolt.

The main task of the prestressing force is to clamp the joined parts together. However, this does not prevent the bolt and nut from slackening and causing frictional forces between the joined parts, which increases resistance to transverse forces.

A joint with high pretension becomes smaller, lighter and cheaper for a given tensile traction, and also less sensitive to fatigue than a joint with low pretension.

The pretension achieved in assembly can subsequently become less, due to settling in the clamped material, i.e., some of the elastic deformation becomes permanent. This risk is especially serious when the joint is clamped over visco-elastic material (cork or rubber packing, painted surfaces, etc.) In order that the pretension should not decrease too much, the bolted joint should have some length of clamp. The elastic elongation of the bolt can then counteract some of the settling.

The design of the joint can have a considerable effect, but for inelastic joints, one can apply the following rule:

length of clamp ≥ diameter of thread, however at least 20 mm.

9.2 PRETENSION METHODS AND PRETENSION PROCEDURE

Various procedures can be used to pretension a bolted joint. The simplest is to tighten "by feel" by hand, or use a percussive machine without a torque bar. However, this gives little or only minimal control of the pretension. In order to achieve a predetermined prestressing force in the joint, it is necessary to use other procedures depending on the demands made on the joint as regards degree of pretension, the scatter of the stress force and the risk of slackening. The procedure most used is the torque control method, i.e., tightening with a specific torque – the tightening torque. For this, a torque wrench is used, or a machine with torque control, for example, a percussive machine with a torque bar or a stall tool.

If the bolted joint is to be used optimally, methods are used which stress the bolts to such an extent that they begin to enter the plastic area. This can be achieved with the angle of rotation method (torque – angle method) or with the yield stress method. The scatter of the prestressing force is then much less.

If high demands are made on the bolted joint, or when there is a risk of personal injury or serious mechanical malfunction if the bolted joint does not work, the pretension procedure and pretension data must be determined through practical tests.

9.3 DIVISION OF TIGHTENING TORQUE

The aim of tightening is to achieve a prestressing force. The greater part of the torque however, is expended to combat friction in the thread and against the contact surface of the head or nut. In unfavourable friction circumstances, perhaps only 10% of the torque remains to achieve the prestressing force. With lubrication methods, it is however, possible for more, but hardly more than 20%, to be utilised to achieve pretension.

Division of tightening torque in normal circumstances.

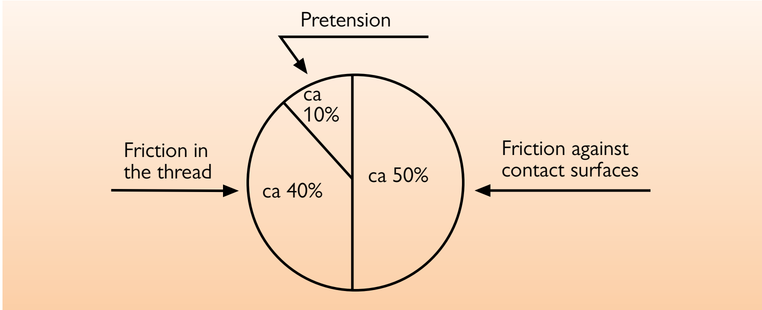


Figure 9.1

9.4 FACTORS WHICH DETERMINE TIGHTENING TORQUE

A rule which a designer should adhere to is that no part of the structure should be loaded in excess of the yield stress in tension. A certain safety factor should also be applied.

A considerable part of torque, about 40%, is used to combat thread friction, resulting in great intensity of torsional stress in the screw or bolt. Torsional stress varies with friction conditions, but constitutes between 10% and 30% of total stress. The effective stress, which is the combination of tensile stress and torsional stress, should not be more than the same as the yield stress in tension. Stress in continuous operation should, however, not exceed the pretension.

Friction depends on the material, surface evenness or surface treatment and lubrication condition of the screw or bolt, nut and contact surfaces (including that of any washers). The surface condition of the thread and use of lock fluid, sealing agent or lubricant affects friction in the thread. Painted contact surfaces can reduce friction and increase pretension, but as pointed out initially, there is also a considerable risk that pretension will decline over time through settling.

Consideration should also be given to uncertainty. The friction coefficient is not constant, which produces a scatter of the prestressing force. Above all, screwdriving machines and impact wrenches, and also torque wrenches, can cause considerable scatter of the torque applied. The torque should therefore not be so high that effective stress exceeds the yield stress of the screw or bolt when pretension is at its maximum.

With a torque wrench of high quality, one can come down to ± 5% scatter of torque (about ± 16% scatter of pretension in normal circumstances).

When a percussive machine is used with torque bar, one can perhaps only get down to a scatter with the torque of ± 20% (about ± 32% scatter of pretension under normal circumstances). During recent years, stall tools have come about, which give only small scatter of torque, as with a high quality torque wrench, i.e., about ± 5%.

In series manufacture, scatter is greater, but it should be possible to demand a maximum of ± 15% scatter of torque with the use of stall tools. If the function of the bolted joint is important, the tightening torque and tolerance of torque should be specified, or the pretension procedure should be specified and also lubrication conditions. The type of screw, bolt or nut can also affect the tightening torque required. For example, screws and bolts with countersunk heads and flange screws and bolts require greater torque than equivalent screws or bolts, the heads of which have a flat underside, as their contact surface is greater. For stud screws and set screws, on the other hand, torque should be reduced, because there is no contact surface.

When mounting self-tapping (thread-forming) screws, torque cannot be easily calculated, as the screws simultaneously make the threads.

Section 9.7.2 gives information for conversion or testing of tightening torques for various types of screw/bolt or nut.

9.5 CALCULATION OF TIGHTENING TORQUE

In order to determine tightening torque, it is necessary to know the following

1. Type of screw or bolt/type of nut
2. Thread diameter of screw or bolt/nut
3. Property class of screw or bolt/nut
4. Friction conditions
5. Tightening procedure

Various tables are available in the technical literature. The tables for a couple of situations which often occur – tightening of untreated, oiled steel bolted joints and of waxed, stainless bolted joints with torque wrench or screwdriving machine/impact wrench with controlled torque (torque scatter max ± 5%) – are given in Tables 9.2 – 9.4, pages [71 – 73].

For other procedures or friction conditions, and for certain types of screw, bolt or nut, values must be recalculated. See Sections 9.7.1 and 9.7.2. The ways in which the tables shall be used are described later, but first some words about the theoretical background.

The tightening torque (Mv) in the tables has been calculated on the basis of the following formula. It is a development of the formula presented by Kellerman and Klein of Germany in the mid-1950s. Different formulae can give different results, but usually the greatest significance is by how much the calculation values used deviate from real conditions.

$$M_v = \frac{k}{\kappa\left(1 + \frac{S_F}{F_{Fm}}\right)} (d + P) \times A_s \times \sigma_s \times 10^{-3}$$

The constituent factors are as follows:

M_v = tightening torque, Nm

k = factor in the torque equation (see following explanation)

κ = relationship between effective stress and tensile stress (see following explanation)

S_F = scatter of pretension in tightening

F_{Fm} = $\sigma_F \times A_s$ = median preload, N

d = the major diameter of the screw thread, mm

P = thread pitch, mm

A_s = stress area of thread, mm^2. See Table 9.2

σ_s = the general designation of $R_{p0.2}$ or R_{eL} in formulae, N/mm^2

$R_{p0.2}$= stress yield stress at 0.2% elongation, N/mm^2

R_{eL} = lower yield stress in tension, N/mm^2

The factor k takes account of the effect of the thread pitch and friction on torque and in its basic form is expressed:

$$k = \frac{d_2 \times \tan(\varphi + \rho') + D_k \times \mu_u}{2(d + P)}$$

Thanks to the geometric connection and when one, for the most part, has the same friction coefficient in thread and contact surface, k can instead be written:

$$k = \frac{\left[0.161 \times P + \mu_{tot} \; (0.583 \times d_2 \; + 0.5 \times D_k)\right]}{d + P}$$

An analysis of the k value in various thread diameters and friction conditions shows that the discrepancy is not greater than about ± 5% if the expression is finally simplified to:

$$k = 1.078 \times \mu_{tot} + 0.0168$$

The constituent factors are as follows:

d_2 = the average diameter of the screw thread, mm

φ = the angle of pitch of the thread

ρ' = the friction angle of the thread (depending on the friction coefficient μg in the thread, and is obtained from tan $\rho' = \mu_g$)

D_k = the friction diameter of the contact surface, mm

μ_u = the friction coefficient of the contact surface (see Table 9.1).

μ_{tot} = the friction coefficient active in torque-force exchange (see Table 9.1)

The factor κ takes account of the torque stress which arises in the screw or bolt as a result of thread friction. The torque stress reduces the possibility of loading the screw axially. Using the working hypothesis of deviation for calculation of an effective stress (comparison stress), the following is obtained:

$$\kappa = \frac{\sigma_e}{\sigma_F} = \sqrt{1 + \frac{12}{d^2_{A_s}} \left(\frac{P}{\pi} + 1.155 \times \mu_g \times d_2 \right)^2}$$

The constituent factors are as follows:

σ_e = effective stress, which can be maximum = σ_s N/mm^2

σ_F = the pretension of the screw or bolt, N/mm^2

$d_{A_s} = \sqrt{4 A_s/\pi}$ = diameter of the stress area, mm

P = thread pitch, mm

μ_g = friction coefficient of the thread (see Table 9.1)

d_2 = the average diameter of the screw/bolt thread, mm

The values of k and κ obtained from the formulae with various friction coefficients (i.e., different material, surface qualities and lubrication) are given in Table 9.1. The value of κ is mainly due to the friction coefficient of the thread (μ_g), and in the table has therefore been given as independent of thread size, and value of k.

9.6 DEGREE OF PRETENSION

The relationship between pretension (σ_F) and the yield stress in tension of the screw/bolt or its proof stress (σ_s) is called the degree of pretension and is calculated from the formula:

$$G_F = \frac{F_{Fm}}{F_s} = \frac{\sigma_F}{\sigma_s} = \frac{1}{\kappa \left(1 + \frac{S_F}{F_{Fm}}\right)}$$

The constituent factors are as follows:

G_F = degree of pretension

$F_{Fm} = \sigma_F \times A_s$ = mean prestressing force, N

$F_s = \sigma_s \times A_s$ = the yield load of the screw or bolt, N

σ_F = the pretension of the screw or bolt, N/mm^2

σ_s = general designation of $R_{p0.2}$ or R_{eL} in formulae, N/mm^2

S_F = the extent of pretension scatter in tightening, N (see Table 9.1).

The degree of pretension cannot be freely chosen. The possible pretension is limited both by the friction conditions and by the uncertainty of the tightening. Some friction conditions and a certain amount of tightening procedure therefore give a determined degree of pretension because the effective stress is not permitted to be greater than the nominal yield stress in tension (σ_s). The degree of tensile stress (G_F) of various friction coefficients with tightening using a torque wrench, screwdriving machine or impact wrench with a maximum of ±5% torque scatter is shown in Table 9.1.

9.7 USING THE TABLES

So that you can decide on tightening torque more easily, we have included tables with calculation values, conversion factors for various friction conditions and calculated tightening torques for conditions which commonly occur. The tables apply for bolted joints with screws, bolts and nuts that have plain contact surfaces at right-angles to the axis of the thread with an major diameter of about 1.5 d (d = the thread's nominal major diameter). For other contact surfaces and for tapping (thread-forming) screws, the tightening torque must be converted in accordance with Section 9.7.2.

The tightening torque given in the tables consists of recommendations with experience from assembly cases and laboratory tests. The values can be used for general assembly, where one can accept the scatter of the pretension which is denoted by S_F/F_{Fm} in Table 9.1.

If an exactly determined pretension is required, it is necessary to carry out test assemblies for the assembly in question. It can even be necessary to study friction separately for thread and contact surface when different materials come into contact with one another. It can also be necessary to choose special combinations of surface treatment and lubricants in order to obtain the desired qualities. Tightening machines with special qualities can also be necessary.

If extremely high utilisation of the screws or bolts is demanded, it can be necessary to pretension them in the plastic area. This is achieved in accordance with the angle of rotation method (torque angle method) or in accordance with the yield stress method. In the case of series assembly, computerised systems are used in accordance with these methods.

9.7.1 CONVERSION FOR VARIOUS FRICTION CONDITIONS

Table 9.1 contains values which can be used in the formulae in calculating tightening torque. Also in the tables, a conversion factor (C) is used for converting tightening torque taken from other tables, so that it can be applied to other friction conditions.

The conversion factor (C) is 1.00 for steel bolts and nuts which are untreated (not surface treated) before receiving a light oil coating, as the two torque tables below for steel screw joints refer to this combination. In the case of more effective lubrication, for example, with molybdenum disulphide, (MoS_2), friction is reduced, by which the conversion factor becomes 0.86, i.e., a 14%.reduction of the table value for tightening torque.

The table also shows that the degree of pretension (G_F) increases from 0.71 to 0.75 when lubricated with molybdenum sulphide, thanks to lower torsional stress (lower κ) and less pretension force scatter (S_F), despite the fact that the torque is reduced. If one selects screws and nuts which are untreated and dry instead of untreated and with a light coating of oil, the friction increases, but despite that, torque should be reduced because the conversion factor is reduced from 1.00 to 0.96. The reason for this is that the higher friction increases the intensity of torsional stress, while at the same time, scatter of the pretension force increases. Therefore, the degree of pretension (G_F) must be reduced so that the effective stress does not exceed the yield stress in tension (σ_s). The degree of pretension thus provides valuable information regarding how well the joints are utilised.

The conversion factor (C) is also 1.00 for stainless steel bolts and nuts which are waxed, because the last torque table refers to this combination. Lubrication with oil or emulsion instead increases friction and pretension force scatter, which means that the torque should be reduced in this case as well (the conversion factor is reduced from 1.00 to 0.84). Otherwise, the yield stress in tension (σ_s) could be exceeded.

9.7.2 CONVERSION FOR CERTAIN TYPES OF SCREW AND BOLT /NUT

The tightening torque which is obtained from the Torque Tables 9.2-9.4, after any conversion to other friction relationships in accordance with Section 9.7.1, sometimes also needs to be converted with reference to type of screw, bolt or type of nut.

Such conversion is required when the screw, bolt or nut which one is tightening has a contact surface which is not flat and at right-angles to the axis of the thread, or the major diameter of which substantially deviates from the size of 1.5d (d = the thread's nominal major diameter).

For screws and nuts of more deviating size or deviating shape of the contact surface, recommendations are given below for conversion of the tightening torque.

When considerable exactitude is required, it is however necessary to establish the replacement of torque/force through testing in order to be able to determine appropriate tightening torques. The torque cannot be easily calculated for mounting self-tapping screws either, because the screws simultaneously make the threads and the form of screw thus has an effect. Standard SS 3392, which is included in SMS handbook 517, gives instructions regarding how torque can be determined for such screws.

9.7.2.1 *Screws and Bolts with Countersunk Heads*

On account of the size of the contact surface and countersink angle of the screws and bolts, greater frictional force against the contact surface is needed, and therefore the tightening torque must be increased by about 30%.

9.7.2.2 *Collar Screws and Bolts and Collar Nuts*

These also have a greater contact surface than normal screws, bolts and nuts, and thus greater friction radius, for which reason the tightening torque must be increased by about 10%.

9.7.2.3 *Set Screws*

When assembling with set screws, no friction needs to be overcome against any contact surface. On the other hand, the resistance of the underlay against screwing must be overcome. The shape of the tip of the screw and the type of underlay (flat or cylindrical surface, pre-drilled hole, etc.) has an effect, but the torque required is 50%-70% of that required for normal screws. Set screws which are pointed require lower torque, while set screws with a flat-fronted end or with a recess at the end have a greater friction radius, and therefore require higher torque.

9.7.2.4 *Stud Bolts*

To mount stud bolt joints it is necessary to tighten the bolt in the threaded hole first, and then tighten the nut of the joint. In tightening the bolt, the torque does not need to overcome friction against any contact surface. According to the explanation given above about the division of tightening torque, tightening the nut should give roughly half of the torque required for pretension of the bolt.

9.8 TIGHTENING TORQUE FOR STEEL BOLTED JOINTS

A steel bolted joint of property class 8.8 in accordance with SS-EN20898-1 with M10 thread requires a tightening torque of 47 Nm, according to Torque Table 9.2 for steel bolted joints with metric coarse threads. An increase of strength to property class 12.9 increases the torque requirement to 79 Nm. A bolted joint in the lowest property class included, which is 4.6, requires 17 Nm for M10, less than 1/4 of that required for property class 12.9. From this it can be seen how important it is to adapt torque to property class and not just to the diameter of the bolt.

In many cases, the manufacturers of screwdriving machines and impact wrenches state that a machine is appropriate for a certain diameter.

This information is, however, of no value at all for the user. What the user needs to know is what torque range a machine is usable for. According to the example, a torque of 79 Nm was appropriate for property class 12.9 and bolt M10. Roughly the same torque, 81 Nm is required for a screw/bolt of property class 8.8 with thread diameter M12. In these two cases, a machine which gives 75-90 Nm can be selected.

The following examples show how the tables can be used:

Hexagonal head bolts M10 of property class 8.8 and nuts of property class 8 and washers of a hardness minimum 200 Hb. All fasteners are bright zinc plated and dry. For tightening, a screwdriver shall be used with torque control which can be set, accurate to within ± 5% scatter.

From the torque table, the tightening torque of M_v = 47 Nm is obtained for untreated, lightly oiled, steel, bolted joints. Similarly, σ_s = 640 N/mm^2 is obtained and A_s = 58 mm^2.

This gives a yield load of:
$F_s = \sigma_s \times A_s = 640 \times 58$ N = 37120 N = 37.1 kN

From Table 9.1, the following calculation values are obtained with conversion factor for friction:

$S_F/F_{Fm} = \pm 0.29$ $G_F = 0.62$ $C = 0.96$

No recalculation is required with reference to the type of bolt.

This gives:

Tightening torque
$= M_v \times C = 47 \times 0.96$ Nm = 45 Nm

Mean pretension force
$F_{Fm} = F_s \times G_F = 37.1 \times 0.62$ kN = 23 kN

The pretension force scatter

$$S_F = \frac{S_F}{F_{Fm}} \times F_{Fm} = \pm 0.29 \times 23 \text{ kN} = \pm 6.7 \text{ kN}$$

9.9 TIGHTENING TORQUE FOR STAINLESS BOLTED JOINTS

In order for stainless bolts (including so-called acid resistant bolts) to be given pretension, effective lubrication is required – otherwise the threads jam up. The mechanical property values in accordance with SS-ISO 3506-1 for stainless bolts do not conform to those for normal steel bolts. A separate torque table, 9.4, has therefore been included for stainless bolted joints. The torque values are for products which are waxed, which is considered to be a normal condition. See also Chapter 8.12. It should be pointed out that percussive machines should not be used in tightening stainless bolted joints. The degree of pretension in this case is used in comparison with other friction conditions. See Table 9.1.

Lubrication with molybdenum disulphide (MoS_2) gives a similar friction condition to that obtained from waxing. The following examples show how the table can be used:

A hexagonal head bolt M10 of property class A4 – 80 is pretensioned with a waxed nut of the same property class. Tightening is performed with a torque wrench on the nut.

From Torque Table 9.4, a tightening torque of $M_v = 44$ Nm is obtained for waxed stainless bolted joints.
Similarly $\sigma_s = 600$ N/mm^2 $A_s = 58$ mm^2 is obtained.

This gives a yield load of $Fs = \sigma_s \times A_s = 600 \times 58$ N = 34800 N = 34.8 kN

From Table 9.1, the following conversion values and conversion factors are obtained for the friction relationship:

$S_F/F_{Fm} = \pm\ 0.23 \quad G_F = 0.65 \quad C = 1.00$

No conversion is required in respect of the type of bolt.

This gives:
Tightening torque
$= M_v \times C = 44 \times 1.00$ Nm $= 44$ Nm

Mean pretension force
$F_{Fm} = F_S \times G_F = 34.8 \times 0.65$ kN $= 22.6$ kN

The pretension force scatter

$$S_F = \frac{S_F}{F_{Fm}} \times F_{Fm} = \pm\ 0.23 \times 22.6 \text{ kN} = \pm\ 5.2 \text{ kN}$$

9.10 TABLES

The section contains an assisting table with conversion values, conversion factors and torque tables.

List of Tables:

TABLE 9.1

CALCULATION VALUES AND CONVERSION FACTORS (C) FOR VARIOUS FRICTION CONDITIONS (TIGHTENING WITH TORQUE WRENCH OR SCREWDRIVING MACHINE/IMPACT WRENCH WITH TORQUE CONTROLS. TORQUE DEVIATION MAXIMUM ± 5%).

Material, surface condition 1) Screw/bolt	Nut or thread in the material	Lubrication condition	μ_{tot} μ_g μ_u	$\pm \frac{S_F}{F_{Fm}}$	k	κ	G_F	C 3)
Steel, untreated	Steel, untreated	dry	0.14	0.29	0.168	1.24	0.62	0.96
		oil	0.125	0.16	0.152	1.21	0.71	1.00
		MoS_2	0.10	0.16	0.125	1.15	0.75	0.86
		wax	0.06	0.11	0.082	1.08	0.83	0.63
Steel. phos	Steel, phos or untreated	dry	0.125	0.29	0.152	1.21	0.64	0.90
		oil	0.10	0.16	0.125	1.15	0.75	0.86
		MoS_2	0.08	0.11	0.103	1.11	0.81	0.77
		wax	0.06	0.11	0.082	1.08	0.83	0.63
Steel, fzb, fzy or fzm	Steel, fzb, fzy, fzm or untreated	dry	0.14	0.29	0.168	1.24	0.62	0.96
		oil/emulsion	0.10	0.16	0.125	1.15	0.75	0.86
		wax	0.06	0.11	0.082	1.08	0.83	0.63
	Light metal	oil/emulsion	0.125	0.23	0.152	1.21	0.67	0.94
Steel ZIB	Steel ZIB or untreated	gleitmo® 602	0.14	0.29	0.168	1.24	0.62	0.96
		or gleitmo® 603	0.14	0.29	0.168	1.24	0.62	0.96
Steel, fzv	Steel, fzv or untreated	oil (delivered condition)	0.14	0.16	0.168	1.24	0.69	1.07
		dry	0.20	0.29	0.232	1.41	0.55	1.17
		oil/emulsion	0.14	0.16	0.168	1.24	0.69	1.07
		wax	0.06	0.11	0.082	1.08	0.83	0.63
	Light metal	oil/emulsion	0.16	0.29	0.189	1.29	0.60	1.04
Steel. epoxy	Steel, epoxy or untreated	dry	0.20	0.29	0.232	1.41	0.55	1.17
		oil	0.14	0.16	0.168	1.24	0.69	1.07
		emulsion	0.10	0.16	0.125	1.15	0.75	0.86
		wax	0.06	0.11	0.082	1.08	0.83	0.63
Stainless steel 2)	Stainless steel	wax	0.14	0.23	0.168	1.24	0.65	1.00
	or light metal 2)	oil/emulsion	0.20	0.29	0.232	1.41	0.55	0.84

1) Untreated = untreated, phos = phosphated, fzb = zinc electroplated + bright chromated, fzy = zinc electroplated + yellow chromated, fzm = mechanically zinc coated, ZIB = zinc/iron with black chromating, fzv = hot dip galvanized, epoxy = coating based on epoxy resin.
2) Stainless steel also covers so-called acid resistant steel, as in SS-ISO 3506-1.

3) The conversion factor C in conversion of tightening torque to some other material, other surface treatment or other lubrication condition has been set at 1.00 for untreated, unoiled, steel bolts and nuts and for stainless steel, waxed bolts and nuts. The torque in the tables below refers to these combinations. It is easy to recalculate to other combinations by multiplying by the factor C, which applies according to the table

TABLE 9.2

TIGHTENING TORQUE (M_V) IN Nm FOR UNTREATED, LIGHTLY OILED STEEL BOLTED JOINTS WITH THE USE OF TORQUE WRENCH OR SCREWDRIVING MACHINE/IMPACT WRENCH WITH TORQUE CONTROL. (TORQUE DEVIATION MAXIMUM ± 5%). METRIC COARSE THREADS.

Thread M	d mm	P mm	A_s mm^2	Property class in accordance with SS-EN 29898-1				
				4.6	5.8	8.8	10.9	12.9
1.6	1.6	0.35	1.27	0.065	0.10	0.17	0.24	0.29
1.8	1.8	0.35	1.70	0.096	0.16	0.25	0.36	0.43
2	2	0.4	2.07	0.13	0.22	0.35	0.49	0.58
2.2	2.2	0.45	2.48	0.17	0.29	0.46	0.64	0.77
2.5	2.5	0.45	3.39	0.26	0.44	0.70	0.98	1.2
3	3	0.5	5.03	0.46	0.77	1.2	1.7	2.1
3.5	3.5	0.6	6.78	0.73	1.2	1.9	2.7	3.3
4	4	0.7	8.78	1.1	1.8	2.9	4.0	4.9
4.5	4.5	0.75	11.3	1.6	2.6	4.1	5.8	7.0
5	5	0.8	14.2	2.2	3.6	5.7	8.1	9.7
6	6	1	20.1	3.7	6.1	9.8	14	17
7	7	1	28.9	6.1	10.1	16.1	23	27
8	8	1.25	36.6	8.9	15	24	33	40
10	10	1.5	58	17	29	47	65	79
12	12	1.75	84.3	30	51	81	114	136
14	14	2	115	48	80	128	181	217
16	16	2	157	74	123	197	277	333
18	18	2.5	192	103	172	275	386	463
20	20	2.5	245	144	240	385	541	649
22	22	2.5	303	194	324	518	728	874
24	24	3	353	249	416	665	935	1120
27	27	3	459	360	600	961	1350	1620
30	30	3.5	561	492	819	1310	1840	2210
33	33	3.5	694	663	1100	1770	2480	2980
36	36	4	817	855	1420	2280	3210	3850
39	39	4	976	1100	1830	2930	4120	4940
42	42	4.5	1121	1360	2270	3640	5110	6140
45	45	4.5	1306	1690	2820	4510	6340	7610
48	48	5	1473	2040	3400	5450	7660	9190
52	52	5	1758	2620	4370	6990	9830	11800
56	56	5.5	2030	3270	5440	8710	12200	14700
60	60	5.5	2362	4050	6750	10800	15200	18200
64	64	6	2676	4900	8170	13100	18400	22000
68	68	6	3055	5910	9860	15800	22200	26600
72	72	6	3460	7060	11800	18800	26500	31800
76	76	6	3889	8340	13900	22200	31300	37500
80	80	6	4344	9770	16300	26100	36600	44000
85	85	6	4948	11800	19600	31400	44200	53000
90	90	6	5591	14000	23400	37400	52700	63200
95	95	6	6273	16600	27600	44200	62200	74600
100	100	6	6995	19400	32300	51700	72700	87300
Nominally $\sigma_s = R_{eL}$ or $R_{p0.2}$ N/mm^2				240	400	640	900	1080
$\frac{k}{\kappa\left(1+\frac{S_F}{F_{Fm}}\right)} \times \sigma_s$ N/mm^2				26.16	43.60	69.76	98.10	117.72

TABLE 9.3

TIGHTENING TORQUE (M_V) IN Nm FOR UNTREATED, LIGHTLY OILED STEEL BOLTED JOINTS WITH THE USE OF TORQUE WRENCH OR SCREWDRIVING MACHINE/IMPACT WRENCH WITH TORQUE CONTROL. (TORQUE DEVIATION MAXIMUM ± 5%). METRIC FINE THREADS.

Thread	d	P	A_s	Property class in accordance with SS-EN 20898-1				
M	mm	mm	mm^2	4.6	5.8	8.8	10.9	12.9
2 x 0.25	2	0.25	2.45	0.14	0.24	0.38	0.54	0.65
2.2 x 0.25	2.2	0.25	3.03	0.19	0.32	0.52	0.73	0.87
2.5 x 0.25	2.5	0.25	3.70	0.28	0.46	0.74	1.0	1.2
3 x 0.35	3	0.35	5.60	0.49	0.82	1.3	1.8	2.2
3.5 x 0.35	3.5	0.35	7.90	0.80	1.3	2.1	3.0	3.6
4 x 0.5	4	0.5	9.79	1.2	1.9	3.1	4.3	5.2
4.5 x 0.5	4.5	0.5	12.8	1.7	2.8	4.5	6.3	7.5
5 x 0.5	5	0.5	16.1	2.3	3.9	6.2	8.7	10
6 x 0.75	6	0.75	22.0	3.9	6.5	10	15	17
8 x 1	8	1	39.2	9.2	15	25	35	42
10 x 1.25	10	1.25	61.2	18	30	48	68	81
10 x 1	10	1	64.5	19	31	49	70	84
12 x 1.5	12	1.5	88.1	31	52	83	117	140
12 x 1.25	12	1.25	92.1	32	53	85	120	144
14 x 1.5	14	1.5	125	51	84	135	190	228
16 x 1.5	16	1.5	167	76	127	204	287	344
18 x 1.5	18	1.5	216	110	184	294	413	496
20 x 1.5	20	1.5	272	153	255	408	574	688
22 x 1.5	22	1.5	333	205	341	546	768	921
24 x 2	24	2	384	261	435	696	979	1170
27 x 2	27	2	496	376	627	1000	1410	1690
30 x 2	30	2	621	520	866	1390	1950	2340
33 x 2	33	2	761	697	1160	1860	2610	3130
36 x 3	36	3	865	883	1470	2350	3310	3970
Nominally $\sigma_s = R_{eL}$ or $R_{p0.2}$ N/mm^2				240	400	640	900	1080
$\dfrac{k}{\kappa\left(1 + \dfrac{S_F}{F_{Fm}}\right)} \times \sigma_s$ N/mm^2				26.16	43.60	69.76	98.10	117.72

TABLE 9.4

TIGHTENING TORQUE (M_V) IN Nm FOR WAXED STAINLESS (ALSO ACID-RESISTANT) BOLTED JOINTS WITH THE USE OF TORQUE WRENCH OR SCREWDRIVING MACHINE/IMPACT WRENCH WITH TORQUE CONTROL. TORQUE DEVIATION MAXIMUM ± 5%. METRIC COARSE THREADS.

Thread	d	P	A_s	Property class in accordance with SS-ISO 3506-1							
				Austenitiska (A)			Ferritic (F) and martensitic (C)				
M	mm	mm	mm²	50	70	80	45	50	60	70	80
1.6	1.6	0.35	1.27	0.057	0.12	0.16	0.068		0.11		0.17
2	2	0.4	2.07	0.11	0.25	0.33	0.14		0.22		0.35
2.5	2.5	0.45	3.39	0.23	0.50	0.66	0.28		0.45		0.70
3	3	0.5	5.03	0.41	0.87	1.2	0.48		0.79		1.2
3.5	3.5	0.6	6.78	0.64	1.4	1.8	0.76		1.3		2.0
4	4	0.7	8.78	1.0	2.0	2.7	1.1		1.9		2.9
5	5	0.8	14.2	1.9	4.1	5.4	2.3		3.7		5.8
6	6	1.0	20.1	3.3	7.0	9.3	3.9		6.3		9.9
8	8	1.25	36.6	7.8	17	22	9.3		15		24
10	10	1.5	58	15	33	44	18		30		47
12	12	1.75	84.3	27	57	76	32		52		82
14	14	2	115	43	91	121	51		83		130
16	16	2	157	65	140	187	78		127		199
18	18	2.5	192	91	195	260	108		178		277
20	20	2.5	245	127	273	364	152		249		388
22	22	2.5	303	171	367	490	204		335		523
24	24	3	353	220	472	629	262		430		671
27	27	3	459	318	682	909	379		621		969
30	30	3.5	561	434	930	1240	517		848		1320
33	33	3.5	694	585	1250	1670	697		1140		1780
36	36	4	817	755	1620	2160	899		1470		2300
39	39	4	976	969	2080	2770	1150		1890		2950
Nominally $\sigma_s = R_{eL}$ or $R_{p0.2}$ N/mm²				210	450	600	250		410		640
$\dfrac{k}{\kappa\left(1 + \dfrac{S_F}{F_{Fm}}\right)} \times \sigma_s$ N/mm²				23.10	49.50	66.00	27.50		45.10		70.40

WHY DOES IT BREAK?

10

10.1 STATIC FRACTURES

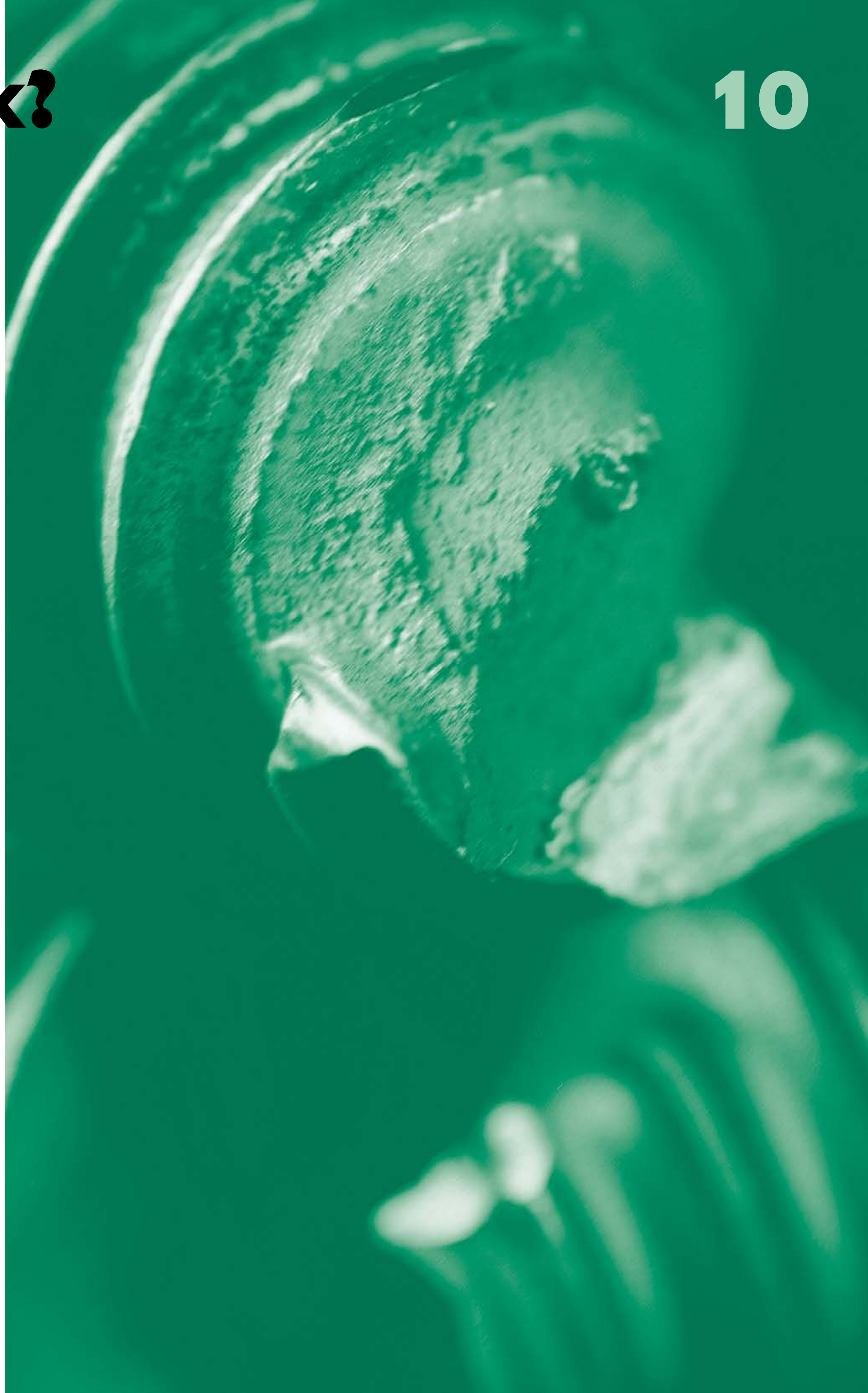
There can be several reasons for a bolted joint breaking. In static load, i.e., pure extension strain without any side thrust, the fracture can have the appearance of one of the three examples shown below.

Figure 10.1 A B C

A - The nut is too weak in relation to the bolt and the threads of the nut sheared off.

B – The nut is stronger than the bolt, but too short, and therefore it sheared the threads of the bolt.

C – The nut is stronger than, or equally as strong as the bolt, and also has sufficient height to make the bolt break. This is the type of fracture one would prefer, because the pieces come apart and the fracture can easily be noticed. In the other cases, the nut can remain on the joint and perhaps the fact that it is broken is not discovered.



10.2 FRACTURE WITH EXCESSIVE AXIAL LOAD

Here, you can see how the bolt has been elongated and contraction occurred before the bolt broke. The cross section also shows a typically viscous fracture appearance. Because the load was higher than the calculated load, the remedy is either to increase the number of bolts or increase their size, or alternatively choose a higher property class.

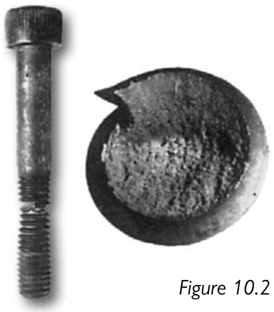


Figure 10.2

10.3 TORQUE FRACTURE WITH EXCESSIVE PRETENSION

If the joint has been subjected to excessive pretension during tightening, a fracture occurs with the appearance shown here. It is recognised through the weak spiral structure of the fractured surface and through the fact that the fracture always occurs in the threaded part of the bolt. The remedy can be better control of the tightening torque.

Figure 10.3

10.4 TORSIONAL FRACTURE

In a pure torsional fracture, the bolt has been twisted off without pretension. Such a fracture can occur if the bolt is too long, when the end of the bolt has become caught in a threaded blind hole. The surface of the fracture is shiny and has a spiral structure. The remedy is to recalculate the joint and with better dimensions.

Figure 10.4

10.5 FATIGUE FRACTURE

Fatigue fracture can occur in several different forms, and all have their characteristic appearance. What they all have in common is a more-or-less developed “mussel shell appearance”. The typical mussel structure of the fracture is shown in Figure 10.5. The fracture occurred through the fatigue tension having pulsed around a mean tension, for which reason, a linear structure developed.

Figure 10.5

10.5.1 PULSATING AXIAL LOAD WITH HIGH PRETENSION

If the bolt is subjected to varying tension loads without lateral stress, a fracture surface is obtained that looks like the one in this picture.

The fracture surface consists of two areas, one ring-shaped with grey fine-granular structure and a central but coarser structure where the area became so small that it was unable to withstand the load and sheared off. The solution is a design change, stiffer clamped material or to change to a bolt with a waist which has been turned down, a so-called bolt with extendable shank. In this way, a better relationship between the compressing and compressed spring constants of the parts is obtained, and also less pretension.

Figure 10.6

10.5.2 PULSATING AXIAL LOAD WITH LOW PRETENSION

The bolt in this picture broke due to a pulsating tensile strain, but under lower pretension. The appearance of the fracture surface differs from that of the previous bolt, but in both cases, the fracture occurred in the first supporting thread. The solution is the same as in the previous case. One can also increase the fatigue strength by rolling the thread after tempering, and also ensuring that the contact surfaces are as smooth as possible in order to avoid micro-settling in the joint.

Figure 10.7

10.5.3 PULSATING TENSION AND FLEXING STRESS

If a bolt is subjected to a pulsating tension load and also to a superimposed bending load, the fracture looks like this. At the side of the central, coarser fracture surface, two grey zones can be seen.

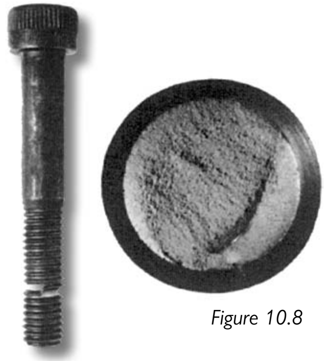


Figure 10.8

10.5.4 ONE-SIDED FLEXING FATIGUE

If the load was one-sided, pulsating repeated impact bending can occur, leading to the appearance of this fracture.

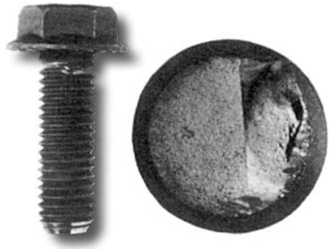


Figure 10.9

10.5.5 ALTERNATING BENDING STRESS

If the bending stress alternated from one side to the other, a fracture as shown in Figure 10.10 occurs. The final fracture occurs in the middle of the bolt, and the fatigue fracture can be fine-grained or mussel-shaped.

The remedy for flexing fatigue is redesign through introducing a load-bearing medium to absorb the bending stress before it gets into the bolted joint. Also, better surface evenness of the clamped parts is required, and better elasticity relationship between clamped and clamping parts of the joint.

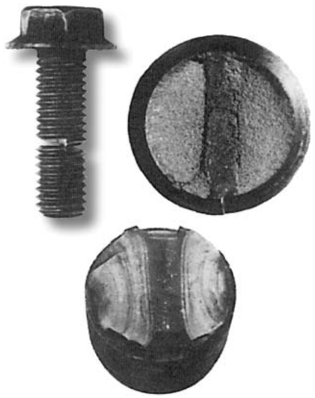


Figure 10.10

10.6 HYDROGEN EMBRITTLEMENT FRACTURE

If the bolt has been incorrectly tempered or has not been annealed, a so-called hydrogen embrittlement fracture can occur. The bolt is then too hard, and the fracture occurs immediately, without elongation, roughly in the same way as one breaks a glass rod.

Similar types of fracture can occur in hydrogen embrittlement when the correct dehydrogenation has not occurred or has been carried out too late after the electrolytic treatment. It is difficult to check whether hydrogen has forced its way into the edges of the nodules of the material. Normally, a pretension test is used, in which the bolt is left in a pretensioned condition for at least 48 hours under a stress which is close to the yield stress of the bolt.

If, after this period, the bolt is again subjected to load with a clamping torque without fracturing, it is assumed that the bolt is free from damaging hydrogen. Hydrogen embrittlement fractures are shown below.

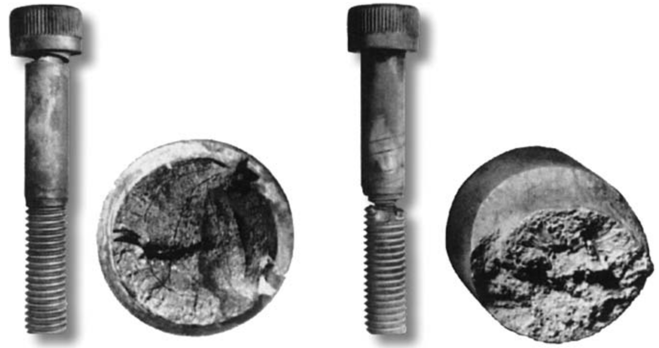


Figure 10.11

The remedy for embrittlement fracture in high-strength screws and bolts, apart from the right heat treatment, can be mechanical zinc coating or some other type of surface treatment, for example POLYSEAL, which excludes the presence of hydrogen.

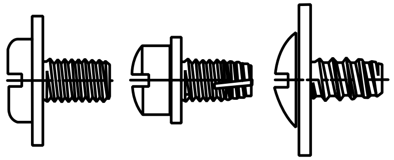
The images on the preceding pages are from SCHRAUBEN VADEMECUM by D Blume/K H Illner, published by Bauer & Schaurte Korcher GMBH.

11 WHAT IS AVAILABLE AND WHEN SHOULD IT BE USED?

11.1 IN-PLACE COST – THE STORY BEHIND A CONCEPT

It began back in the 1930s, when the United States suffered one of its worst economic shocks – the Wall Street Crash, which also affected the European economies.

Companies searched frantically for processes and products which could reduce manufacturing costs, not least in the assembly process. Subsequently, during the 1960s, screws, bolts and nuts with pre-mounted washers, so-called SEMS and KEPS were developed, and the concept of In-Place-Cost (IPC) was introduced.



Example of SEMS bolts

Figure 11.1

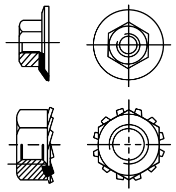


Figure 11.2 Example of KEPS nuts

In the late 1960s, BULTEN adopted the concept, using the Swedish abbreviation PPK *(På-Platsen-Kostnad)* and developed it to a full application and product-development philosophy. The main emphasis was on the assembly process in industry, and BULTEN developed or adopted many products which gave cost savings in one way or another.

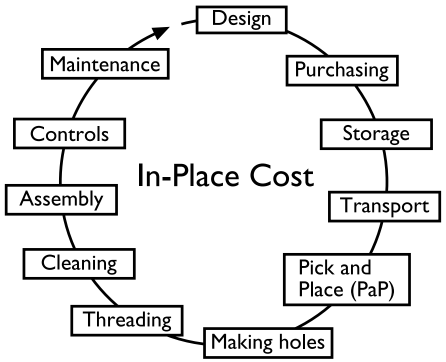


Figure 11.3

At first, it was difficult to get the fact across that money could be saved on such simple things as fasteners. Fasteners were often purchased by the youngest and least experienced people in a purchasing department. The easiest procedure for such a person was to play safe and save money for the company, by buying cheap. The in-place cost products were often expensive to buy, although cheap in assembly, but the buyer was not concerned with that. Only when there were problems on the shop floor was the importance of in-place cost understood, and slowly awareness increased about the advantages of these products. When analyses showed that only 15% of a product's assembly cost consisted of the purchase price of the fasteners, while the remaining 85% consisted mainly of peripheral costs, the value of the in-place cost concept was finally understood.

Nowadays, it is obvious that suppliers must be able to offer their customers application services. Quite often genuine, bona fide sub-suppliers establish their own applications centres and functional testing laboratories in order to be able to live up to the description of "First Class Supplier". BULTEN was a pioneer in this area, which can be seen from the product presentation below.

The in-place cost concept has spread to many other areas, products and processes, as can be seen from the accompanying booklet "In-Place Cost – a way of thinking". There, the reader can see associations with his own applications and savings.

11.2 THE ONE-WAY SYSTEM

The one-way system was originally developed to facilitate the assembly of powerline pylons, and make it more efficient. Because of strong winds, ice coating and for other reasons, powerlines are subject to vibration, which migrates to the powerline pylons, the bolted joints of which can shake loose. By making a punch mark on the projecting bolt thread after attaching the nut, the nut was prevented from coming off, but there was no other check of the tensile stress in the joint than by hitting the structure with a sledgehammer. If the bolts "rattled", the tensile stress had been lost.

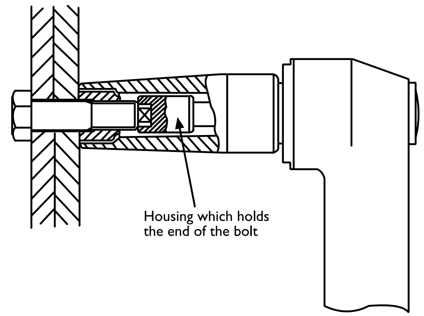


Figure 11.4

Therefore, a wholly metallic, thick locking nut of hot zinc-coat type was developed, but it was then discovered that it was then necessary to hold against the bolt head, as the nuts were stiff on the bolt threads due to friction locking. This was inconvenient and also quite dangerous when assembling at a height.

After the problem had been given a deal of thought, it was solved by attaching a key grip to the end of the bolt, to grip while the nut was screwed down. The system required the development of a special tool, and the entire system came to be called BUFO® Grip. Apart from powerline pylons, it also became popular in other areas where it was difficult to apply a holder to the bolt head, for example, in the automotive industry.

11.3 THE BULTEN'S BUFO® PIERCE NUT SYSTEM

The round pierce nut was first developed in Japan for use in their automotive industry. When the Swedish automotive industry began to show interest in the nut, BULTEN AUTOMOTIVE began to manufacture it in their Kalix factory. Below is a brief account of the product and its use. Additionally, there is a lengthy brochure in English. Apart from demand, the main reason for launching the nut was the fact that it fitted neatly into the in-place cost concept.

11.3.1 PROPERTIES

Compared with weld nuts, the pierce nut has the following advantages:

- saves electricity
- no smoke
- no welding splatter, which might mean cleaning the thread of the nut
- avoids intermediary steps between the pressing and welding stations
- reduces lead times
- avoids difficulties with coated sheet metal
- more secure orientation of the nut against sheet metal and a more exact right-angle, thus avoiding assembly out of square.

In comparison with square pierce nuts, there are also other advantages:

- easier feed and less risk of the nuts getting stuck in the feed shoe
- more even distribution of pressure between nut and sheet metal
- no stress concentrations due to sharp corners
- reduced risk of stress cracks in the sheet metal

If one combines unthreaded pierce nuts with tapping screws of the TAPTITE II or DUO-TAPTITE type, a further reduction of in-place cost is achieved, thanks to:

- reduced risk of cross threading
- not necessary to plug or mask the nut holes
- not necessary to clean threads after painting or other surface finishing
- a vibration-proof joint is obtained
- extra locking elements are not needed.

A further advantage of BULTEN's system for round pierce nuts is to be able to install with a consistent force using a gas spring. See Figure 11.10. The system guarantees correct torsional resistance, draw-through force and slackening force, irrespective of vibration in whatever sheet metal guage it is used with.

Figure 11.5

11.3.2 DIMENSIONS

Nuts are available in thread sizes M5 up to M10, dimensions are given in the table below.

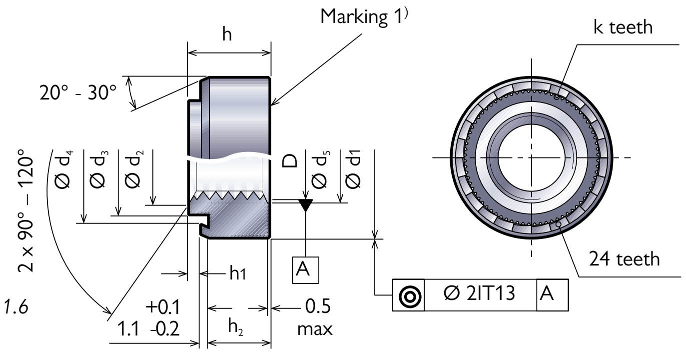


Figure 11.6

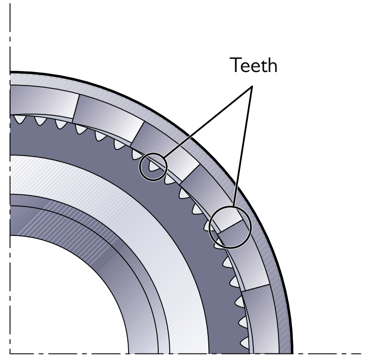


Figure 11.7

TABLE 11.1 SIZES AND TOLERANCES

Thread		M5				M6			
h	min	5.85				5.85			
	max	6.15				6.15			
d_1	min	14.4				14.4			
	max	14.6				14.6			
d_2		6				7			
d_3	min	8.3				8.3			
	max	8.6				8.6			
d_4	min	11.6				11.6			
	max	11.9				11.9			
d_5		6				7			
k, no.		36				36			
Plate gauge	h1	h2				h1		h2	
min	max	min	max	min	max	min	max	min	max
0.55	1.2	0.25	0.55	4.3	4.7	0.25	0.55	4.3	4.7
(1.2)	1.6	0.85	1.15	3.7	4.1	0.85	1.15	3.7	4.1
(1.6)	2.0	1.25	1.55	3.3	3.7	1.25	1.55	3.3	3.7

Thread		M8						M10			
h	min	–						9.85			
	max	–						10.15			
d_1	min	16.4						19.9			
	max	16.6						20.1			
d_2		9.5						11			
d_3	min	10.8						13.8			
	max	11.1						14.1			
d_4	min	13.9						16.9			
	max	14.2						17.2			
d_5		9						11			
k, no.		45						60			
Plate gauge		h		h1		h2		h1		h2	
min	max	min	max	min	max	min	max	min	max	min	max
0.65	1.2	6.85	7.15	0.35	0.65	5.2	5.6				
(1.2)	1.6	6.85	7.15	0.85	1.15	4.7	5.1				
(1.6)	2.0	6.85	7.15	1.25	1.55	4.3	4.7	1.25	1.55	7.3	7.7
(2)	3.0	7.85	8.15	1.85	2.15	4.7	5.1	1.85	2.15	6.7	7.1

Plate gauge	Marking
0.55– 1.2	▪
(1.2) – 1.6	▪ ▪
(1.6) – 2	▪ ▪ ▪
(2) – 3	▪ ▪ ▪ ▪

1) The marking system shows which plate gauge the nut is suitable for.

11.3.3 INSTALLATION

BULTEN's pierce nuts are fed into the tool head through a feed hose (a). When the press has returned to its original position and all previous functions of the current operation have been completed, the air actuator (b) automatically draws back so that the pillar (c) enables the punch (d) to move downwards and thus a new pierce nut is fed to the punch head. A sensor organ (e) is located in the punch head and checks that the nut has come into the right position.

The assembly process starts with the upper press tool moving vertically downwards. Upon contact with the sheet metal, the punch (f) punches the pierce nut (g) down into the sheet metal (h) and against the punching die (i). The pierce nut now punches a hole in the sheet metal, and the punched piece of scrap metal falls through the hole in the die. At the same time, the circular collar of the die is forced into the circular, lightened impression made by the pierce nut in the sheet metal in its contact plane, so that a permanent, secure joint is obtained.

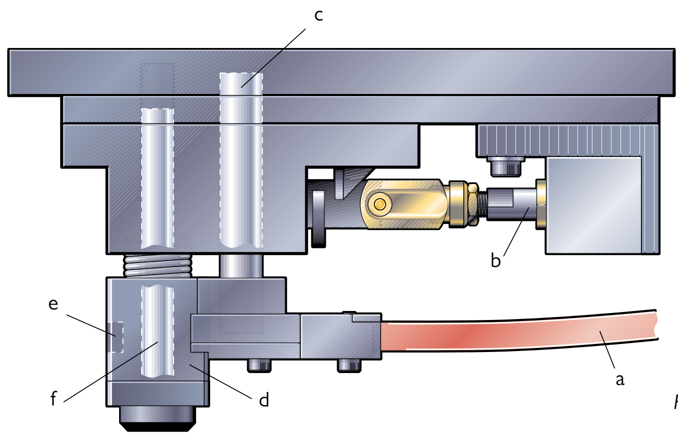


Figure 11.8

The assembly sequence of the nut is shown in the illustrations below.

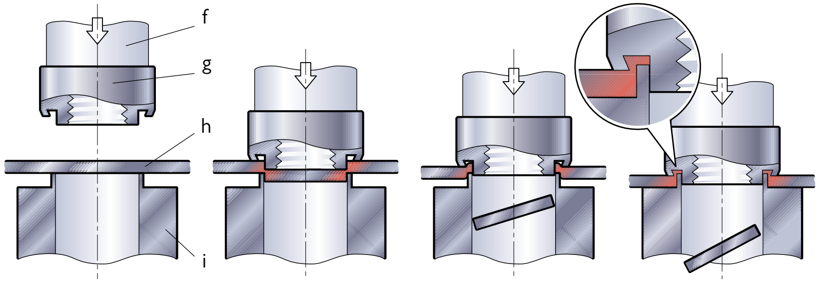
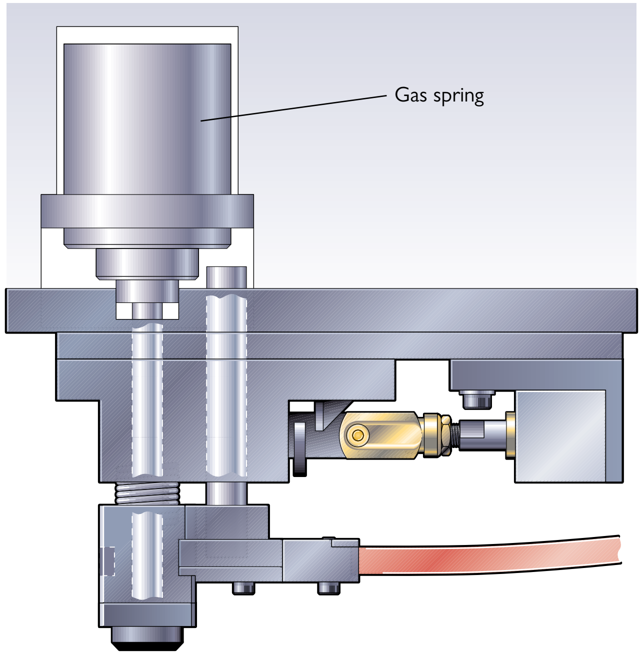


Figure 11.9



Tool with gas spring for constant press force.

Figure 11.10

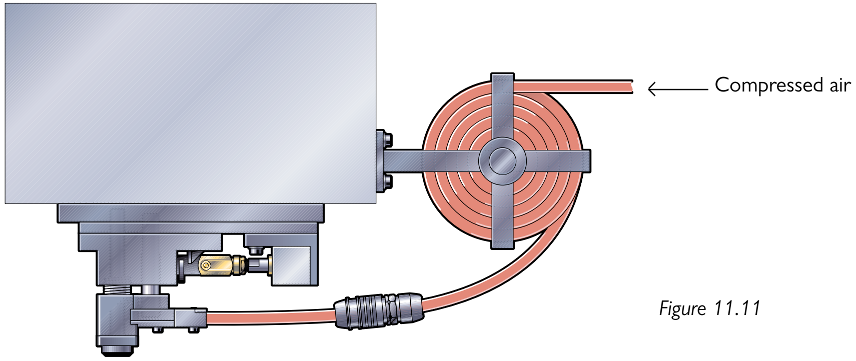


Figure 11.11

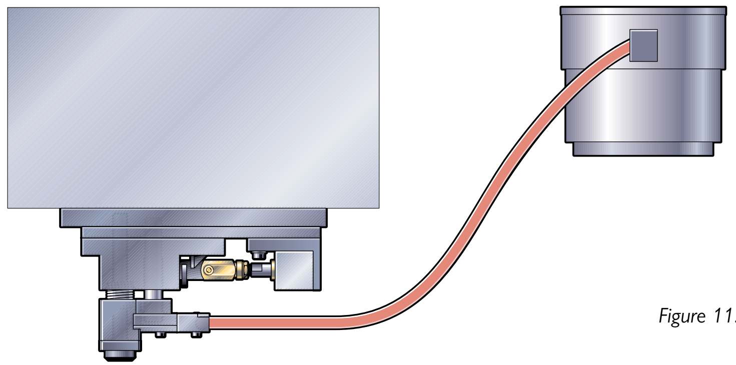


Figure 11.12

Figure 11.13

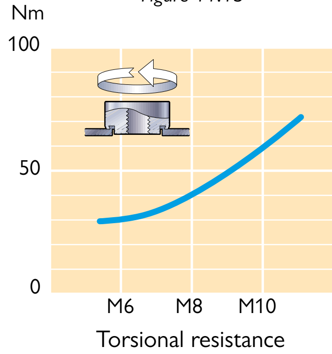


Figure 11.14

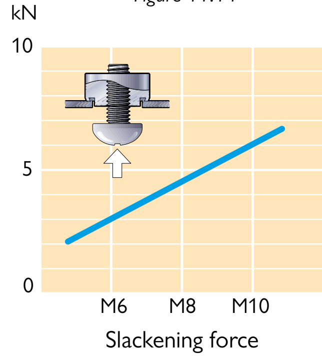
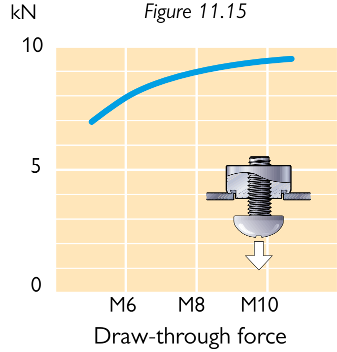


Figure 11.15



The nut feed can be run either by a compressed air feed through a hose, Figure 11.11 or a vibration feeder, Figure 11.12. Choice of feed system is determined by the space at the assembly station.

As regards strength, the nuts satisfy the requirements for property class 8 in accordance with SS-EN 20898-2. The nuts are suitable for use in sheet metal of gauges 0.55 to 3 mm with an tensile strength of R_m max 400 N/mm^2.

The pierce nuts can also be applied in joints of light sheet metal. The nuts are normally zinc electroplated and bright chromated, but other surface treatments can also be obtained. Typical values of torsional resistance, draw-through force and slackening force for nuts mounted in 1.2 mm sheet metal are shown in Figures 11.13 – 11.15.

11.4 TRILOBULAR TAPPING SCREWS

For tapping holes in steel of normal hardness, TAPTITE® tapping screws have shown superior in-place cost effect. This is even more marked in threading in light metal, and also in reinforced siluminum material.

A look back:
A well-known specialist in the fastener industry in the USA developed and patented a trilobular screw thread which cuts the internal threads in a hole plastically, without making burrs. The basic trilobular concept was so successful that it was followed by a great many other products, developed for various applications. There are now licensed manufacturers throughout the world, and it is estimated that each year 15 billion screws of this basic form are used, and that each screw gives a saving of between Euro 0.02 and Euro 0.05. In Europe, almost three billion are used each year.

The most popular and widely distributed types today are:

TAPTITE 2000®
DUO-TAPTITE®
CORFLEX®

Stainless and acid-resistant screws with TAPTITE 2000 thread or DUO-TAPTITE thread are also manufactured in large quantities.

11.4.1 TWO METHODS OF CUTTING THREADS IN MATERIAL AND MOUNTING SCREWS

The old method involved mounting the screw in a tapped hole. Because there was often a gap between the internal and external threads, the screw could easily be shaken loose if it was subjected to vibration.

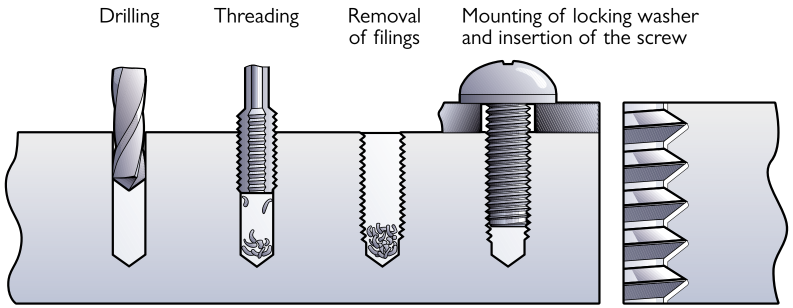


Figure 11.16

The new method involves allowing the screw to tap its own thread in the unthreaded hole.

The TAPTITE screw does this plastically and without creating metal residue. A stronger joint is obtained and it is considerably more secure against being shaken loose; also it does not need any extra locking elements.

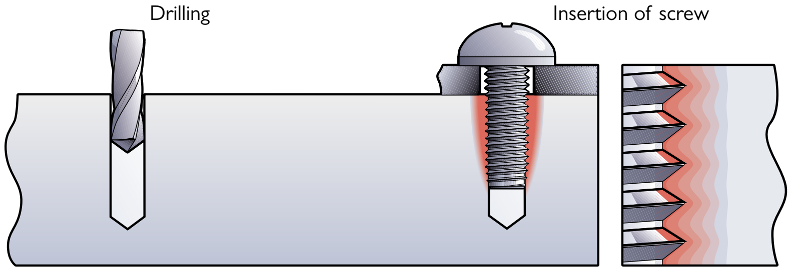
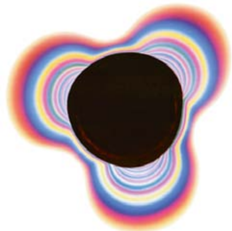


Figure 11.17



The three lobes of the screw thread press threads into the wall of the hole.

Figure 11.18

11.4.2 TYPES OF HEAD

All the common types of head are manufactured with internal and external grips in the diameter range of M3 – M12, and there are also coarser and finer variants.

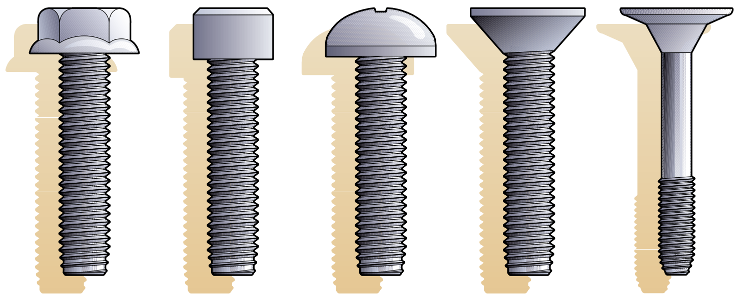


Figure 11.19

Flange screws with a hexagonal or TORX® grip work extremely well on trilobular screws when a high level of tensile stress is required, including against a soft base, for example, light metal. As regards internal grip, the TORX grip has shown itself to be especially good. The TORX grip makes it possible to manufacture screws of which the head is as strong as the thread. The grip of the screw and the tightening tool withstand such high torque that the screw can be twisted off. Thanks to TORX, screws with countersunk heads can now be given correct pretension as well.

11.4.3 **THREADS**

TAPTITE II screws tap strong, solid threads into drilled, stamped or pressed holes in sheet metal or cast items of ductile metal.

When a TAPTITE II screw is turned through a hole in a material, each lobe of the thread forms and cold deforms the material of the wall of the hole into a thread of unbroken sequence.

The TAPTITE II thread (Figure 11.20) is designed so that the torque is lower than with earlier TAPTITE screws. On account of elastic spring-back, the material partly fills up behind the lobes and creates a larger contact surface between the threads, which gives considerable resistance to slackening and high joint strength.

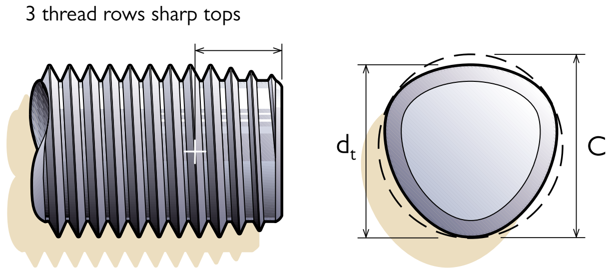


Figure 11.20

DUO-TAPTITE screws have an extra large trilobularity on the end of the screw, Figure 11.21, partly to facilitate the entry of the screw into the hole, and partly to give the thread optimal tapping ability.

At the strength-supporting part of the thread, the trilobularity is less in order to increase the strength of the joint. Two stabilising threads on the end of the screw ensure that the screw enters the hole easily and straight, and grips in with minimal axial force.

In extensive laboratory tests, the DUO-TAPTITE screws consistently out-performed other tapping screw types in every area with high function requirements.

The DUO-TAPTITE screw is especially appropriate for CORFLEX® hardened fasteners in power transmission joints with varying loads.

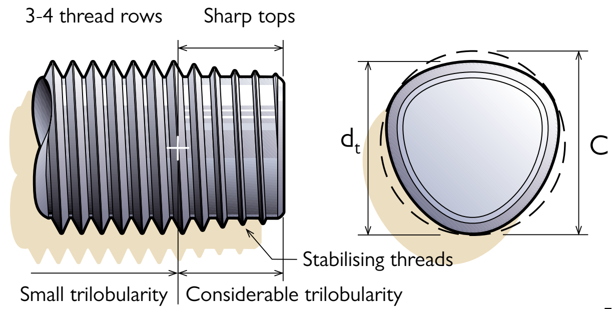


Figure 11.21

TABLE 11.2 THREAD DIAMETERS

Thread	TAPTITE II				DUO-TAPTITE			
	d_t		C		d_t		C	
	max	min	max	min	max	min	max	min
M2.5	2.48	2.39	2.57	2.48	2.52	2.44	2.57	2.48
M3	2.97	2.88	3.07	2.98	3.02	2.93	3.07	2.98
M4	3.94	3.84	4.08	3.98	4.01	3.91	4.08	3.98
M5	4.93	4.82	5.09	4.98	5.01	4.90	5.09	4.98
M6	5.90	5.77	6.10	5.97	6.00	5.87	6.10	5.97
M8	7.88	7.72	8.13	7.97	8.00	7.85	8.13	7.97
M10	9.85	9.67	10.15	9.97	10.00	9.82	10.15	9.97
M12	11.83	11.62	12.18	11.97	12.00	11.80	12.18	11.97

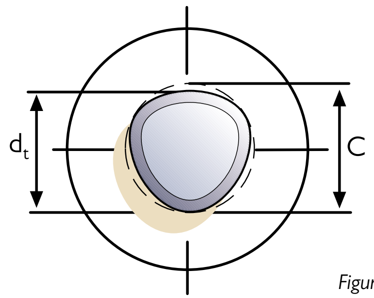


Figure 11.22

11.4.4 VARIOUS HARDENINGS

TAPTITE screws are available case-hardened and tempered.

Case hardening increases the carbon content in the surface area, to a depth of about 0.2 mm. The surface acquires a hardness of about 450 HV. The tensile strength of the screw is then at least 930 N/mm^2, for which reason the strength can be compared with class 8.8. Case hardening, however, means that the screw does not have the same elasticity as a tempered screw, and should therefore not be used in power joints or under dynamic stress.

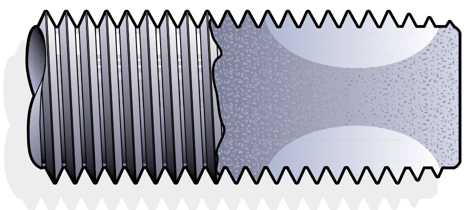
The case-hardened screws are most often used in thinner dimensions up to M8, and can tap threads in steel of a hardness up to 250 HB, and in aluminium, magnesium, zinc and copper alloys.

Some cast goods of iron and magnesium can be brittle. Contact us for guidance.

Screws with CORFLEX hardening can either be neutrally hardened, CORFLEX-N, or induction hardened, CORFLEX-I. The neutrally hardened screws are tempered to property class 10.9 and can form threads plastically in formable, non-iron metals, the hardness of which does not exceed 100 HB.

CORFLEX-I is tempered to property class 10.9 and has an induction-hardened tip. The hardness of the tip is at least 450 HV, i.e., as hard as case-hardened TAPTITE and with equally good tapping properties. Because the induction-hardened zone is limited to the tip of the screw, neither the ductility nor the elasticity of the screw is affected.

CORFLEX hardened screws can therefore advantageously be used in high strength joints exposed to dynamic loads and vibration.



The light colour threads in Figure 11.23 are the induction-hardened tip.

Figure 11.23

11.4.5 HOLE RECOMMENDATIONS

TABLE 11.3 DRILLED OR PUNCHED HOLES IN CARBON STEEL OF 110-130 HB

Thread	Thickness of material or length of engagement						
	0.5-1.5	1.5-2.5	2.5-4	4-6.5	6.5-10	10-15	15-
	Hole diameter D i mm *)						
M2.5	2.25	2.30	2.35	2.35			
M3	2.70	2.75	2.80	2.80	2.80		
M4	3.60	3.65	3.65	3.70	3.75		
M5		4.55	4.60	4.65	4.70		
M6		5.45	5.50	5.55	5.60	5.65	
M8			7.30	7.40	7.45	7.55	7.60
M10			9.20	9.25	9.30	9.40	9.45
M12				11.05	11.15	11.25	11.35

TABLE 11.4 DRILLED OR PUNCHED HOLES IN LIGHT METAL OF 80-120 HB

Thread	Thickness of material or length of engagement						
	0.5-1.5	1.5-2.5	2.5-4	4-6.5	6.5-10	10-15	15-
	Hole diameter D i mm *)						
M2.5	2.25	2.25	2.30	2.35			
M3	2.70	2.75	2.75	2.80	2.80		
M4	3.60	3.60	3.65	3.70	3.75		
M5		4.55	4.60	4.60	4.65		
M6		5.45	5.45	5.50	5.55	5.60	
M8			7.30	7.35	7.40	7.50	7.55
M10			9.20	9.20	9.25	9.35	9.40
M12				11.05	11.10	11.20	11.25

*) The diameters are recommended median values – not drill diameters. Recommended hole tolerances up to M6 are ±0.05 mm and for M8 and coarser, ±0.075 mm.

TABLE 11.5 DIE-CAST HOLES IN LIGHT METAL

Thread	Hole diameter* D2	Hole diameter* D3	d min	K min	L min
M2.5	2.38	2.24	4.2	8.3	7.5
M3	2.85	2.68	5	9.8	9
M4	3.80	3.58	7	12.8	12
M5	4.75	4.48	9	15.8	15
M6	5.70	5.37	10	18.8	18
M8	7.70	7.27	14	24.8	24
M10	9.60	9.06	17	30.8	30
M12	11.60	10.96	20	36.8	36

* Tolerance = +0/-0.1

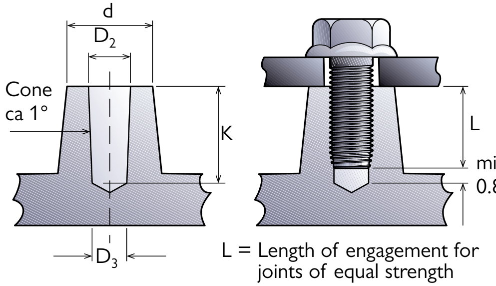


Figure 11.24

TABLE 11.6 DRILLED HOLES IN DIE-CAST LIGHT METAL

Thread	D*	S min	K min	L min
M2.5	2.25	1.3	8.3	7.5
M3	2.75	1.5	9.8	9
M4	3.70	2	12.8	12
M5	4.60	3	15.8	15
M6	5.55	3.2	18.8	18
M8	7.45	3.7	24.8	24
M10	9.30	4	30.8	30
M12	11.20	4.5	36.8	36

* Tolerance = +0.1/-0

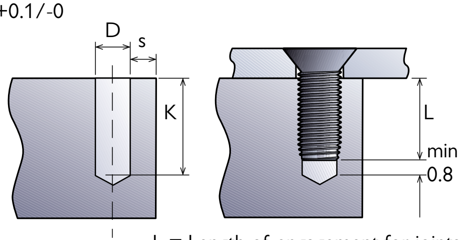


Figure 11.25

L = Length of engagement for joints of equal strength

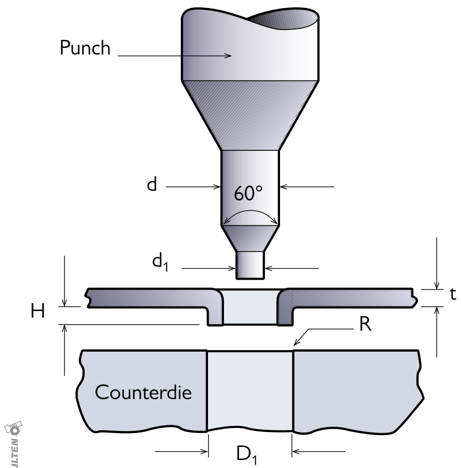
TABLE 11.7 COLLARED HOLES IN SHEET METAL

Thread	Hole diameter D in mm H11
M2.5	2.25
M3	2.70
M4	3.60
M5	4.55
M6	5.40
M8	7.30
M10	9.15
M12	11.00

When collaring holes, a punching tool can be used, as shown in Figure 11.26. The dimension rules apply for sheet metal of hardness 110-130 HB. Sheet metal harder than that should be annealed to avoid the formation of cracks.

TABLE 11.8 DIMENSIONS FOR HOLE COLLARS

Thread	Thickness of material t mm: Under 1	1-1.5	1.5-2	2-2.5	2.5-3	3-4
	Collar height H mm					
M2.5	1.0	1.0	1.1			
M3	1.2	1.2	1.3	1.3		
M4	1.3	1.4	1.5	1.5	1.6	
M5		1.6	1.8	2.0	2.3	2.5
M6		1.9	2.1	2.4	2.6	2.8
M8			2.6	3.0	3.2	3.5
M10				3.7	3.9	4.3
M12				3.9	4.3	4.7



D_1 = d+1t to d+1.2t

d = recommended hole diameter D

d_1 = 0.5d however at least = t

R = 0.1t

t = sheet thickness

Minimum sheet thickness =
= 0.2 x thread diameter.

Maximum sheet thickness =
= 0.75 x thread diameter

Figure 11.26

11.4.6 MOUNTING AND STRENGTH

Screws with TAPTITE II threads or DUO-TAPTITE threads are best mounted with a screw-driving machine of good torque control. 300 to 1500 rpm is appropriate.

The revolution speed must be adapted to thread diameter, form of head and mounting conditions in general. We have found pneumatic angle wrenches particularly appropriate for thread diameters above M6. Tightening torque for through screws in holes in material which is as strong as the screw is given in Tables 11.9 and 11.10. The tapping torque is considerably lower, perhaps 1/2 to 2/3 of the tightening torque. If the hole in the material is weaker than the screw, one should determine the torque of the tapping and tightening to fracture when testing mounting. Appropriate mounting torque is midway between these torques. When mounting in blind holes, there is simultaneous tapping and pretension, i.e., the two torques are added to one another. For correct execution, tests should be carried out from case to case. Percussive machines are inappropriate.

TABLE 11.9 STRENGTH AND TORQUE WITH CASE HARDENING

Thread	Failure load min kN	Torsional strength min Nm	Tightening torque** Nm	Tapping torque Nm
M2.5	3.15	1.2	0.8	≈0.5
M3	4.68	2.1	1.4	≈0.8
M4	8.17	4.9	3.4	≈2
M5	13.2	10	6.6	≈4
M6	18.7	17	11	≈7
M8	34.0	42	28	≈17
M10	53.9	85	55	≈30
M12	78.4	150	94	≈50

TABLE 11.10
STRENGTH AND TORQUE WITH CORFLEX HARDENING

Thread	Failure load min kN	Torsional strength* min Nm	Tightening strength** Nm	Tapping torque Nm
M5	14.2	9.3	7.3	=4
M6	20.1	16	13	≈7
M8	36.6	40	30	≈17
M10	58.0	81	58	≈30
M12	84.3	142	100	≈50

* In accordance with ISO EN 20898-7 property class 10.9.
** In through holes, as strong as the screw after the thread has been tapped.

11.4.7 QUALITY

Figure 11.27

Trilobular screws with TAPTITE II threads or DUO-TAPTITE threads have advantages for the user. The system satisfies total quality control requirements. The TAPTITE II thread was developed to satisfy requirements which accompany static process control (SPS). Other products in the system have also been adapted to SPS. Uniformity of execution and function gives the user higher product security and reduced in-place cost.

11.4.8 AREAS OF USE

Trilobular, capped screws are found everywhere that parts are joined together with screws, and where it is possible to screw into the material. Case-hardened TAPTITE II screws are mainly used in M8 diameter and below. Above M6 diameter, it is increasingly common to use CORFLEX hardened screws. In the range of M10 and M12, CORFLEX-I DUO-TAPTITE screws are much used in structures where there are stringent strength and safety requirements.

White goods in your home are held together with trilobular tapped screws. TAPTITE II screws are under the microwave oven housing. Cable clips on the electricity cables and the short-circuiting strap of the grill element can be mounted with screws, the undersides of which have a special shape for extra secure contact.

All kinds of electrical products work thanks to TAPTITE II screws. In switchgear for apparatus and machinery, for example, the screws are in switches and other actuators. An entire switchgear cabinet with bars for various installations can be held together and earthed with TAPTITE II screws. The parts are in contact with one another via the tapped thread in the material, and a special form under the screw head which breaks through paint or an oxidised layer. Also, the equipment can be protectively earthed at one point with the yellow-green cable. There are also screws with the same function in our modern AXE telephone exchanges.

The anchoring points on which car seat belts are secured are of extreme importance. Careless assembly can lead to damaged threads in the material and loss of strength. Often, the threads in material are hidden, and it is not possible to check whether there is welding spatter in the nut or whether the fastener has been correctly mounted. Therefore some companies insist that tapping screws shall be used in unthreaded holes in the material. CORFLEX-1 screws with DUO-TAPTITE threads and entry dogs have shown themselves extremely suitable. The risk of incorrect mounting is avoided, and the joints are given the correct strength.

11.4.9 PROPERTIES

- Lower in-place cost is the most important aspect.
- Case-hardened or tempered up to property class 10.9 for advanced mounting.
- Tapping of holes in material is avoided, also avoided are fixtures, thread tapping, thread control and gauges, and the range of tools and measuring and test devices required is reduced.
- Tapping without the formation of chips avoids the necessity of cleaning blind holes and reduces the risk of short circuits in electrical components.
- Friction between internal and external threads secures the joint without the need to pick and mount washers or other so-called locking elements. Nor do the screws need to be prepared with plastic plugs or similar. The number of variants and thus the cost of stock can thus be reduced.
- Cleaning paint from threads is also avoided. Either the screw can tap the thread in the hole or it can remove the paint from a hole which has already been tapped.
- Mounting is faster and easier with low tapping torque, including for thread diameters up to M16.
- Lower assembly costs, less operator fatigue
- The strength of joints is high thanks to the tapped, internal thread. This can mean fewer screws or smaller screw sizes.

11.4.10 EXTRUDE-TITE®

This is another example of a screw which has been developed to solve specific assembly problems. It is specially designed for thin sheet metal joints. The trilobular thread goes right out to the tip of the screw (1), while the load-bearing part (2) is of reduced ovality for better strength. In thin sheet metal, the screw collars the hole on insertion so that the length of engagement can increase up to double compared with the thickness of the original material. Appropriate hole diameters are about 75% of those in Table 11.7.

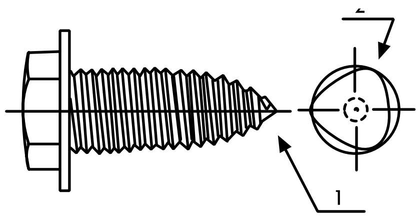


Figure 11.28

Another use is when several sheet metal parts are to be assembled with holes that perhaps do not centre exactly over one another. The screws are also appropriate when they are to go through several items of material, of which some can be soft or tough.

11.5 PLASTITE® FOR TAPPING IN PLASTIC

The usual method is to screw a so-called sheet metal screw with a thread profile of 60° into the hole in the plastic, but the plastic can be brittle and the circular screw thread can cause powerful compressive stress in the plastic, which can lead to cracks forming. Therefore, PLASTITE was developed within the TAPTITE family, a thread especially adapted for screwing into plastic with a minimal risk of cracks forming.

11.5.1 THREADS

PLASTITE is available with two separate thread profile angles, 45° and 60°.

PLASTITE 60 is the original type, while PLASTITE 45 has been developed for harder structural plastics.

TABLE 11.11 THREAD DIAMETERS OF PLASTITE 45

Thread mm	C min	C max	d_t min	d_t max
2.5 x 1.4	2.41	2.53	2.37	2.49
3 x 1.5	2.92	3.04	2.87	2.99
3.5 x 1.65	3.42	3.54	3.34	3.46
4 x 1.75	3.89	4.04	3.79	3.94
5 x 2.2	4.89	5.04	4.79	4.94
6 x 2.5	5.89	6.04	5.78	5.93
8 x 3	7.86	8.04	7.71	7.89

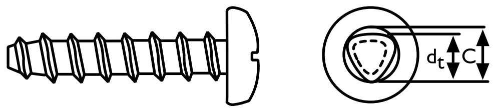


Figure 11.29

TABLE 11.12 THREAD DIAMETERS OF PLASTITE 60

Dimension No. – Threads per inch	C min	C max	d_t min	d_t max
2 – 28	2.18	2.34	2.11	2.26
4 – 20	3.07	3.23	2.97	3.12
6 – 19	3.58	3.73	3.48	3.63
8 – 16	4.55	4.70	4.39	4.55
10 – 14	5.23	5.38	5.13	5.28
12 – 11	5.82	5.97	5.69	5.84
14 – 10	6.86	7.01	6.65	6.81

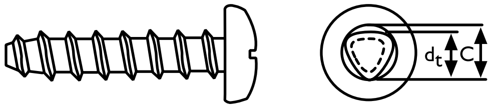


Figure 11.30

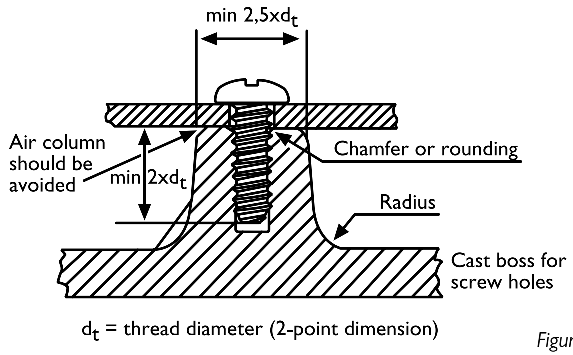


Figure 11.31

11.5.2 HARDENING

PLASTITE screws are made of carbon steel and case-hardened in the same way and to the same hardness as case-hardened TAPTITE screws.

11.5.3 HOLE RECOMMENDATIONS

Guideline values for hole diameters for PLASTITE 45

TABLE 11.13

Thread mm	Guideline for hole diameters[1)]
2.5 x 1.4	1.80–1.90
3 x 1.5	2.25–2.40
3.5 x 1.65	2.70–2.85
4 x 1.75	3.10–3.30
5 x 2.2	3.70–3.95
6 x 2.5	4.60–4.85
8 x 3	6.40–6.70

Guideline values for hole diameters for PLASTITE 60

TABLE 11.14

Dimension No. – Threads per inch	Guideline for hole diameters[1)]
2 – 28	1.90–2.00
4 – 20	2.50–2.70
6 – 19	3.00–3.20
8 – 16	3.80–4.00
10 – 14	4.40–4.70
12 – 11	4.80–5.10
14 – 10	5.60–6.00

1) The lower limits for soft plastic (AMIDE, ETHENE, PROPENE etc.).
The upper limits for hard or semi-hard thermo plastic (ACETAL, ACRYL, CARBONATE, STYRENE, etc. and hard plastics such AMINO, ESTER, FENO, URETHANE).

11.5.4 MOUNTING

PLASTITE is best mounted with a low-speed screwdriver of not more than 1000 rpm. At a faster speed, there is a risk that frictional heat can affect the strength of the joint.

11.5.5 QUALITY

PLASTITE screws are produced with the same high, uniform quality of all other BUFAB products, and satisfy all requirements which can be demanded for the function of tapping screws.

11.5.6 ADVANTAGES

High thread pitch gives a large shear area in the plastic. Considerable difference between tapping and tightening torques, and rapid mounting.

Large thread profile height gives considerable thread cover with minimal effect of hole variations.

Trilobular form gives low tapping torque and good locking thanks to the elasticity of the material.

11.5.7 RESULTS FROM TESTS WITH PLASTITE SCREWS

The diagram shows a comparison of joint strengths between PLASTITE 60 screws and ordinary "sheet metal screws".

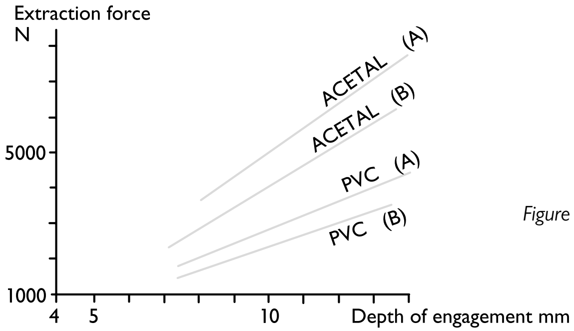


Figure 11.32

Ultimate tensile strength, (A) PLASTITE 60, thread 8-16 (B) "Sheet metal screw" ST 4.2

The two diagrams below show the results of functional tests in ABS plastic with PLASTITE 45 of 4 mm thread diameter with a thread length of engagement of 10 mm.

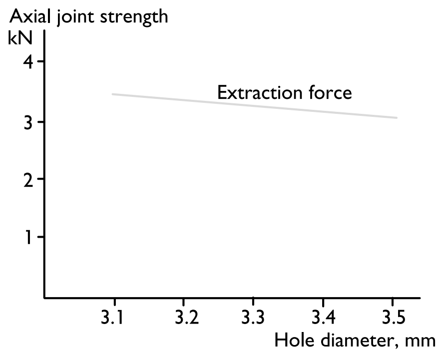


Figure 11.33

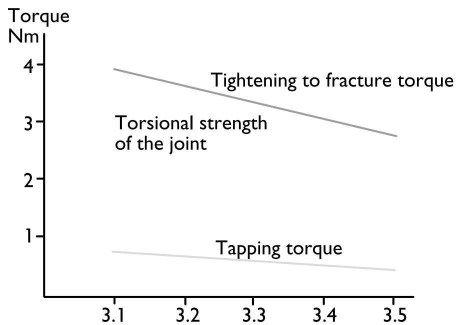


Figure 11.34

11.6 REMFORM® FOR TAPPING IN PLASTIC

Development has continued as regards screwing in plastic, and two new thread forms have been produced which are increasingly replacing the older forms. The new types are called REMFORM and REMFORM F. The main characteristic of these screws is the asymmetrical thread profile, which distributes stress around the hole in the material better and enables the carrying flank to better resist the extraction force, irrespective of whether it is caused by external tensile traction or by the tightening torque. They are manufactured in carbon steel and in stainless, and acid-resistant steel.

11.6.1 THREADS

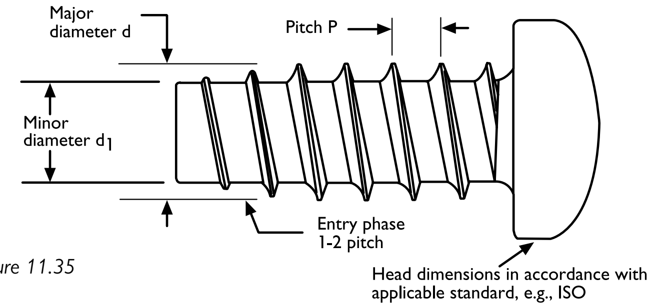


Figure 11.35

TABLE 11.15 THREAD DIMENSIONS

Thread d	Pitch P	Major diameter d max	Major diameter d min	Minor diameter d_1 min
RF2	1.00	2.10	2.00	1.17
RF2.5	1.15	2.60	2.50	1.47
RF3	1.35	3.10	3.00	1.90
RF3.5	1.55	3.60	3.50	2.22
RF4	1.75	4.10	4.00	2.55
RF4.5	2.00	4.65	4.50	2.87
RF5	2.25	5.15	5.00	3.19
RF6	2.65	6.15	6.00	3.84
RF8	3.00	8.15	8.00	5.65

11.6.2 STRENGTH

REMFORM carbon steel screws are tempered and have a high level of strength with material properties comparable to property class 10.9. Thanks to high torsional strength compared with other coarse pitch screws, and superior thread tapping ability, REMFORM can be used in long depths of engagement.

These long depths of engagement make REMFORM superior competitive screws, because greater strength can easily be obtained from the threaded hole due to greater length of engagement instead of through greater screw diameter (which gives rather limited functional advantages and at high cost). The greatest length of engagement at which a screw should be inserted is the depth at which the threads in the material have gained such a level of strength that the screw would rather break than that the hole threads would strip.

With weaker screws, longer depths of engagement cannot be utilised, because the tapping torque for greater thread engagement approaches the torque breaking strength of the screw. Also, when the length of engagement is not sufficient for the screw to break, the difference between REMFORM's high thread shearing torque and low tapping torque is a broad range within which the tightening torque can be selected.

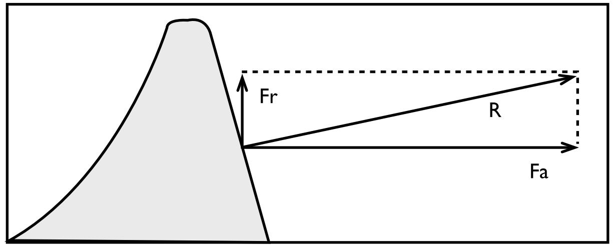


Figure 11.36

When a screw is tightened, the supporting thread flank of the thread in the material is prevented from being pulled towards the screw head. Almost all force arises between the supporting thread flank of the screw and the thread flank of the counter-thread. The entry thread is of the greatest importance for the formation of the counter-thread. The pointed angle which becomes a radial form in the flank of the clearance thread effectively promotes the redistribution of the material and its flow.

The coved form of the flank of the REMFORM screw's clearance thread reduces the tangential stresses by reducing radial stress in tapping and in connection with tightening the screw. The cross angle of the supporting thread flank transmits the greater part of the resulting force along the axis of the screw in connection with the tightening of the screw, and thus reduces the radial forces which otherwise would crack the material in a boss.

11.6.3 RECOMMENDED HOLES

For plastic material in the left-hand column, multiply the minimum outer diameter of the thread by the factor in the right-hand column in order to obtain a rough hole diameter d_3. The hole diameters obtained in this calculation are only to be considered as recommendations. They are based on theoretical calculations for depths of engagement equivalent to two thread diameters.

TABLE 11.16

Material	Factor for hole diameter
PA-6 30 GV	0.85
PA-6	0.83
PA-6.6	0.80
PP	0.80
PPO	0.85
PS	0.80
PE	0.80
ABC	0.78
PC	0.85

A cast column (tower) for screw holes can be dimensioned with proportions in accordance with the Figure, where d is the outer diameter of the thread. The hole diameter d_3 is d x factor in accordance with the table. For example: Hole diameter for thread RF 4 of plastic type PA-6 will be $d_3 = 4.0 \times 0.83 = 3.3$ mm.

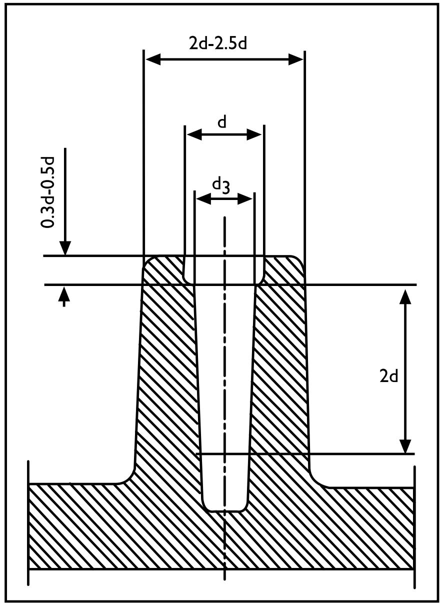


Figure 11.37

11.6.4 MOUNTING

A broad margin between tapping and fracture torques makes mounting easier by allowing mounting torques with comfortable margins of torque for tapping and tightening to fracture. This wide margin of torque improves safety as all screws will be fully tightened and the uncertainty of the risk of joints tightened to fracture is eliminated.

The unique thread form with sharp tops gives a low tapping torque because the material flows and is effectively redistributed.

In mounting, where other screws break because of torque when the torque fracture point is approached, RENFORM screws offer greater security thanks to their higher torque fracture strength. Where other screws cause fractures by stripping the threads in the material, the cross fall of the RENFORM thread's carrying flank ensures that forces are linked out axially instead of radially, which increases resistance to thread shearing.

11.6.5 QUALITY

The REMFORM thread has been developed into a product of the highest quality with the best function for secure joints. Consideration has been given to current manufacturing methods. This gives well-filled threads and correct form along the entire length of the screw.

11.6.6 REMFORM OFFERS

- High torsional strength
- Low tapping torque
- High tightening to fracture torque
- Broad range for mounting torque
- Minimal risk of cracks forming
- Deep depths of engagement where required
- High joint strength
- Low in-place cost

11.6.7 RESULTS OF TESTS WITH REMFORM SCREWS

The figures given here are typical values from laboratory tests. Function testing is essential to establish appropriate hole diameter, length of engagement, tightening torque and other factors.

Tapping torque and tightening to fracture torque

Screw diameter 4 mm
Length of engagement 8 mm

Screw diameter 6 mm
Length of engagement 12 mm

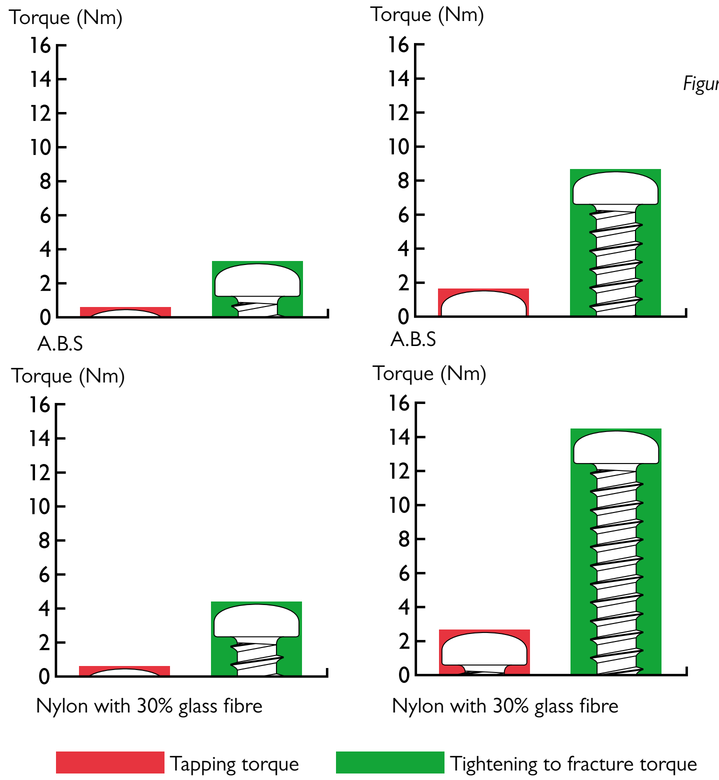


Figure 11.38

Extraction force

Screw diameter 4 mm
Length of engagement 8 mm

Screw diameter 6 mm
Length of engagement 12 mm

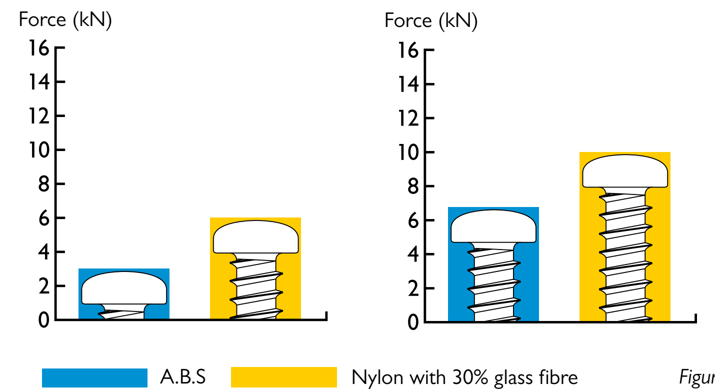


Figure 11.39

11.6.8 AREAS OF USE

REMFORM screws are ideal for joining thermoplastic resin, structural plaster, wood and other malleable material such as die-castings. Use REMFORM screws for demanding assembly, such as narrow pillars with long depths of engagement and high forces.

Examples of products suitable for REMFORM:

- Motor vehicle parts
- Vacuum cleaners
- Washing machines
- Hair driers
- Sound and image products
- Computers
- Electrical hand tools
- Telephone and communication equipment
- Furniture
- Light metal products

11.7 REMFORM® F FOR SCREWING INTO METAL

In order to take advantage of the good qualities of the REMFORM profile in brittle light metal, for example, magnesium, a variant with a finer thread pitch was developed, tempered to achieve material properties comparable to property class 10.9. It has the designation of REMFORM F and allows depths of engagement equivalent to 2-3xd in light metal, giving a joint equally as strong as the screw. The gripping forces are roughly the same as those of a 10.9 joint. Contact us for advice regarding the choice of hole, etc.

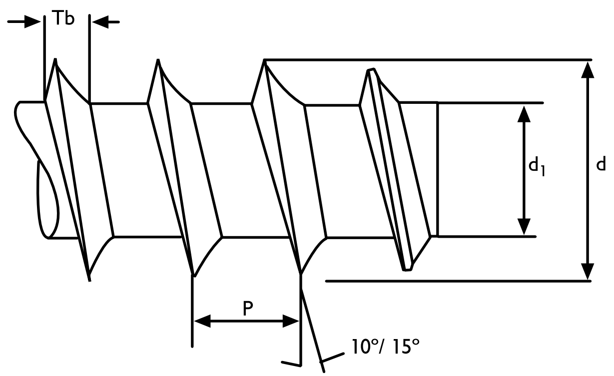


Figure 11.40

TABLE 11.17 THREAD DIMENSIONS OF REMFORM F

Thread d	Pitch P	Major diameter d max	Major diameter d min	Minor diameter d_1 min	Thread width T_b nom
2	0.60	2.10	2.00	1.33	0.476
2.5	0.70	2.60	2.50	1.66	0.571
3	0.80	3.10	3.00	2.00	0.665
3.5	0.95	3.60	3.50	2.33	0.758
4	1.05	4.10	4.00	2.63	0.776
5	1.25	5.15	5.00	3.32	0.877
6	1.40	6.15	6.00	4.06	1.049
8	1.75	8.15	8.00	5.54	1.301

11.8 SECURING A BOLTED JOINT

As long as bolted joints have existed, there has always been anxiety about the risk of them coming loose with the loss of tensile stress. There have been countless unsuccessful attempts to develop methods of preventing joints being shaken loose. We will not go into these here, but will just mention some methods that we at BUFAB have found do work, and we start with a nut.

11.8.1 FS LOCK NUT

The FS lock nut is a purely metallic lock nut, which differs from other lock nuts available on the market today.

FS lock nuts can be delivered with all types of surface treatment and in property classes 8, 10 and 12.

11.8.1.1 *How the FS Lock Nut Works*

The secret of the locking properties of the FS nut is the sprung, threaded, steel lock washer which is built into the nut body. Locking takes place by the lock washer springing when it is screwed in, and clamping around the screw thread. Locking occurs both axially and radially. The FS lock nut can be mounted and dismounted several times without the locking ability being noticeably impaired. This is due to the characteristics of the spring steel of the washer.

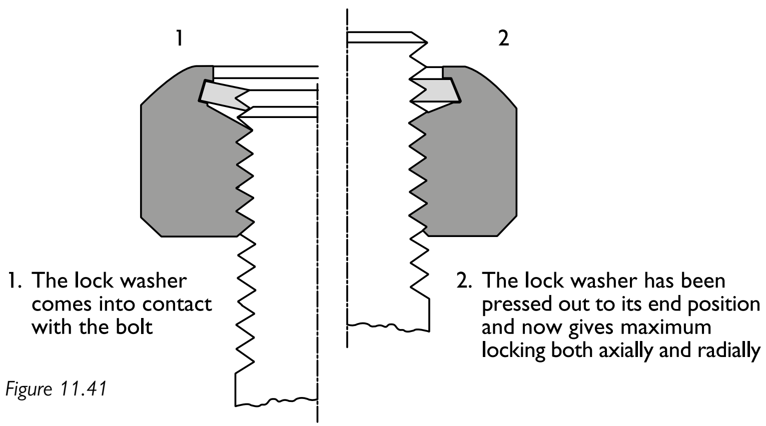


Figure 11.41

11.8.1.2 *Mechanical properties*

FS lock nuts have values which exceed the industry norms. Figure 11.42 shows the friction torque when slackening FS lock nut M10 of property class 10.

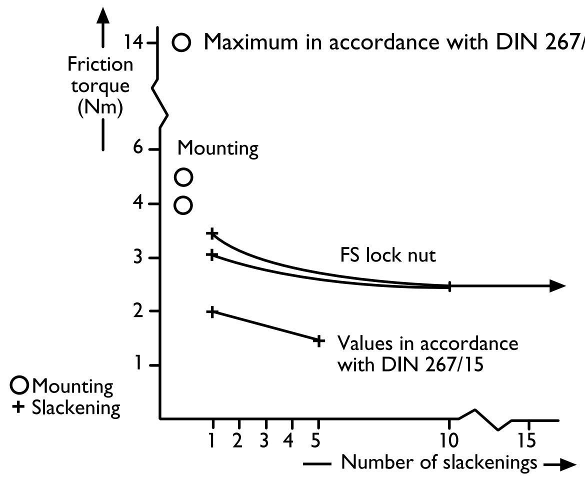


Figure 11.42

11.8.1.3 *Important advantages*

- Double-action lock washer which locks both axially and radially.
- Can be used several times without the locking ability (= friction torque) being noticeably impaired.
- Does not damage the screw thread.
- Friction torque as required for a particular structure.

11.8.1.4 *Special range*

FS lock nuts are available in stainless material (A, F or C), see Chapter 8, giving good corrosion resistance. The frictional torque of FS lock nuts can be set to tight tolerances during manufacture– high or low, as required. FS lock nuts can also be delivered with inch threads, fine threads or special threads. They are available as hexagonal, flanged and collar nuts. Other dimensions, material or types are available upon request or in accordance with drawing.

Examples of areas of use are: locomotives and railway cars, the automotive industry, boats, the machinery and engineering industry, the processing and civil engineering industry and the assembly of floating bridges and in applications with high working temperatures.

11.8.2 **BUFO® LOC**

Another example of a locking product is BUFO LOC. This is based on Omni-TECHNIK's well known micro encapsulation process and was developed in order to avoid a sticky, expensive problem.

11.9.2.1 *Function*

The concept of encapsulating locking liquid in small micro bubbles which are crushed when the bolt or screw is screwed into the nut or the hole, was realised in BUFO LOC. The bubbles which encapsulate the locking liquid are held intact by a bonding agent which is applied to the fastener in advance and stored. The locking is activated when the fastener is mounted, the bubbles are crushed and the liquid begins to harden.

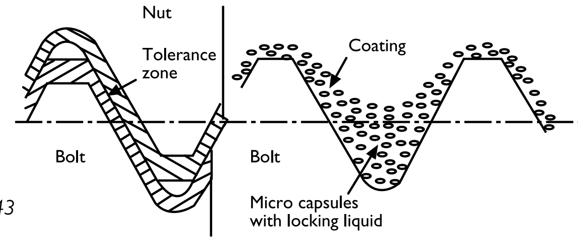


Figure 11.43

11.8.2.2 *Mechanical properties*

Thanks to the "dry" locking fluid, it has been possible to substantially reduce in-place costs. The locking ability is fully equal to liquid locking agents. Products treated with BUFO LOC are both locked and sealed. The full effect is achieved after about a 12-hour hardening time at room temperature. There are two types, BUFO LOC 30 with a yellow coating which gives sealing and moderate locking, and BUFO LOC 80 with a red coating which gives sealing and hard locking. BUFO LOC should be seen as a product that is used once, and should be replaced with new fasteners after disassembly. Treated fasteners can be stored for at least three years. The locking torque for dimension M10 is BUFO LOC 30 = 14 to 18 Nm and for BUFO LOC 80 = 25 to 30 Nm. The temperature ranges are BUFO LOC 30 = -50° to +120°C and BUFO LOC 80 = -50° to +170°C.

11.8.2.3 *Advantages of* **BUFO LOC**

- Shorter mounting time.
- Lower handling and storage costs.
- Wide temperature working range.
- Good sealing and resistance against oils, solvents, acids and alkalis.
- Good locking properties, i.e., excellent vibration resistance.
- Not poisonous and not allergenic (applies to coated fastener, ready to use).

11.8.3 **POWERLOK®**

The good locking properties of the TAPTITE thread, and its protection against being shaken loose and losing tensile stress were also needed for use in tapped holes in materials which cannot be formed plastically. Therefore, a thread was developed with a trilobular cross section, and a special thread top of 30° angle which, in mounting, is pushed into the root of the tapped thread giving a locking effect. This screw is called POWERLOK, and it eliminates all separate locking elements. It has also been used as a tapping screw in light metal with extremely good locking effect. The first rows of thread are under-dimensioned to facilitate entry.

11.8.3.1 *Function*

A stress-optical study of a POWERLOK joint seen from the end of the screw gives this colourful, three-leaf clover image. The trilobular form of the screw which strengthens the locking effect can be clearly seen here. The stress concentrations prevent the screw from loosening.

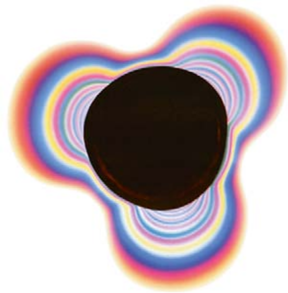
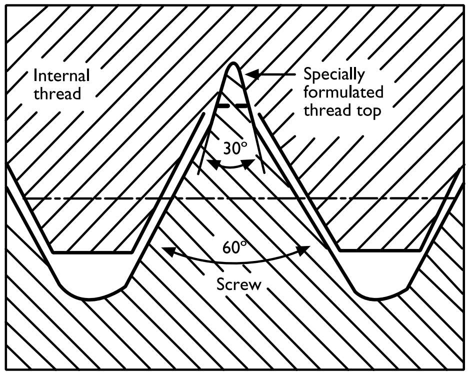


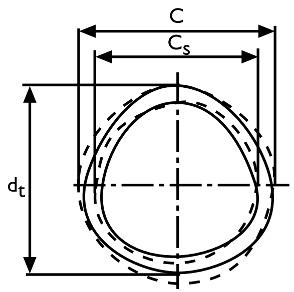
Figure 11.44

11.8.3.2 *Threads*

Threads in accordance with Table 11.18 can be combined with most forms of head.



Cross section through thread top. *Figure 11.45*



Thread seen from the screw end.

Figure 11.46

TABLE 11.18 THREAD SIZES

Thread	C		d_t		Tip C_s
	min	max	min	max	max
M4	4.10	4.22	3.96	4.08	4.00
M5	5.13	5.26	4.97	5.10	5.00
M6	6.15	6.30	5.95	6.10	6.00
M8	8.20	8.35	7.95	8.10	8.00
M10	10.25	10.40	9.95	10.10	10.00
M12	12.30	12.45	11.95	12.10	12.00

11.8.3.3 *Hardening*

The screws are normally tempered to property class 10.9 with hardness HB 304-361.

11.8.3.4 *Mounting*

POWERLOK is best mounted with a low-speed screwdriver, i.e., not more than 1000 rpm. POWERLOK is very suitable for automatic assembly, where both hand-held screwdrivers and stationary units can be used.

Recommended tightening torque is given in the table below, and applies for the surface treatments phosphating, POLYSEAL or alternatively zinc electroplating + wax.

TABLE 11.19 TIGHTENING TORQUE

Thread	Tightening torque in Nm 1)
M4	3.5
M5	7.0
M6	12.0
M8	29.0
M10	57.0
M12	99.0

1) Applies for alternative surface treatments of phosphating, POLYSEAL and zinc electroplating + anti-friction treatment.

11.8.3.5 *Areas of use*

POWERLOK can be used everywhere that other types of lock screw or securing elements are used in mounting in holes in material.

Thanks to its trilobular form, POWERLOK can also be used a as tapping screw in aluminium and other soft material. Appropriate hole diameters should be obtained by testing in each individual case. The locking ability of the screw is then even better than with pre-tapped holes. Because this screw also locks non-pre-tensioned, it is appropriate as an adjusting and clamping screw.

11.8.4 LOCTITE DRI-LOK PLASTIC®, THE SCREW WITH A PLASTIC COATING

This product is also one of the “dry” thread-locking type. Plastic is applied to the external thread and can be adapted to the customer’s requirements as regards locking effect, temperature dependence and chemical influence.

11.8.4.1 *Function*

The pre-coated surface increases the pressure against the thread flanks of the hole in the material. The result is that the flanks on the uncoated side of the threads are also forced together. The friction which is generated prevents the screw from coming loose.

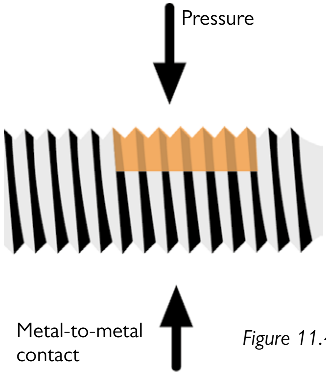


Figure 11.48

11.8.4.2 *Mechanical properties*

- Controllable strength, direct after application.
- Can be used at least five times.
- Makes possible adjustment of the assembled components.
- Satisfies requirements in accordance with German standard DIN 267, Teil 28 and the requirements specified by the automotive industry and suppliers of genuine spare parts.

Figure 11.49

11.8.5.3 ***Mounting example***

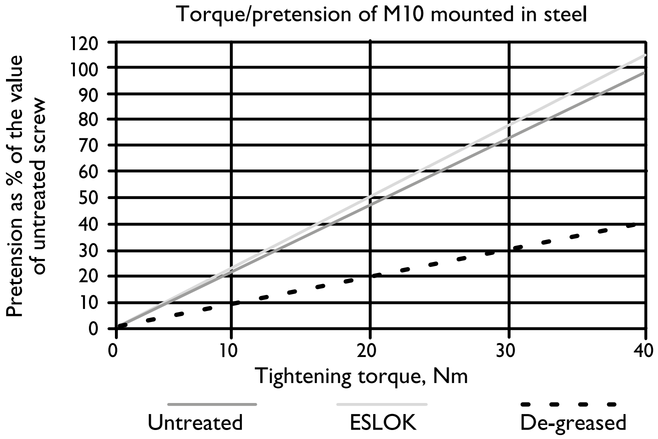


Figure 11.50

11.9 TORX®

For a long time, a more effective grip for tightening than the ordinary hexagonal form was sought. For weak screws, various cross recesses were developed early on, while it took longer for coarser screws which required greater tightening torque.

The difficulties were mainly of a marketing nature, because in most cases new forms of grip required new tightening tools, which in turn required effective coverage of the market so that customers should be able to have quick, easy access to both the screws and to the tools. Many firms tried to do this, but most failed as regards the sale of the tools, and this was the case with BULTEN's attempts with the HIGRIP product.

In the end, a company called CAMCAR in the USA succeeded in producing a new grip geometry, TORX, and acquired a sufficient number of licensed manufacturers both for the screw and the tool that market coverage was satisfactory.

One can compare the popularity of the TORX system with the TAPTITE threads. Thanks to their function and continuing technical development, both have established international dominance. The TORX grip can be put into most types of head.

11.9.1 COMPARISON OF PROPERTIES

Hexagon

The leverage effect on the corner of the head when tightening gives considerable force, which seeks to destroy the socket. In the best case, if one ignores play, the driving surface forms an angle of 60° against the radius out to one corner.

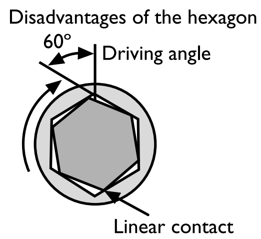


Figure 11.51

The torque-transferring force acts at a right-angle to the radius. This means in the best case a linear contact between the corner of the fastener head and the driving surface of the socket. In practice, however, one must always allow for play between the head and the tool. Therefore, the corner will lie slightly into the driving surface, and the force that threatens to destroy the socket increases.

Cross recess

The cross recess basically has a conical shape. When mounting, it is often difficult to keep the tip of the screwdriver exactly aligned with the screw, it often is a little out of true.

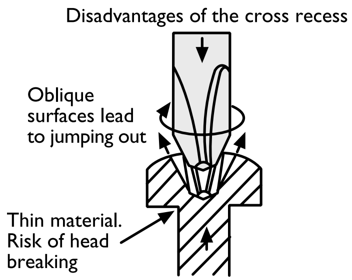


Figure 11.52

When torque is applied, a force arises which wants to push the tip of the screwdriver out of the slot. Initially, this jumping out can be counteracted by harder pressure on the tool, but with worn tools and high torque, it is difficult to prevent jumping out. Also, as a rule, jumping out also causes damage to surrounding surfaces.

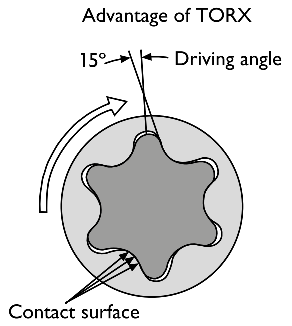


Figure 11.53

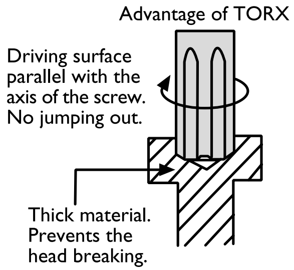


Figure 11.54

TORX

The enclosed curved form of the TORX system gives a very small drive angle, only 15°, which considerably reduces the radial breaking forces. Thanks to the fact that the curvatures engage, contact surfaces are formed and not linear contact.

The form of the grip also leads to a better overlap between internal and external grip. The so-called grip cover and surface contact eliminate deformation.

Higher torque can be transferred, while there is much less wear than is the case with the hexagonal grip. Especially with internal grip, the superior function of the system is obvious.

The driving surfaces of the TORX system are parallel with the axis of the screw. Therefore, there is no jumping out effect. The TORX system is so effective that the length of engagement of the tool can be reduced compared with other slot types. The thickness of material between hole bottom and screw shank can therefore be increased, which reduces the risk of the head breaking. With cross screws, this part is the weakest section.

11.9.2 TORSIONAL STRENGTH OF TORX SCREWDRIVER TIPS

TABLE 11.20 OUTER DIAMETER AND TORSIONAL STRENGTH OF TORX SCREWDRIVER TIPS

Grip	Outer diameter A ref	Torsional strength in Nm Min
T8	2.40	2.6
T10	2.80	4.52
T15	3.35	7.69
T20	3.95	12.7
T25	4.50	19.0
T30	5.60	37.4
T40	6.75	65.1
T45	7.95	103
T50	8.95	159
T55	11.35	257
T60	13.45	445
T70	15.70	700
T80	17.50	1050

For the optimum life of screwdriver tips with regard to torsion fatigue, we recommend that grip sizes are selected so that a maximum of 50% of the torsional strength is utilised in mechanical mounting.

11.9.3 LENGTH OF LIFE

Thanks to the fact that cylindrical surfaces are in contact between tool and screw grip, there is little wear. When the driving surfaces are also parallel with the axis of the screw, all axial forces are eliminated, which means that the tip of the screwdriver does not jump out. Therefore, the life of the screwdriver tip is several times longer than with cross recesses, for example.

11.9.4 ERGONOMICS

Internally, the TORX grip does not give any jumping out effect during tightening. The axial force on the tool can therefore be kept low. This is a big advantage for the person carrying out the mounting, who does not need to put full force on the tool in order to prevent it jumping out. Stress on arms and back is considerably reduced and makes mounting less tiring. Nor does he need to risk injuring hands and arms, which is common with other grip types if tools jump out.

11.9.5 LOW IN-PLACE COST

The TORX system reduces in-place cost thanks to faster mounting, long life of tools and thus fewer stoppages. Because jumping out is eliminated, no damage is caused to screws, tools or surrounding areas. Also, the muscles and joints of the assembly personnel suffer less strain when mounting with hand-held screwdrivers.

11.9.6 MAIN FORMS

The TORX grip can be applied to most standardised main forms, and has shown itself particularly appropriate for screws with countersunk heads. The high torsional strength makes it possible to utilise the tensile strength of these screws fully without deforming the screw grip, as happens with countersunk screws with internal hexagonal holes.

11.9.7 TORX ON SPECIAL SCREWS

For certain types of special screw, for example, in the automotive industry, the TORX grip has given considerable advantages, as designers have better opportunities for optimising the function of the screw joint. For example, the screws for securing seat belts, cylinder head studs and other screws in the engine and seats can be mentioned. In the design of special screws for tensioned screw joints, it is important that the strength of the screws is in harmony. The shearing area which is formed between the bottom of the grip and the shank of the screw must be at least 1.4 times greater than the nominal stress area A_s of the thread for the head not to come loose in an axial tensile test.

In order to meet all strength requirements in accordance with SS - ISO 898-1, the shear area must be at least 1.8 times greater than the nominal stress area.

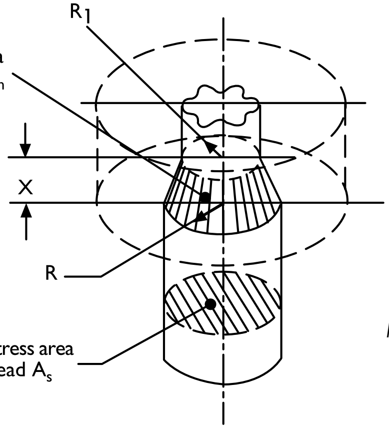


Figure 11.55

$A_h = \pi \times S\,(R+R_1)$

$S = \sqrt{(R - R_1)^2 + X^2}$

R_1 is half the diameter of the grip

R is the radius of the screw shank

X is the head height minus the grip depth

Requirements for harmony of strength:

Axial tension $A_h/A_s \geq 1.4$

Angled tension $A_h/A_s \geq 1.8$

11.10 TORX PLUS®

The TORX system has been further developed into a product called TORX PLUS, which can also be put into the main types of head. The grip geometry consists of elliptical forms.

11.10.1 THE GEOMETRY

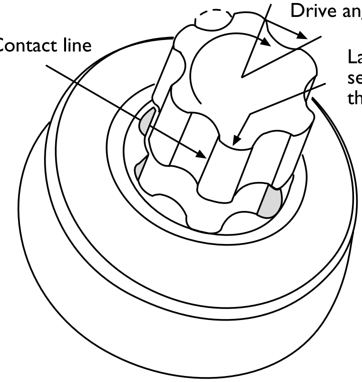
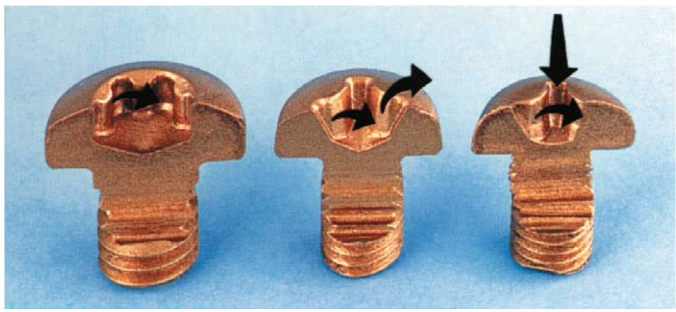


Figure 11.56

The improved design comprises:

- Drive angle 0° to eliminate radial forces which create tension in the screw grip (improved torque transfer).
- Larger cross section area in the collars.
- Elliptical geometry.
- Better filling throughout the entire length of the collar.
- Greater strength at screwdriver tips, permitting greater twisting torque.
- The tolerance of the screwdriver tips has been reduced by 50% to improve the fit between tool and screw grip.

Torque | Torque Jumping Out | Torque Axial Force



TORX PLUS | Cross recess | Cross recess

Figure 11.57

11.10.2 TORSIONAL STRENGTH OF TORX PLUS SCREWDRIVER TIPS

TABLE 11.21 OUTER DIAMETER AND TORSIONAL STRENGTH OF TORX PLUS SCREWDRIVER TIPS

Grip	Outer diameter A ref	Torsional strength in Nm Min
10 IP	2.82	5.42
15 IP	3.35	6.92
20 IP	3.94	16.1
25 IP	4.52	23.5
30 IP	5.61	47.2
40 IP	6.76	82.1
45 IP	7.92	137.4
50 IP	8.94	194.5
55 IP	11.33	352.0
60 IP	13.44	566.0
70 IP	15.72	910.0
80 IP	17.79	1290.0

Note: In order to obtain maximum life of screwdriver tips, we recommend that a maximum of 50% of torsional strength is utilised.

11.10.3 FATIGUE STRENGTH

The TORX PLUS grip system allows either 100% more screws to be mounted per screwdriver tip at a given tightening torque, or greater tightening torque over the life of the screwdriver tip compared with the TORX system.

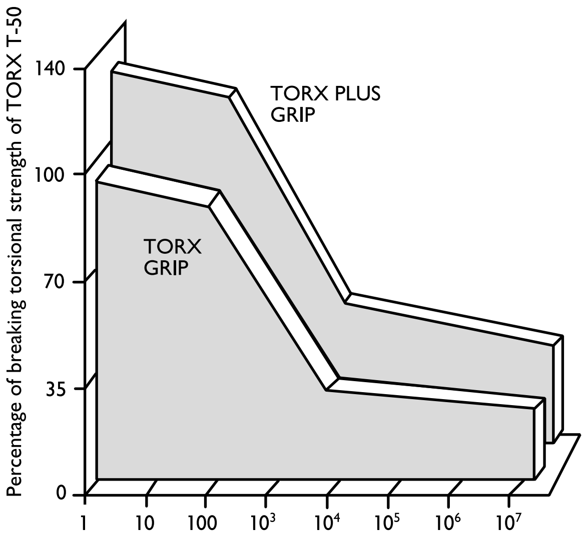


Figure 11.58

11.10.4 ADVANTAGES OF TORX PLUS

TORX PLUS gives:

- An average of double life for screwdriver tips.
- An average of 25% greater torsional strength.
- Greater transferred tightening torque.
- Even less risk of jumping out and lower axial load.
- Less physical load for the assembly personnel.
- Longer tool life than with other gripping systems.
- Greater productivity and reliability.
- Lower production costs in the long run.

11.10.5 AUTOSERT®

AUTOSERT allows higher rotation speeds when engaging on the assembly line.

11.10.5.1 *Geometry*

AUTOSERT (auto searching) is a supplement to the TORX PLUS system. An inclined guiding surface across each collar with the TORX PLUS grip allows a higher engagement rotation speed in automatic mounting by robots and in many other situations where screwdriver tips may rotate continuously.

AUTOSERT also creates a self-centring gripping function, which improves conditions for manual assembly where the screwdriver tip does not rotate continuously.

Rounded head with TORX PLUS grip and AUTOSERT

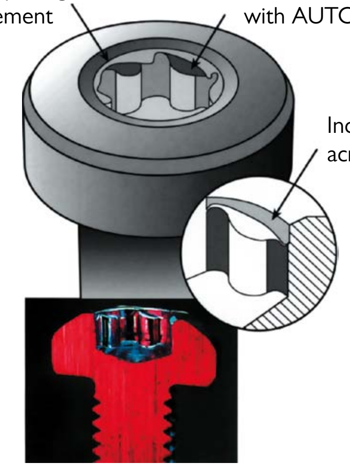


Figure 11.59

11.10.5.2 ***Engagement test***

Laboratory tests show that AUTOSERT engages 100% correctly at 700 revolutions per minute when TORX PLUS screwdriver tips are used.

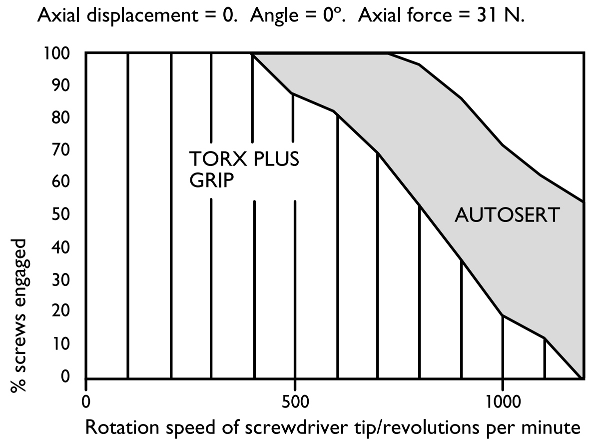


Figure 11.60

11.10.5.3 ***Advantages of AUTOSERT:***

- Self centring and engagement.
- The screwdriver tip engages in less than 1/6 of a revolution.
- Larger opening for the tool to slide down into the grip.
- Allows high rotation speed for engaging compared with standard grip.
- Ideal for robotised or automated mounting, and also for manual mounting.
- Improves the ability of the screwdriver tip to engage when it is at an angle or is displaced relative to the grip.
- In test mounting, users have found that the screwdriver tip suffers less wear.

11.10.6 COMPARISON BETWEEEN TORX AND TORX PLUS

TORX screwdriver tip
The geometry of the TORX grip is comprised of arcs, giving many advantages compared with other systems.

The TORX PLUS grip
The geometry of the TORX PLUS grip is comprised of ellipses and is an improvement on the TORX system.

Combination of systems
An internal TORX PLUS grip can be combined with a TORX system screwdriver tip if there are service problems in the field.

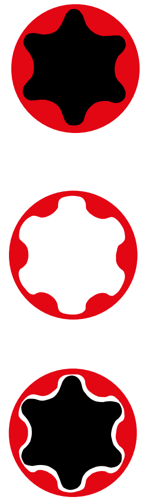


Figure 11.61

11.10.7 MANUFACTURERS AND TOOL DISTRIBUTORS

Figure 11.62

Using the TORX PLUS system, which is even better, is an excellent way of reducing in-place costs when mounting fasteners. In all cases where the TORX grip is used nowadays, one can use the TORX PLUS grip advantageously; it is available from a large number of fastener manufacturers throughout the world.

The TORX PLUS tool is available from a large number of manufacturers and tool suppliers throughout the world.

DO YOU NEED HELP?

This chapter is intended to provide help in design work as regards finding some important standards and necessary dimensions in the mounting of a threaded fastener.

Sections 12.1 – 12.7 list titles of standards divided into groups depending on content.

Standards are given with various designations, SS, SS-ENG, SS-EN ISO, depending on basis.

That is followed by Section 12.8 where we present selected parts of some of the most commonly used product standards. Then Section 12.9 which in essence is about dimensions required for mounting. Section 12.10 gives suitable depths of engagement in steel of various property classes to obtain holes in the material which have equal strength to the screws of various property classes. The basic dimensions of metric ISO threads and drill diameters for drilling before tapping are given in Section 12.11. Finally, in Section 12.12, we present some tables with tolerances and fundamental deviations of threaded fasteners.

12.1 TERMINOLOGY

SS-ISO 1891

Bolts, screws, nuts and accessories – Terminology and nomenclature.

SS-EN-20225

Fasteners – Bolts, screws, studs and nuts – Symbols and designation of dimensions.

12.2 THREADS

ISO 724

ISO general-purpose metric screw threads – Basic dimensions.

SS-ISO 965-1

ISO general-purpose metric screw threads – Tolerances – Part 1: Principles and basic data.

SS-ISO 965-2

ISO general-purpose metric screw threads – Tolerances – Part 2: Limits of sizes for general purpose bolt and nut threads – Medium quality.

SS-ISO 965-3

ISO general-purpose metric screw threads – Tolerances – Part 3: Deviations for constructional threads.

SS-EN ISO 1478

Fasteners – Tapping screw thread.

12.3 IN GENERAL

SS-ISO 272

Fasteners – Hexagon products – Widths across flats.

SS-EN 20273

Fasteners – Clearance holes for bolts and screws.

SS-ISO 888

Fasteners – Bolts, screws and studs – Nominal lengths, and thread lengths for general purpose bolts.

SS 3584

Fasteners – Bolts and screws with ISO metric screw threads – Short dog point with truncated cone end (entry dog) – Dimensions

12.4 TOLERANCES

SS-ISO 4759/1

Tolerances for fasteners – Part 1: Bolts, screws and nuts with thread diameters ≥ 1.6 and ≤ 150 mm and product grades A, B and C.

SS-ISO 4759/3

Tolerances for fasteners – Part 3: Washers to metric bolts, screws and nuts with thread diameters from 1 up to 150 mm – product grades A and C.

12.5 MECHANICAL PROPERTIES

SS-EN 20898-1 (SS-ISO898-1)

Mechanical properties of fasteners – Part1: Bolts, screws and studs.

SS-EN 20898-2 (SS-ISO898-2)

Mechanical properties of fasteners – Part2: Nuts with specified proof load values – Coarse thread.

SS-ISO 2320

Prevailing torque type steel hexagon nuts – Mechanical and performance properties.

SS-EN ISO 2702

Heat-treated steel tapping screws – Mechanical properties (ISO 2702:1992).

SS 3392

Tapping screws – Determination of strength and assembly properties.

12.6 SURFACE TREATMENT

ISO 4042

Threaded components – Electroplated coatings.

12.7 PRODUCTS

SS-EN 24014 (SS-ISO 4014)

Fasteners – Hexagon head bolts – Product grades A and B.

SS-EN 24017 (SS-ISO 4017)

Fasteners – Hexagon head screws – Product grades A and B.

ISO 4162[1])

Hexagon flange bolts – Small series.

SS-ISO 4762

Hexagon socket head cap screws – Product grade A.

SS-EN 24032 (SS-ISO 4032)

Fasteners – Hexagon nuts, style 1 – Product grades A and B.

ISO 4161[2])

Hexagon nuts with flange – Product grade A.

SS-ISO 7040

Prevailing torque type hexagon nuts (with non-metallic insert), style 1 – Property classes 5, 8 and 10.

SS-ISO 7043

Prevailing torque type hexagon nuts with flange (with non-metallic insert).

SS-ISO 7044

Prevailing torque type all-metal hexagon nuts with flange.

SS-EN ISO 7045

Pan head screws with type H or type Z cross recess – Product grade A (ISO 7045:1994).

SS-EN ISO 7046-1

Countersunk flat head screws (common head style) with type H or type Z cross recess – Product grade A – Part 1: Steel or property class 4.8 (ISO 7046-1:1994).

SS-EN ISO 7046-2

Cross recessed countersunk flat head screws (common head style) – Grade A – Part 2: Steel of property class 8.8, stainless steel and non-ferrous metals. (ISO 7046-2:1990).

1) Will be issued during 1999 as SS-EN 1662
2) Will be issued during 1999 as SS-EN 1661

SS-EN ISO 7049

Cross recessed pan head tapping screws (ISO 7049:1983).

SS-EN ISO 7050

Cross recessed countersunk (flat) head tapping screws (common head style) (ISO 7050:1983).

ISO 4026

Hexagon socket set screws with flat point.

ISO 4027

Hexagon socket set screws with cone point.

ISO 4028

Hexagon socket set screws with dog point.

ISO 4029

Hexagon socket set screws with cup point.

12.8 EXTRACTS FROM SOME PRODUCT STANDARDS

This section contains important parts of standards for some of the most common products. In some cases dimensions have been included from some recently revised standards that are expected to be issued in Sweden not later than 1999.

Index of tables

12.8.1 EXTRACT FROM SS-EN 24014, 1992 (SS-ISO 4014) PART-THREADED HEXAGONAL BOLTS

Product grades A and B.-

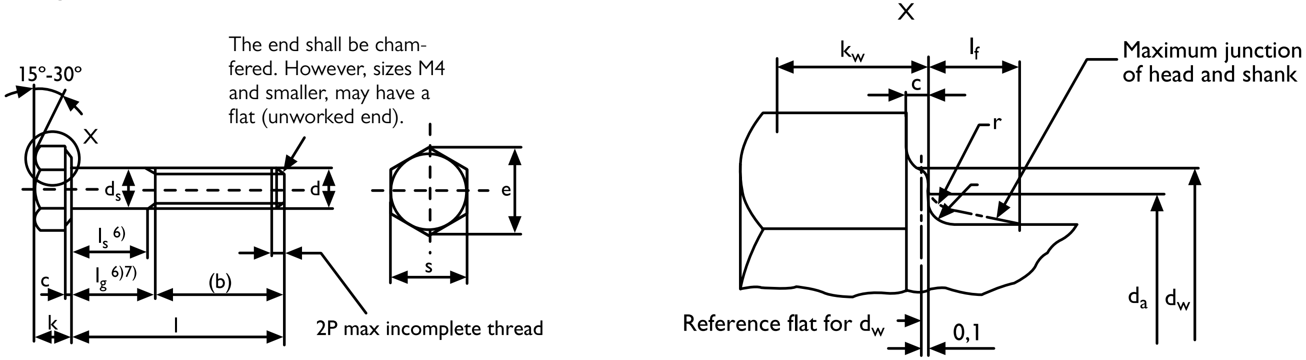


Figure 12.1

TABLE 12.1 PREFERRED THREADS

Dimensions in mm

Thread d		M1.6	M2	M2.5	M3	M4	M5	M6	M8	M10	M12	M16	M20	M24	M30	M36	M42	M48	M56	M64
P 1)		0.35	0.4	0.45	0.5	0.7	0.8	1	1.25	1.5	1.75	2	2.5	3	3.5	4	4.5	5	5.5	6
b ref.	2)	9	10	11	12	14	16	18	22	26	30	38	46	54	66	–	–	–	–	–
	3)	15	16	17	18	20	22	24	28	32	36	44	52	60	72	84	96	108	–	–
	4)	28	29	30	31	33	35	37	41	45	49	57	65	73	85	97	109	121	137	153
c	min	0.1	0.1	0.1	0.15	0.15	0.15	0.15	0.15	0.15	0.15	0.2	0.2	0.2	0.2	0.2	0.3	0.3	0.3	0.3
	max	0.25	0.25	0.25	0.4	0.4	0.5	0.5	0.6	0.6	0.6	0.8	0.8	0.8	0.8	0.8	1	1	1	1
d_a	max	2	2.6	3.1	3.6	4.7	5.7	6.8	9.2	11.2	13.7	17.7	22.4	26.4	33.4	39.4	45.6	52.6	63	71
d_s	nominal = max	1.6	2	2.5	3	4	5	6	8	10	12	16	20	24	30	36	42	48	56	64
	Product grade A min	1.46	1.86	2.36	2.86	3.82	4.82	5.82	7.78	9.78	11.73	15.73	19.67	23.67	–	–	–	–	–	–
	Product grade B min	1.35	1.75	2.25	2.75	3.70	4.70	5.70	7.64	9.64	11.57	15.57	19.48	23.48	29.48	35.38	41.38	47.38	55.26	63.26
d_w	Product grade A min	2.27	3.07	4.07	4.57	5.88	6.88	8.88	11.63	14.63	16.63	22.49	28.19	33.61	–	–	–	–	–	–
	Product grade B min	2.30	2.95	3.95	4.45	5.74	6.74	8.74	11.47	14.47	16.47	22	27.7	33.25	42.75	51.11	59.95	69.45	78.66	88.16
e	Product grade A min	3.41	4.32	5.45	6.01	7.66	8.79	11.05	14.38	17.77	20.03	26.75	33.53	39.98	–	–	–	–	–	–
	Product grade B min	3.28	4.18	5.31	5.88	7.50	8.63	16.89	14.20	17.59	19.85	26.17	32.95	39.55	50.85	60.79	71.3	82.6	93.56	104.86
l_f	max	0.6	0.8	1	1	1.2	1.2	1.4	2	2	3	3	4	4	6	6	8	10	12	13
k	nominal	1.1	1.4	1.7	2	2.8	3.5	4	5.3	6.4	7.5	10	12.5	15	18.7	22.5	26	30	35	40
	Product grade A min	0.975	1.275	1.575	1.875	2.675	3.35	3.85	5.15	6.22	7.32	9.82	12.285	14.785	–	–	–	–	–	–
	Product grade max	1.225	1.525	1.825	2.125	2.925	3.65	4.15	5.45	6.58	7.68	10.18	12.715	15.215	–	–	–	–	–	–
	Product grade B min	0.9	1.2	1.5	1.8	2.6	3.26	3.76	5.06	6.11	7.21	9.71	12.15	14.65	18.28	22.08	25.58	29.58	34.5	39.5
	Product grade max	1.3	1.6	1.9	2.2	3.0	3.74	4.24	5.54	6.69	7.79	10.29	12.85	15.35	19.12	22.92	26.42	30.42	35.5	40.5
k_w 5)	Product grade A min	0.68	0.89	1.1	1.31	1.87	2.35	2.7	3.61	4.35	5.12	6.87	8.6	10.35	–	–	–	–	–	–
	Product grade B min	0.63	0.84	1.05	1.26	1.82	2.28	2.63	3.54	4.28	5.05	6.8	8.51	10.26	12.8	15.46	17.91	20.71	24.15	27.65
r	min	0.1	0.1	0.1	0.1	0.2	0.2	0.25	0.4	0.4	0.6	0.6	0.8	0.8	1	1	1.2	1.6	2	2
s	nominal = max	3.2	4	5	5.5	7	8	10	13	16	18	24	30	36	46	55	65	75	85	95
	Product grade A min	3.02	3.82	4.82	5.32	6.78	7.78	9.78	12.73	15.73	17.73	23.67	29.67	35.38	–	–	–	–	–	–
	Product grade B min	2.90	3.70	4.70	5.20	6.64	7.64	9.64	12.57	15.57	17.57	23.16	29.16	35	45	53.8	63.1	73.1	82.8	92.8

1) P = pitch of the thread
2) For lengths $l_{nom} \leq 125$ mm
3) For lengths 125 mm < $l_{nom} \leq 200$ mm
4) For lengths $l_{nom} > 200$ mm

5) $k_w = 0.7\ k_{min}$
6) $l_{gmax} = l_{nom} - b$
$l_{smin} = l_{gmax} - 5\ P$
7) l_g is minimum clamping length

12.8.2 EXTRACT FROM SS-EN 24032, 1992 (SS-ISO 4032) HEXAGONAL NUTS, STYLE 1

Product grades A and B

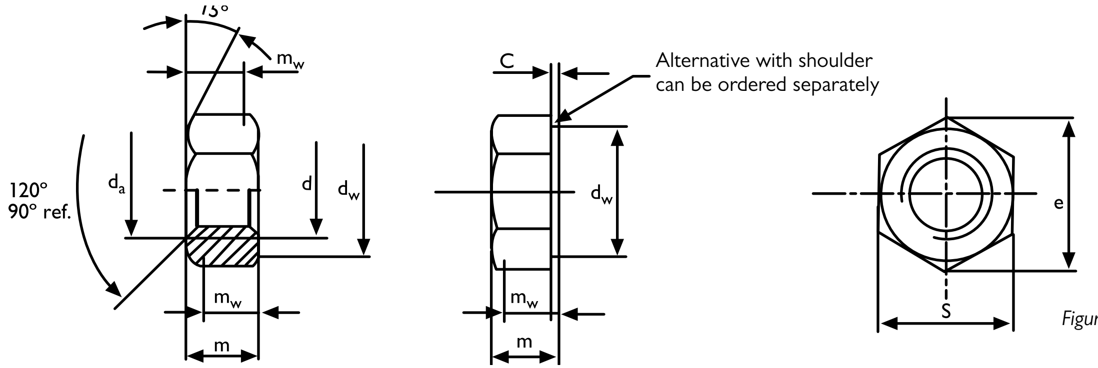


Figure 12.2

TABLE 12.2 PREFERRED SIZES

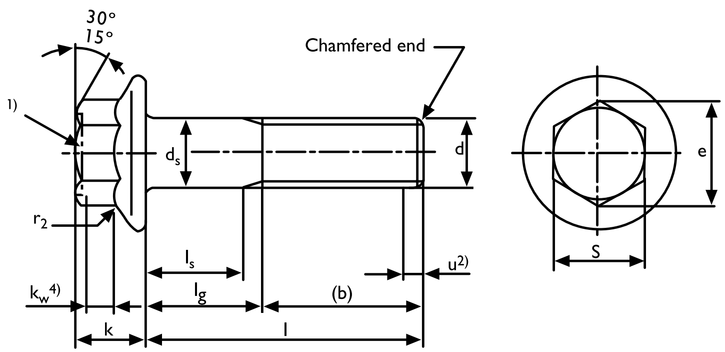
Dimensions in mm

Thread d		M1.6	M2	M2.5	M3	M4	M5	M6	M8	M10	M12	M16	M20	M24	M30	M36	M42	M48	M56	M64
P [1)]		0.35	0.4	0.45	0.5	0.7	0.8	1	1.25	1.5	1.75	2	2.5	3	3.5	4	4.5	5	5.5	6
c	max	0.2	0.2	0.3	0.4	0.4	0.5	0.5	0.6	0.6	0.6	0.8	0.8	0.8	0.8	0.8	1	1	1	1
	min	0.1	0.1	0.1	0.15	0.15	0.15	0.15	0.15	0.15	0.15	0.2	0.2	0.2	0.2	0.2	0.3	0.3	0.3	0.3
d_a	min	1.6	2	2.5	3	4	5	6	8	10	12	16	20	24	30	36	42	48	56	64
	max	1.84	2.3	2.9	3.45	4.6	5.75	6.75	8.75	10.8	13	17.3	21.6	25.9	32.4	38.9	45.4	51.8	60.5	69.1
d_w	min	2.4	3.1	4.1	4.6	5.9	6.9	8.9	11.6	14.6	16.6	22.5	27.7	33.3	42.8	51.1	60	69.5	78.7	88.2
e	min	3.41	4.32	5.45	6.01	7.66	8.79	11.05	14.38	17.77	20.03	26.75	32.95	39.55	50.85	60.79	71.3	82.6	93.56	104.86
m	max	1.3	1.6	2	2.4	3.2	4.7	5.2	6.8	8.4	10.8	14.8	18	21.5	25.6	31	34	38	45	51
	min	1.05	1.35	1.75	2.15	2.9	4.4	4.9	6.44	8.04	10.37	14.1	16.9	20.2	24.3	29.4	32.4	36.4	43.4	49.1
m_w	min	0.8	1.1	1.4	1.7	2.3	3.5	3.9	5.2	6.4	8.3	11.3	13.5	16.2	19.4	23.5	25.9	29.1	34.7	39.3
s nom =	max	3.2	4	5	5.5	7	8	10	13	16	18	24	30	36	46	55	65	75	85	95
	min	3.02	3.82	4.82	5.32	6.78	7.78	9.78	12.73	15.73	17.73	23.67	29.16	35	45	53.8	63.1	73.1	82.8	92.8

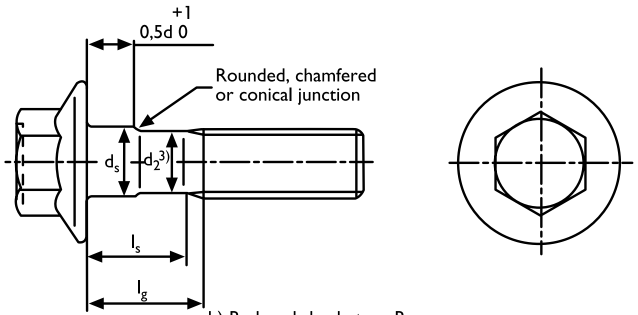
1) P = pitch of the thread

12.8.3 EXTRACT FROM SS-EN 1662, 1998*) FLANGED BOLTS WITH HEXAGONAL HEAD

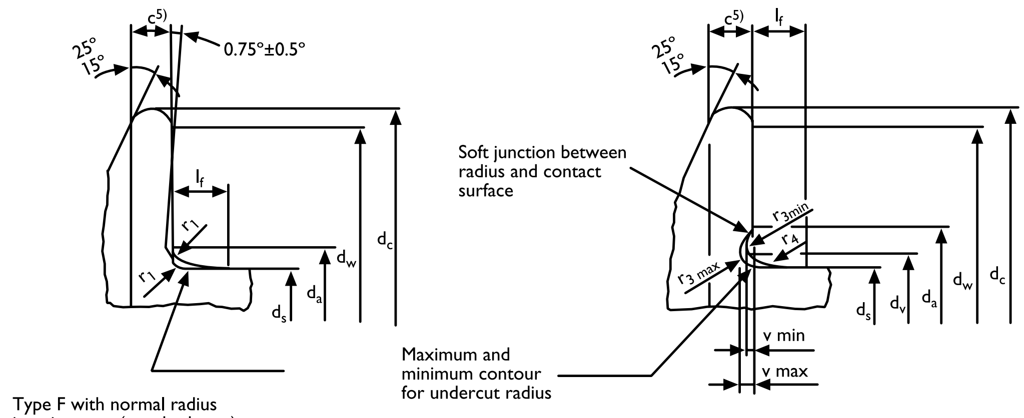
Product grade A



a) Coarse shank
(standard type)



b) Reduced shank, type R
(only available on special order)



Type F with normal radius
junction area (standard type)

Type U with undercut radius
junction (on request or optional)

c) Form of the contact surface

Figure 12.3

1) The topside of the head can either be flat or have a chamfered or spherical countersinking as requested
2) Incomplete thread, ≤ 2P
3) d_2 equals roughly the pitch diameter of the thread or the thread rolling diameter
4) k_w is minimum engagement depth
5) c is measured at $d_w$min

*) EN 1662 is issued in Europe instead of ISO 4162.

TABLE 12.3

Dimensions in mm

Thread d		M5	M6	M8	M10	M12	(M14) [1]	M16
P [2]		0.8	1	1.25	1.5	1.75	2	2
b ref	[3]	16	18	22	26	30	34	38
	[4]	–	–	28	32	36	40	44
	[5]	–	–	–	–	–	–	57
c	min	1	1.1	1.2	1.5	1.8	2.1	2.4
d_a Type F	max	5.7	6.8	9.2	11.2	13.7	15.7	17.7
d_a Type U	max	6.2	7.5	10	12.5	15.2	17.7	20.5
d_c	max	11.4	13.6	17	20.8	24.7	28.6	32.8
d_s	max	5.00	6.00	8.00	10.00	12.00	14.00	16.00
	min	4.82	5.82	7.78	9.78	11.73	13.73	15.73
d_v	max	5.5	6.6	8.8	10.8	12.8	14.8	17.2
d_w	min	9.4	11.6	14.9	18.7	22.5	26.4	30.6
e	min	7.59	8.71	10.95	14.26	17.62	19.86	23.15
k	max	5.6	6.9	8.5	9.7	12.1	12.9	15.2
k_w	min	2.3	2.9	3.8	4.3	5.4	5.6	6.8
l_f	max	1.4	1.6	2.1	2.1	2.1	2.1	3.2
r_1	min	0.2	0.25	0.4	0.4	0.6	0.6	0.6
r_2 [6]	max	0.3	0.4	0.5	0.6	0.7	0.9	1
r_3	max	0.25	0.26	0.36	0.45	0.54	0.63	0.72
	min	0.10	0.11	0.16	0.20	0.24	0.28	0.32
r_4	ref	4	4.4	5.7	5.7	5.7	5.7	8.8
s	max	7.00	8.00	10.00	13.00	16.00	18.00	21.00
	min	6.78	7.78	9.78	12.73	15.73	17.73	20.67
v	max	0.15	0.20	0.25	0.30	0.35	0.45	0.50
	min	0.05	0.05	0.10	0.15	0.15	0.20	0.25

1) Sizes in brackets should be avoided
2) P = pitch of the thread
3) For lengths $l_{nom} \leq 125$ mm
4) For lengths 125 mm $< l_{nom} \leq 200$ mm
5) For lengths $l_{nom} > 200$ mm
6) Radius r_2 applies to the junctions between the corners of the hexagonals and between the sides and flanges of the hexagonals

12.8.4 **EXTRACT FROM SS-EN 1661, 1998*) HEXAGON NUTS WITH FLANGE**

Product grade A

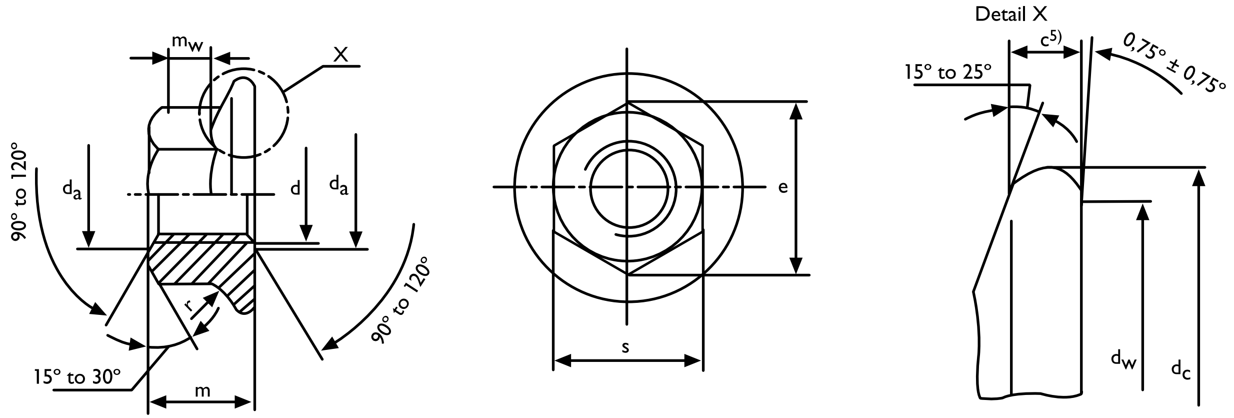


Figure 12.4

TABLE 12.4

Dimensions in mm

Thread d		M5	M6	M8	M10	M12	(M14) 1)	M16	M20
P 2)		0.8	1	1.25	1.5	1.75	2	2	2.5
c	min	1	1.1	1.2	1.5	1.8	2.1	2.4	3
d_a	min	5	6	8	10	12	14	16	20
	max	5.75	6.75	8.75	10.8	13	15.1	17.3	21.6
d_c	max	11.8	14.2	17.9	21.8	26	29.9	34.5	42.8
d_w	min	9.8	12.2	15.8	19.6	23.8	27.6	31.9	39.9
e	min	8.79	11.05	14.38	17.77	20.03	23.36	26.75	32.95
m	max	5	6	8	10	12	14	16	20
	min	4.7	5.7	7.64	9.64	11.57	13.3	15.3	18.7
m_w 3)	min	2.5	3.1	4.6	5.9	6.8	7.7	8.9	10.7
s	max	8	10	13	16	18	21	24	30
	min	7.78	9.78	12.73	15.73	17.73	20.67	23.67	29.16
r 4)	max	0.3	0.36	0.48	0.6	0.72	0.88	0.96	1.2

1) Sizes in brackets should be avoided if possible
2) P = pitch of the thread
3) Minimum grip height
4) Radius r applies to the junction between the corners of the hexagonals and between the sides and flanges of the hexagonals
5) c measured at $d_w$min

*) EN 1661 is issued in Europe instead of ISO 4161.

12.8.5 EXTRACT FROM SS-EN ISO 4762, 1998 (SS-ISO 4762) HEXAGONAL SOCKET HEAD CAP SCREWS

Product grade A

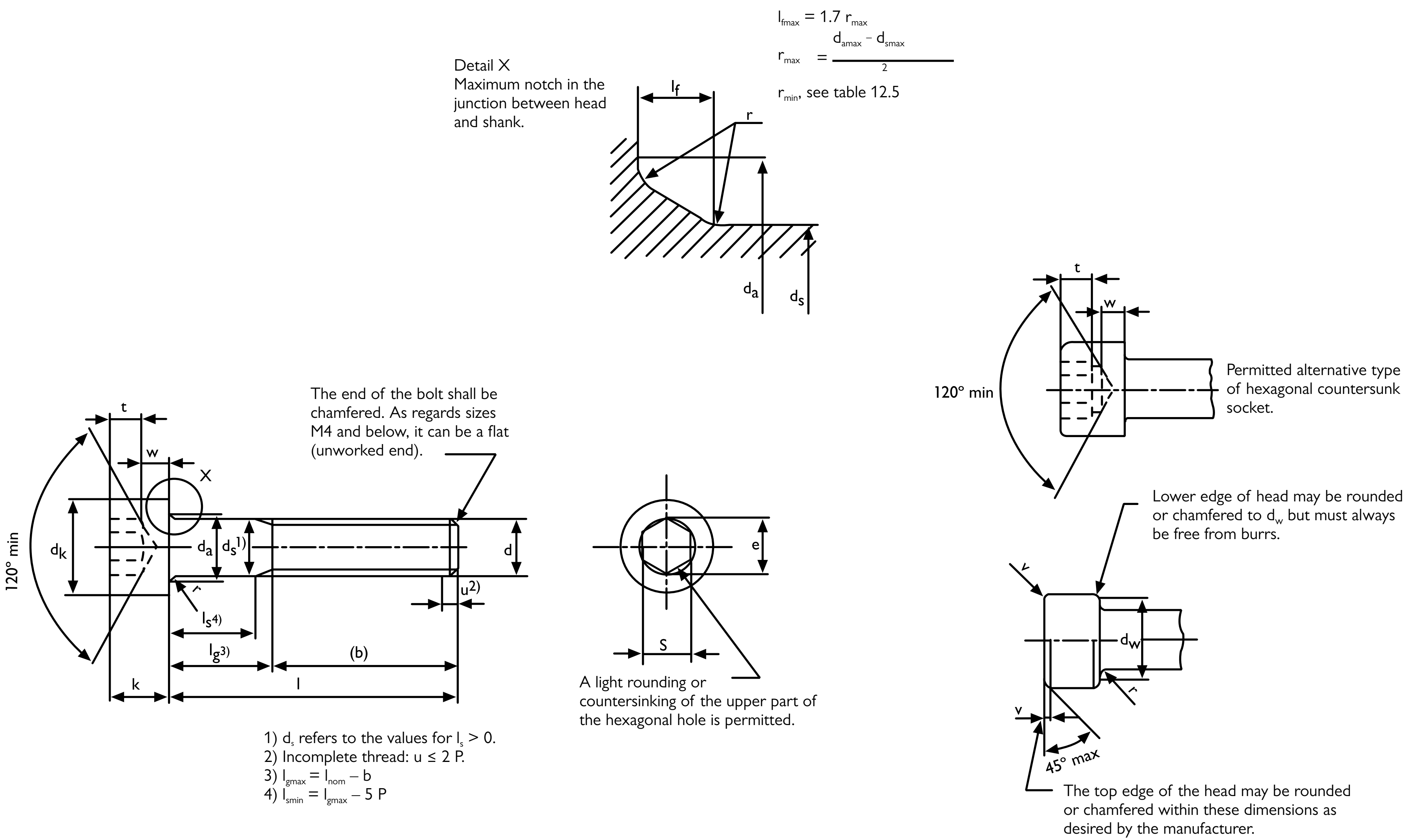


Figure 12.5

TABLE 12.5

Dimensions in mm

Thread d		M1.6	M2	M2.5	M3	M4	M5	M6	M8	M10	M12	(M14) [2]	M16	M20	M24	M30	M36
P [1]		0.35	0.4	0.45	0.5	0.7	0.8	1	1.25	1.5	1.75	2	2	2.5	3	3.5	4
b	ref	15	16	17	18	20	22	24	28	32	36	40	44	52	60	72	84
d_k	max [3]	3	3.8	4.5	5.5	7	8.5	10	13	16	18	21	24	30	36	45	54
	max [4]	3.14	3.98	4.68	5.68	7.22	8.72	10.22	13.27	16.27	18.27	21.33	24.33	30.33	36.39	45.39	54.46
	min	2.86	3.62	4.32	5.32	6.78	8.28	9.78	12.73	15.73	17.73	20.67	23.67	29.67	35.61	44.61	53.54
d_a	max	2	2.6	3.1	3.6	4.7	5.7	6.8	9.2	11.2	13.7	15.7	17.7	22.4	26.4	33.4	39.4
d_s	max	1.6	2	2.5	3	4	5	6	8	10	12	14	16	20	24	30	36
	min	1.46	1.86	2.36	2.86	3.82	4.82	5.82	7.78	9.78	11.73	13.73	15.73	19.67	23.67	29.67	35.61
e	min [5]	1.73	1.73	2.3	2.87	3.44	4.58	5.72	6.86	9.15	11.43	13.72	16	19.44	21.73	25.15	30.85
l_f	max	0.34	0.51	0.51	0.51	0.6	0.6	0.68	1.02	1.02	1.45	1.45	1.45	2.04	2.04	2.89	2.89
k	max	1.6	2	2.5	3	4	5	6	8	10	12	14	16	20	24	30	36
	min	1.46	1.86	2.36	2.86	3.82	4.82	5.7	7.64	9.64	11.57	13.57	15.57	19.48	23.48	29.48	35.38
r	min	0.1	0.1	0.1	0.1	0.2	0.2	0.25	0.4	0.4	0.6	0.6	0.6	0.8	0.8	1	1
s	nom	1.5	1.5	2	2.5	3	4	5	6	8	10	12	14	17	19	22	27
	min	1.52	1.52	2.02	2.52	3.02	4.02	5.02	6.02	8.025	10.025	12.032	14.032	17.05	19.065	22.065	27.065
	max [6]	1.545	1.545	2.045	2.56	3.071	4.084	5.084	6.095	8.115	10.115	12.142	14.142	17.23	19.275	22.275	27.275
	max [7]	1.56	1.56	2.06	2.58	3.08	4.095	5.14	6.14	8.175	10.175	12.212	14.212				
t	min	0.7	1	1.1	1.3	2	2.5	3	4	5	6	7	8	10	12	15.5	19
v	max	0.16	0.2	0.25	0.3	0.4	0.5	0.6	0.8	1	1.2	1.4	1.6	2	2.4	3	3.6
d_w	min	2.72	3.48	4.18	5.07	6.53	8.03	9.38	12.33	15.33	17.23	20.17	23.17	28.87	34.81	43.61	52.54
w	min	0.55	0.55	0.85	1.15	1.4	1.9	2.3	3.3	4	4.8	5.8	6.8	8.6	10.4	13.1	15.3

1) P = pitch of the thread
2) Sizes in brackets should be avoided
3) For knurled heads
4) For smooth-sided heads
5) $e_{min} = 1.14 \times s_{min}$
6) For property class 12.9
7) For all other property classes

12.8.6 EXTRACT FROM SS-EN ISO 4757, 1994, CROSS RECESSES FOR CROSS-RECESSED SCREWS AND BOLTS

This standard now includes type H (previously Phillips) and type Z (previously Pozidriv), which was not possible previously for patent reasons. Apart from dimensions, the standard also contains interpretations and instructions regarding how the cross recesses should be controlled. Tables 12.6 and 12.7 only give the dimensions of the cross recesses, the widths and depths of which can be adapted to the various forms of head shown in current product standards.

TABLE 12.6
CROSS RECESS TYPE H, THEORETICAL DIMENSIONS

Cross recess no.		0	1	2	3	4
b	0 / –0.03	0.61	0.97	1.47	2.41	3.48
e		0.26–0.36	0.41–0.46	0.79–0.84	1.98–2.03	2.39–2.44
g	+0.05 / 0	0.81	1.27	2.29	3.81	5.08
f		0.31–0.36	0.51–0.56	0.66–0.74	0.79–0.86	1.19–1.27
r	nom	0.3	0.5	0.6	0.8	1
t_1	ref	0.22	0.34	0.61	1.01	1.35
α	0 / –15’	1)	138°	140°	146°	153°
ß	+15’ / 0	7°	7°	5° 45’	5° 45’	7°

1) This is replaced by r_{min} 0.25 mm; r_{max} 0.36 mm

Dimensions in mm

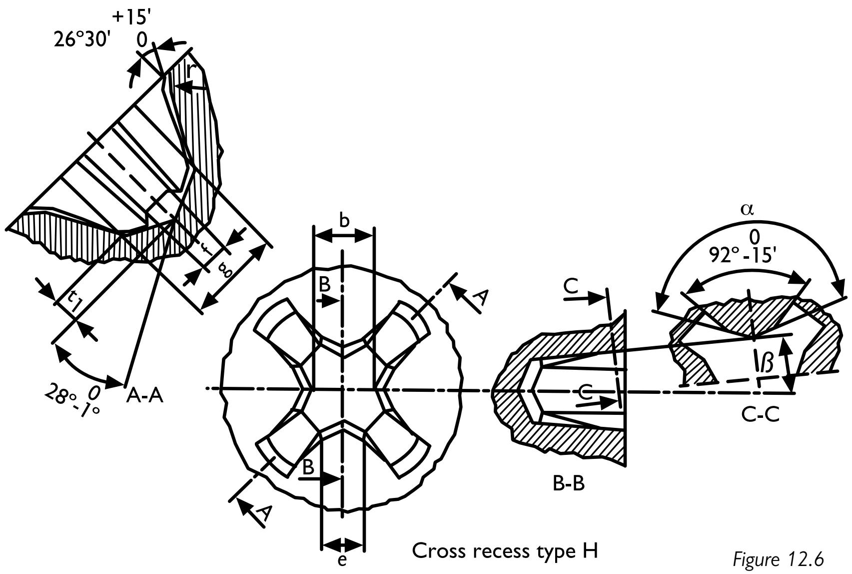


Figure 12.6

TABLE 12.7
CROSS RECESS TYPE Z, THEORETICAL DIMENSIONS

Cross recess no.	0	1	2	3	4
b $^{0}_{-0.05}$	0.76	1.27	1.83	2.72	3.96
f $^{0}_{-0.025}$	0.48	0.74	1.03	1.42	2.16
g $^{0}_{-0.05}$	0.86	1.32	2.34	3.86	5.08
r_1 max	0.30	0.30	0.38	0.51	0.64
r_2 max	0.10	0.13	0.15	0.25	0.38
j max	0.13	0.15	0.15	0.20	0.20
α $^{+15'}_{0}$	7°	7°	5° 45’	5° 45’	7°
ß $^{0}_{-15'}$	7° 45’	7° 45’	6° 20’	6° 20’	7° 45’
y $^{0}_{-15'}$	4° 23’	4° 23’	3°	3°	4° 23’
δ $^{0}_{-7'}$	46°	46°	46°	56° 15’	56° 15’

Dimensions in mm

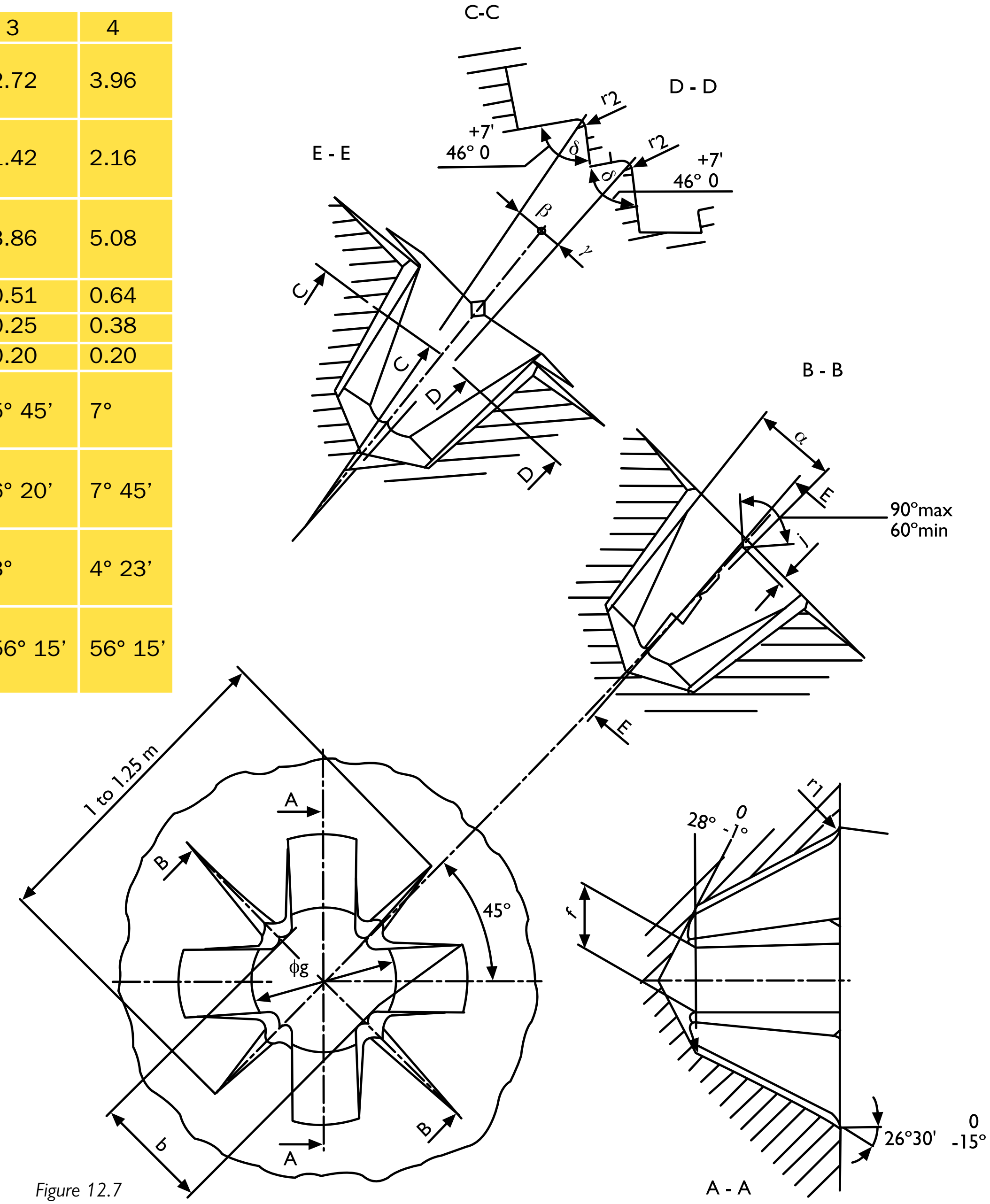


Figure 12.7

12.8.7 EXTRACT FROM SS-EN ISO 7045, 1995 (SS-ISO 7045), PAN HEAD SCREWS WITH TYPE H OR TYPE Z CROSS RECESS

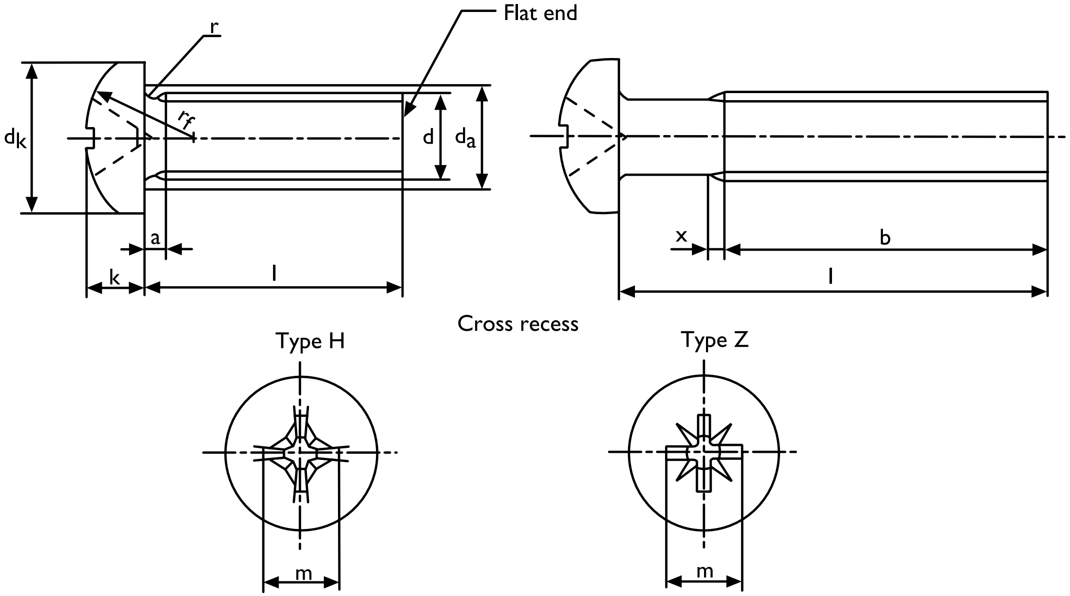


Figure 12.8

TABLE 12.8

Dimensions in mm

Thread d		M1.6	M2	M2.5	M3	(M3.5) [1]	M4	M5	M6	M8	M10
P [2]		0.35	0.4	0.45	0.5	0.6	0.7	0.8	1	1.25	1.5
a	max	0.7	0.8	0.9	1	1.2	1.4	1.6	2	2.5	3
b	min	25	25	25	25	38	38	38	38	38	38
d_a	max	2	2.6	3.1	3.6	4.1	4.7	5.7	6.8	9.2	11.2
d_k	nom = max	3.2	4.0	5.0	5.6	7.00	8.00	9.5	12.00	16.00	20.00
	min	2.9	3.7	4.7	5.3	6.64	7.64	9.14	11.57	15.57	19.48
k	nom = max	1.30	1.60	2.10	2.40	2.60	3.10	3.70	4.6	6.0	7.50
	min	1.16	1.46	1.96	2.26	2.46	2.92	3.52	4.3	5.7	7.14
r	min	0.1	0.1	0.1	0.1	0.1	0.2	0.2	0.25	0.4	0.4
r_f	≈	2.5	3.2	4	5	6	6.5	8	10	13	16
x	max	0.9	1	1.1	1.25	1.5	1.75	2	2.5	3.2	3.8
Cross recess	no.	0		1		2		3		4	
Type H, m	ref	1.7	1.9	2.7	3	3.9	4.4	4.9	6.9	9	10.1
Type H depth of penetration	max	0.95	1.2	1.55	1.8	1.9	2.4	2.9	3.6	4.6	5.8
	min	0.70	0.9	1.15	1.4	1.4	1.9	2.4	3.1	4.0	5.2
Cross recess Type Z, m	ref	1.6	2.1	2.6	2.8	3.9	4.3	4.7	6.7	8.8	9.9
Type Z depth of penetration	max	0.90	1.42	1.50	1.75	1.93	2.34	2.74	3.46	4.50	5.69
	min	0.65	1.17	1.25	1.50	1.48	1.89	2.29	3.03	4.05	5.24

1) Sizes in brackets should be avoided if possible
2) P = pitch of the thread

12.9 MOUNTING DIMENSIONS FOR BOLTED JOINTS

The following pages give mounting dimensions and gripping lengths for the most common bolted joints. Other dimensions, measurements and tolerances are given in relevant product and basic standards, which are referred to in the figure texts.

Values with two decimal places have been rounded off to one decimal place.

The standards are always subject to change. The most reliable method of always having access to the latest standards is to subscribe to the subject area with SIS, which sells the Swedish standard.

12.9.1 MAIN DIMENSIONS OF SCREWS AND BOLTS

The s and e dimensions of flanged screws and bolts are less than those of hexagonal screws and bolts. The correct dimensions are obtained from the product standards.

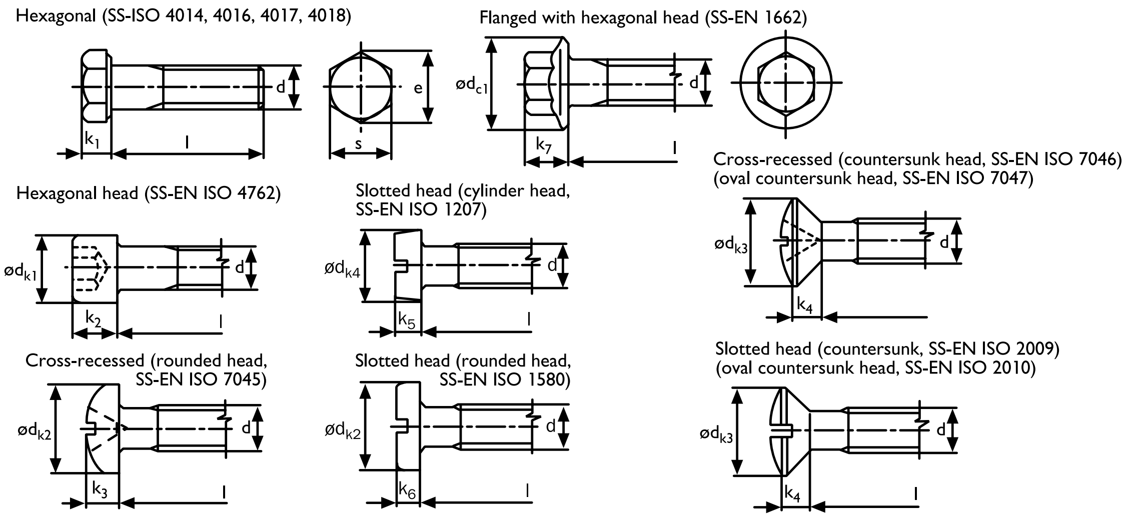


Figure 12.9

TABLE 12.9 SCREWS AND BOLTS

Thread d	d_{c1} max	d_{k1} 1) max	d_{k2} max	d_{k3} max	d_{k4} max	k_1 nom	k_2 max	k_3 max	k_4 max	k_5 max	k_6 max	k_7 max	e 3) min	s max
M1.6		3	3.2	3	3	1.1 2)	1.6	1.3	1	1.1	1		3.4	3.2 2)
M2		3.8	4	3.8	3.8	1.4 2)	2	1.6	1.2	1.4	1.3		4.3	4 2)
M2.5		4.5	5	4.7	4.5	1.7 2)	2.5	2.1	1.5	1.8	1.5		5.5	5 2)
M3		5.5	5.6	5.5	5.5	2 2)	3	2.4	1.65	2	1.8		6	5.5 2)
M4		7	8	8.4	7	2.8 2)	4	3.1	2.7	2.6	2.4		7.7	7 2)
M5	11.4	8.5	9.5	9.3	8.5	3.5	5	3.7	2.7	3.3	3	5.6	8.8	8
M6	13.6	10	12	11.3	10	4	6	4.6	3.3	3.9	3.6	6.9	11.1	10
M8	17	13	16	15.8	13	5.3	8	6	4.65	5	4.8	8.5	14.4	13
M10	20.8	16	20	18.3	16	6.4	10	7.5	5	6	6	9.7	17.8	16
M12	24.7	18				7.5	12					12.1	20	18
(M14)	28.6	21				8.8	14					12.9	23.4	21
M16	32.8	24				10	16					15.2	26.8	24
M20		30				12.5	20						33.5	30
M24		36				15	24						40	36
M30		45				18.7	30						51	46
M36		54				22.5	36						61	55

1) With knurled heads see SS-ISO 4762 for dimension d_{k1}
2) Does not apply to product grade C
3) Does not apply to product grades B and C

12.9.2 WASHERS

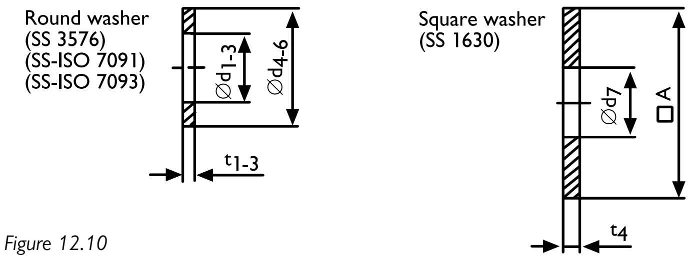


Figure 12.10

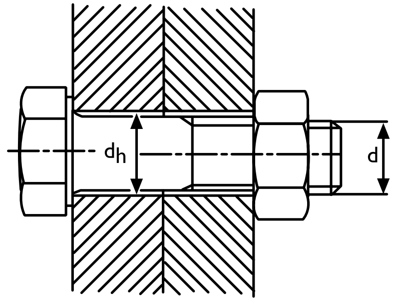


Figure 12.11

TABLE 12.10 WASHERS AND CLEARANCE HOLES

Nominal size	A nom	d_1 4) max	d_2 5) max	d_3 6) max	d_4 4) max	d_5 5) max	d_6 6) max	d_7 max	t_1 4) max	t_2 5) max	t_3 6)	t_4	d_h 7) min	d_h 7) max
M1.6		1.84			4				0.35				1.8	1.94
M2		2.34			5				0.35				2.4	2.54
M2.5		2.84			6				0.55				2.9	3.04
M3		3.38		3.38	7		9		0.55		0.9		3.4	3.58
M4		4.48		4.48	9		12		0.9		1.1		4.5	4.68
M5		5.5	5.8	5.5	10	10	15		1.1	1.2	1.4		5.5	5.68
M6		6.6	7	6.6	12	12	18		1.8	1.9	1.8		6.6	6.82
M8		8.6	9.4	8.6	16	16	24		1.8	1.9	2.2		9	9.22
M10	30	10. 8	11.4	10.8	20	20	30	11.4	2.2	2.3	2.7	3	11	11.27
M12	40	13.3	13.9	13.3	24	24	37	13.9	2.7	2.8	3.3	4	13.5	13.77
(M14)		15.3	15.9	15.3	28	28	44		2.7	2.8	3.3		15.5	15.77
M16	50	17.3	17.9	17.3	30	30	50	18.2	3.3	3.6	3.3	5	17.5	17.77
M20	60	21.3	22.5	22.5	37	37	60	22.8	3.3	3.6	4.6	5	22	22.33
M24	80	25.3	26.5	26.8	44	44	72	26.8	4.3	4.6	6	6	26	26.33
M30	95	31.4	33.6	34	56	56	92	34	4.3	4.6	7	6	33	33.39
M36	110	37.6	40	40	66	66	110	37	5.6	6	9.2	8	39	39.39

4) Round washer, product grade A (SS 3576)
5) Round washer, product grade C (SS-ISO 7091)
6) Large round washer, product grades A and C (SS-ISO 7093)
7) Clearance hole d_h, series median in accordance with SS-ISO 273

Note. For dimensions in excess of 5 mm, the 1st decimal may be rounded off

12.9.3 **NUTS**

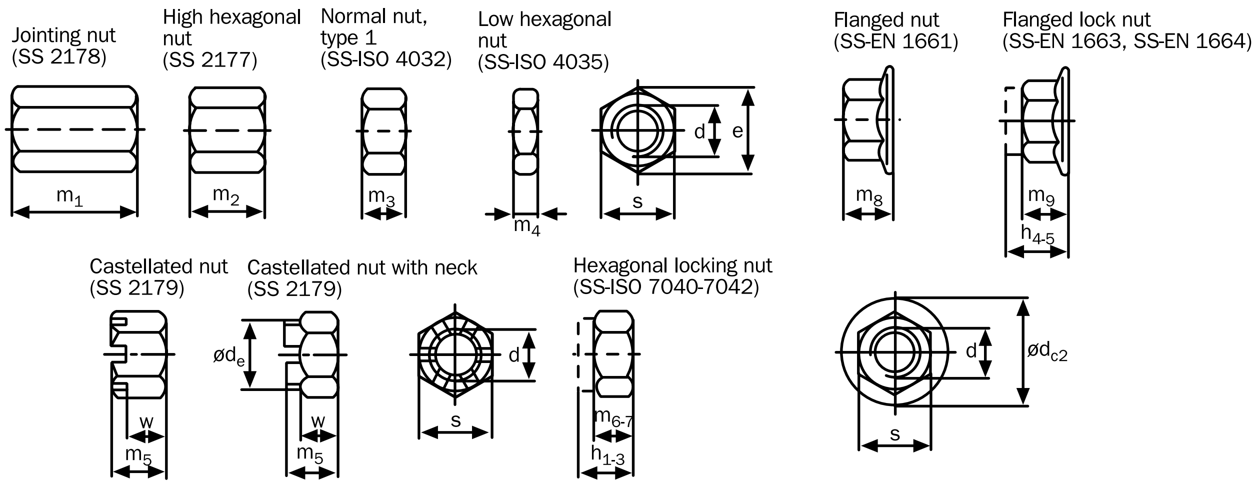


Figure 12.12

TABLE 12.11 NUTS

Thread d	d_{c2}	d_e max	h_1 [8] max	h_2 [9] max	h_3 [10] max	h_4 [11] max	h_5 [12] max	m_1	m_2	m_3 [13] max	m_4 max	m_5 max	m_6 [8] min	m_7 [9] min	m_8 max	m_9 min	e min	s max	w max
M1.6										1.3	1						3.4	3.2	
M2										1.6	1.2						4.3	4	
M2.5										2	1.6						5.5	5	
M3			4.5							2.4	1.8		2.2				6	5.5	
M4			6							3.2	2.2	5	2.9				7.7	7	3.2
M5	11.8		6.8	7.2	5.1	7.1	6.2			4.7	2.7	6	4.4	4.8	5	4.7	8.8	8	4
M6	14.2		8	8.5	6	9.1	7.3		9	5.2	3.2	7.5	4.9	5.4	6	5.7	11.1	10	5
M8	17.9		9.5	10.2	8	11.1	9.4		12	6.8	4	9.5	6.4	7.1	8	7.6	14.4	13	6.5
M10	21.8		11.9	12.8	10	13.5	11.4		15	8.4	5	12	8	8.9	10	9.6	17.8	16	8
M12	26	16	14.9	16.1	12	16.1	13.8	50	18	10.8	6	15	10.4	11.6	12	11.6	20	18	10
(M14)	29.9	18	17	18.3	14.1	18.2	15.9		21	12.8	7	16	12.1	13.4	14	13.3	23.4	21	11
M16	34.5	22	19.1	20.7	16.4	20.3	18.3	50	24	14.8	8	19	14.1	15.7	16	15.3	26.8	24	13
M20	42.8	28	22.8	25.1	20.3	24.8	22.4	50	30	18	10	22	16.9	19	20	18.7	33	30	16
M24		34	27.1	29.5	23.9			50	36	21.5	12	27	20.2	22.6			39.6	36	19
M30		42	32.6	35.6	30				45	25.6	15	33	24.3	27.3			50.9	46	24
M36		50	38.9	42.6	36				54	31	18	38	29.4	33.1			60.8	55	29

8) Hexagonal lock nut, non-metallic insert, type 1 (SS-ISO 7040).
9) Hexagonal lock nut, non-metallic insert, type 2 (SS-ISO 7041).
10) Hexagonal lock nut, fully metallic, type 2 (SS-ISO 7042).
11) Flanged lock nut, fully metallic (SS-EN 1663).
12) Flanged lock nut, fully metallic (SS-EN 1664).
13) Normal nut, type 1 (SS-ISO 4032). Type 2 is in SS-ISO 4033

12.9.4 GRIPPING LENGTHS

The gripping length L_1 can be calculated on the basis of the following formula:

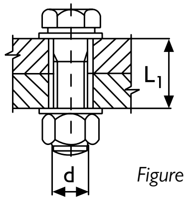


Figure 12.13

$$L_1 = l - m_{1\text{–}9} - 2\,(P + t_{1\text{–}4})$$

where:

l	is nominal bolt length
$m_{1\text{-}9}$	is nut height according to Table 12.11
P	is thread pitch
$t_{1\text{-}4}$	is the thickness of the washer according to Table 12.10

The gripping lengths do not apply to bolts with countersunk heads.

The gripping lengths in Table 12.12 are calculated for bolts with a head without a flange together with nuts in accordance with SS-ISO 4032 (m_3 in Table 12.11) and two round washers in accordance with SS 3576 (t_1 in accordance with Table 12.10) according to the formula:

$$L_1 = l - m_3 - 2\,(P + t_1)$$

The gripping lengths in Table 12.13 are calculated for flanged bolts with flanged nuts in accordance with SS-EN 1661 (m_8 in Table 12.11) but without washers according to the formula:

$$L_1 = l - m_8 - 2P$$

TABLE 12.12 GRIPPING LENGTHS

Thread	Gripping length, L_1 max																		
	Nominal bolt length, l																		
	6	8	10	12	16	20	25	30	35	40	45	50	55	60	65	70	80	90	100
M3	1.5	3.5	5.5	7.5	11.5	15.5	20.5	25.5											
M4		1.6	3.6	5.6	9.6	13.6	18.6	23.6	28.6	33.6									
M5			1.5	3.5	7.5	11.5	16.5	21.5	26.5	31.5	36.5	41.5							
M6				1.2	5.2	9.2	14.2	19.2	24.2	29.2	34.2	39.2	44.2	49.2					
M8					3.1	7.1	12.1	17.1	22.1	27.1	32.1	37.1	42.1	47.1	52.1	57.1	67.1		
M10						4.2	9.2	14.2	19.2	24.2	29.2	34.2	39.2	44.2	49.2	54.2	64.2	74.2	84.2
M12							5.3	10.3	15.3	20.3	25.3	30.3	35.3	40.3	45.3	50.3	60.3	70.3	80.3
(M14)								7.8	12.8	17.8	22.8	27.8	32.8	37.8	42.8	47.8	57.8	67.8	77.8
M16								4.6	9.6	14.6	19.6	24.6	29.6	34.6	39.6	44.6	54.6	64.6	74.6
M20										10.4	15.4	20.4	25.4	30.4	35.4	40.4	50.4	60.4	70.4
M24												13.9	18.9	23.9	28.9	33.9	43.9	53.9	63.9
M30														18.8	23.8	28.8	38.8	48.8	58.8
M36																19.8	29.8	39.8	49.8

TABLE 12.13 FLANGE BOLTS WITH FLANGE NUTS

Thread d	Gripping length, L_1 max																
	Nominal bolt length, l																
	10	12	16	20	25	30	35	40	45	50	55	60	65	70	80	90	100
M5	3.4	5.4	9.4	3.5	13.4	18.4	23.4	28.4	33.4	38.4							
M6		4	8	12	17	22	27	32	37	42	47	52					
M8			5.5	9.5	14.5	19.5	24.5	29.5	34.5	39.5	44.5	49.5	54.5	59.5	69.5		
M10				7	12	17	22	27	32	37	42	47	52	57	67	77	87
M12					9.5	14.5	19.5	24.5	29.5	34.5	39.5	44.5	49.5	54.5	64.5	74.5	84.5
(M14)						12	17	22	27	32	37	42	47	52	62	72	82
M16							15	20	25	30	35	40	45	50	60	70	90

12.9.5 DIAMETERS OF FLATS AND COUNTERSINKING (SS 2173)

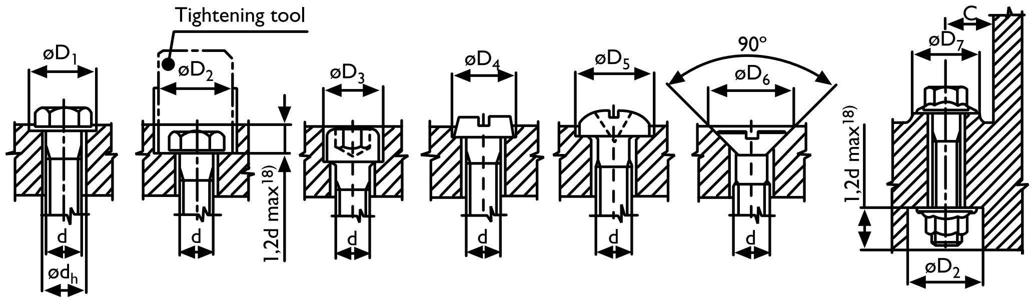


Figure 12.14

TABLE 12.14 DIAMETERS OF FLATS AND COUNTERSINKING

Thread d	D_1 H15	D_2H14 Hand operated socket				Drive socket					D_3 H14	D_4 H14	D_5 H14	D_6 H14	D_7 min	C [14] [16]	[17]
M1.6	5	8 [18]									3.3	3.3	4.3	3.7		6.5	
M2	6	8 [18]									4.3	4.3	5	4.6		6.5	
M2.5	8	10 [18]									5	5	6	5.7		6.5	
M3	8	10 [18]									6	6	7	5.7		6.5	
M4	11	13 [18]	13 [18]			15 [18]					8	8	9	9.7		11	11
M5	11	15 [15]	15 [15][18]			16 [18]					10	10	11	10.7	13	11	15
M6	13	18	18 [18]	18 [18]		18 [18]	20 [18]				11	11	13	12.9	15	13	15
M8	18	20	20 [18]	22 [18]		22 [18]	24 [18]				15	15	18	17.7	20	13	15
M10	24		24 [18]	26 [18]		26 [18]	28 [18]	30 [18]			18	18	22	20.4	24	13	20
M12	26		26 [15]	28		30	30 [18]	33 [18]	36 [18]		20				28	14	25
M14	30		30 [15]	33	36 [18]		36 [18]	36 [18]	40 [18]		24				33	21	25
M16	33			36	40 [18]		40	40	43 [18]		26				36	21	25
M20	40			43 [15]	48			48	53	53 [18]	33				48	23	31
M24	48				53			57	61	61	40					27	31
M30	61				66				71	76	48					34	37
M36	71				78					89	57					40	37
Square shaft for socket, etc		6.3	10	12.5	20	10	12.5	16	20	25							

14) Applies to hexagonal bolts and hexagonal nuts, both with and without flange
15) For flange bolts in accordance with SS-EN 1662 and flange nuts in accordance with SS-EN 1661 except M5 D2 deviates as follows:
M5 : 13 mm
M12 : 28 mm
M14 : 33 mm
M20 : 48 mm

16) Hand operated socket
17) Drive socket
18) When the dimension marked in the illustration is > 1.2 d, D_2 increases

12.10 SCREWS IN TAPPED BLIND HOLES, METRIC ISO THREADS, COARSE PITCH (SS 1964)

Blind holes are theoretically calculated on the shear area of the thread wall and checked with mandrel tensile test to sustain at least the failure load of the screw.

The length of thread engagement L_2 is selected with reference to the breaking strength of the screw and the material in accordance with Table 12.15.

The thread length G, in accordance with Table 12.16, is determined with reference to the fact that the screw can be mounted without a washer

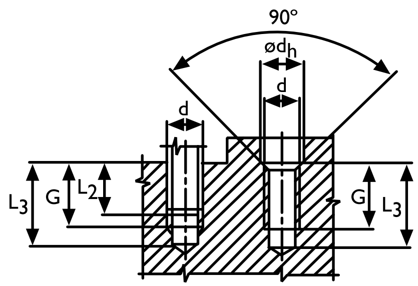


Figure 12.15

TABLE 12.15
LENGTH OF THREAD ENGAGEMENT L_2 IN ACCORDANCE WITH 1964

Ultimate tensile strength of material R_m N/mm² ca	Length of thread engagement L_2 The property class of the screw 4.6	5.8	8.8	10.9	12.9
130-200	1.5 D	-	-	-	-
200-280	1 D	1.5 D	-	-	-
280-350	1 D	1 D	1.5 D	-	-
350-430	1 D	1 D	1.2 D	1.5 D	-
430-540	0.8 D	1 D	1 D	1.2 D	1.5 D
540-650	0.8 D	0.8 D	1 D	1 D	1.2 D
650-750	0.8 D	0.8 D	1 D	1 D	1 D
750-	0.8 D	0.8 D	0.8 D	1 D	1 D

TABLE 12.16
CLEARANCE HOLES d_h IN ACCORDANCE WITH SS-ISO 273 AND THREADED BLIND HOLES

Thread d	d_h series medium	L_2=0.8d G	L_3	L_2=1d G	L_3	L_2=1.2d G	L_3	L_2=1.5d G	L_3
M1.6	1.8	2	4	2.5	4.5	3	5	3.5	5.5
M2	2.4	2.5	5	3	5.5	3.5	6	4	6.5
M2.5	2.9	3.5	6	4	6.5	4.5	7	5	7.5
M3	3.4	4	7	4	7	5	8	6	9
M4	4.5	5	9	6	10	7	11	8	12
M5	5.5	6	10.5	7	11.5	8	12.5	9.5	14
M6	6.6	8	13	9	14	10	15	12	17
M8	9	10	16	11.5	17.5	13	19	15.5	21.5
M10	11	12	19.5	14	21.5	16	23.5	19	26.5
M12	13 [19]	14	22.5	16.5	25	19	27.5	22.5	31
M14	15 [19]	16	25.5	19	28.5	22	31.5	26	35.5
M16	17 [19]	19	28.5	22	31.5	25	34.5	30	39.5
M20	21 [19]	23	34	27	38	31	42	38	49
M24	25 [19]	28	41	32.5	45.5	37	50	45	58
M30	31 [19]	33	48	39	54	45	60	57	72
M36	37 [19]	40	57	47	64	54	71	65	82

19) Series fine

12.11 EXTRACT FROM SS-ISO 724, ISO GENERAL PURPOSE METRIC SCREW THREADS, BASIC DIMENSIONS AND EXTRACT FROM SS 1724. DRILL DIAMETERS PRIOR TO THREADING

TABLE 12.17 THREAD DIAMETERS AND DRILL DIAMETERS

Thread d	M1.6	M2	M2.5	M3	M4	M5	M6	M8	M10	M12	M14	M16
Pitch diameter d_2, D_2	1.373	1.740	2.208	2.675	3.545	4.480	5.350	7.188	9.026	10.863	12.701	14.701
Minor diameter d_1, D_1	1.221	1.567	2.013	2.459	3.242	4.134	4.917	6.647	8.376	10.106	11.835	13.835
Pitch P	0.35	0.4	0.45	0.5	0.7	0.8	1	1.25	1.5	1.75	2	2
Drill diameter for threading	1.25	1.6	2.05	2.5	3.3	4.2	5	6.8	8.5	10.2	12	14

Thread d	M18	M20	M22	M24	M27	M30	M33	M36	M39
Pitch diameter d_2, D_2	16.376	18.376	20.376	22.051	25.051	27.727	30.727	33.402	36.402
Minor diameter d_1. D_1	15.294	17.294	19.294	20.752	23.752	26.211	29.211	31.670	34.670
Pitch P	2.5	2.5	2.5	3	3	3.5	3.5	4	4
Drill diameter for threading	15.5	17.5	19.5	21	24	26.5	29.5	32	35

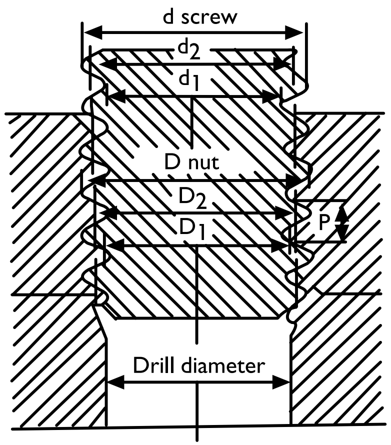


Figure 12.16

TOLERANCES AND FUNDAMENTAL DEVIATION

In this section some tables are presented which give numerical values for standard tolerances IT and fundamental deviations for axes (opposing surfaces) and for holes (facing surfaces). The values are taken from SS-ISO 286-1. “ISO system of limits and fits – Part 1. Bases of tolerances, deviations and fits”.

Also, two tables are presented from SS-ISO 4759-1 which directly affect tolerances for standardised, threaded fasteners.

TABLE 12.18 NUMERICAL VALUES OF STANDARD TOLERANCES IT FOR BASIC DIMENSIONS UP TO 3150

Basic dimensions mm		Standard tolerances																	
		IT1	IT2	IT3	IT4	IT5	IT6	IT7	IT8	IT9	IT10	IT11	IT12	IT13	IT14 [1]	IT15 [1]	IT16 [1]	IT17 [1]	IT18 [1]
Over	up to	Tolerance ranges µm											mm						
–	3	0.8	1.2	2	3	4	6	10	14	25	40	60	0.1	0.14	0.25	0.4	0.6	1	1.4
3	6	1	1.5	2.5	4	5	8	12	18	30	48	75	0.12	0.18	0.3	0.48	0.75	1.2	1.8
6	10	1	1.5	2.5	4	6	9	15	22	36	58	90	0.15	0.22	0.36	0.58	0.9	1.5	2.2
10	18	1.2	2	3	5	8	11	18	27	43	70	110	0.18	0.27	0.43	0.7	1.1	1.8	2.7
18	30	1.5	2.5	4	6	9	13	21	33	52	84	130	0.21	0.33	0.52	0.84	1.3	2.1	3.3
30	50	1.5	2.5	4	7	11	16	25	39	62	100	160	0.25	0.39	0.62	1	1.6	2.5	3.9
50	80	2	3	5	8	13	19	30	46	74	120	190	0.3	0.46	0.74	1.2	1.9	3	4.6
80	120	2.5	4	6	10	15	22	35	54	87	140	220	0.35	0.54	0.87	1.4	2.2	3.5	5.4
120	180	3.5	5	8	12	18	25	40	63	100	160	250	0.4	0.63	1	1.6	2.5	4	6.3
180	250	4.5	7	10	14	20	29	46	72	115	185	290	0.46	0.72	1.15	1.85	2.9	4.6	7.2
250	315	6	8	12	16	23	32	52	81	130	210	320	0.52	0.81	1.3	2.1	3.2	5.2	8.1
315	400	7	9	13	18	25	36	57	89	140	230	360	0.57	0.89	1.4	2.3	3.6	5.7	8.9
400	500	8	10	15	20	27	40	63	97	155	250	400	0.63	0.97	1.55	2.5	4	6.3	9.7
500	630						44	70	110	175	280	440	0.7	1.1	1.75	2.8	4.4	7	11
630	800						50	80	125	200	320	500	0.8	1.25	2	3.2	5	8	12.5
800	1000						56	90	140	230	360	560	0.9	1.4	2.3	3.6	5.6	9	14
1000	1250						66	105	165	260	420	660	1.05	1.65	2.6	4.2	6.6	10.5	16.5
1250	1600						78	125	195	310	500	780	1.25	1.95	3.1	5	7.8	12.5	19.5
1600	2000						92	150	230	370	600	920	1.5	2.3	3.7	6	9.2	15	23
2000	2500						110	175	280	440	700	1100	1.75	2.8	4.4	7	11	17.5	28
2500	3150						135	210	330	540	860	1350	2.1	3.3	5.4	8.6	13.5	21	33

1) Standard tolerances IT14 to IT18 shall not be used for basic dimensions less than or equal to 1 mm

TABLE 12.19 NUMERICAL VALUES OF FUNDAMENTAL DEVIATION FOR AXES

Basic dim. mm		Upper deviation limit es												Values for		
Over	up to	All tolerance grades												IT5 IT6	IT7	IT8
		a [1]	b [1]	c	cd	d	e	ef	f	fg	g	h	js [2]	j		
–	3 [1]	–270	–140	–60	–34	–20	–14	–10	–6	–4	–2	0	Deviation limit = $\pm \frac{ITn}{2}$ where n is the IT grade number	–2	–4	–6
3	6	–270	–140	–70	–46	–30	–20	–14	–10	–6	–4	0		–2	–4	
6	10	–280	–150	–80	–56	–40	–25	–18	–13	–8	–5	0		–2	–5	
10	14															
14	18	–290	–150	–95		–50	–32		–16		–6	0		–3	–6	
18	24															
24	30	–300	–160	–110		–65	–40		–20		–7	0		–4	–8	
30	40	–310	–170	–120												
40	50	–320	–180	–130		–80	–50		–25		–9	0		–5	–10	
50	65	–340	–190	–140												
65	80	–360	–200	–150		–100	–60		–30		–10	0		–7	–12	
80	100	–380	–220	–170												
100	120	–410	–240	–180		–120	–72		–36		–12	0		–9	–15	
120	140	–460	–260	–200												
140	160	–520	–280	–210		–145	–85		–43		–14	0		–11	–18	
160	180	–580	–310	–230												
180	200	–660	–340	–240												
200	225	–740	–380	–260		–170	–100		–50		–15	0		–13	–21	
225	250	–820	–420	–280												
250	280	–920	–480	–300		–190	–110		–56		–17	0		–16	–26	
280	315	–1050	–540	–330												
315	355	–1200	–600	–360		–210	–125		–62		–18	0		–18	–28	
355	400	–1350	–680	–400												
400	450	–1500	–760	–440		–230	–135		–68		–20	0		–20	–32	
450	500	–1650	–840	–480												
500	560					–260	–145		–76		–22	0				
560	630															
630	710					–290	–160		–80		–24	0				
710	800															
800	900					–320	–170		–86		–24	0				
900	1000															
1000	1120					–350	–195		–98		–28	0				
1120	1250															
1250	1400					–390	–220		–110		–30	0				
1400	1600															
1600	1800					–430	–240		–120		–32	0				
1800	2000															
2000	2240					–480	–260		–130		–34	0				
2240	2500															
2500	2800					–520	–290		–145		–38	0				
2800	3150															

1) Fundamental deviation for tolerances a and b shall not be used for basic dimensions less than or equal to 1 mm.

2) For tolerances js7 to js11: If the IT grade, n, is an odd number, it can be rounded off to the nearest lower even number so that the deviation limit, i.e., $\pm \frac{IT}{2}$ can be expressed in whole micrometres

CONTINUATION OF TABLE 12.19 Values of fundamental deviation in micrometres

fundamental deviation		Lower deviation limit el													
IT4 to IT7	up to IT3 and over IT7	All tolerance grades													
	k	m	n	p	r	s	t	u	v	x	y	z	za	zb	zc
0	0	+2	+4	+6	+10	+14		+18		+20		+26	+32	+40	+60
+1	0	+4	+8	+12	+15	+19		+23		+28		+35	+42	+50	+80
+1	0	+6	+10	+15	+19	+23		+28		+34		+42	+52	+67	+97
+1	0	+7	+12	+18	+23	+28		+33		+40		+50	+64	+90	+130
									+39	+45		+60	+77	+108	+150
+2	0	+8	+15	+22	+28	+35		+41	+47	+54	+63	+73	+98	+136	+188
							+41	+48	+55	+64	+75	+88	+118	+160	+218
+2	0	+9	+17	+26	+34	+43	+48	+60	+68	+80	+94	+112	+148	+200	+274
							+54	+70	+81	+97	+114	+136	+180	+242	+325
+2	0	+11	+20	+32	+41	+53	+66	+87	+102	+122	+144	+172	+226	+300	+405
					+43	+59	+75	+102	+120	+146	+174	+210	+274	+360	+480
+3	0	+13	+23	+37	+51	+71	+91	+124	+146	+178	+214	+258	+335	+445	+585
					+54	+79	+104	+144	+172	+210	+254	+310	+400	+525	+690
+3	0	+15	+27	+43	+63	+92	+122	+170	+202	+248	+300	+365	+470	+620	+800
					+65	+100	+134	+190	+228	+280	+340	+415	+535	+700	+900
					+68	+108	+146	+210	+252	+310	+380	+465	+600	+780	+1000
+4	0	+17	+31	+50	+77	+222	+166	+236	+284	+350	+425	+520	+670	+880	+1150
					+80	+130	+180	+258	+310	+385	+470	+575	+740	+960	+1250
					+84	+140	+196	+284	+340	+425	+520	+640	+820	+1050	+1350
+4	0	+20	+34	+56	+94	+158	+218	+315	+385	+475	+580	+710	+920	+1200	+1550
					+98	+170	+240	+350	+425	+525	+650	+790	+1000	+1300	+1700
+4	0	+21	+37	+62	+108	+190	+268	+390	+475	+590	+730	+900	+1150	+1150	+1900
					+114	+208	+294	+435	+530	+660	+820	+1000	+1300	+1650	+2100
+5	0	+23	+40	+68	+126	+232	+330	+490	+595	+740	+920	+1100	+1450	+1850	+2400
					+132	+252	+360	+540	+660	+820	+1000	+1250	+1600	+2100	+2600
0	0	+26	+44	+78	+150	+280	+400	+600							
					+155	+310	+450	+660							
0	0	+30	+50	+88	+175	+340	+500	+740							
					+185	+380	+560	+840							
0	0	+34	+56	+100	+210	+430	+620	+940							
					+220	+470	+680	+1050							
0	0	+40	+66	+120	+250	+520	+780	+1150							
					+260	+580	+840	+1300							
0	0	+48	+78	+140	+300	+640	+960	+1450							
					+330	+720	+1050	+1600							
0	0	+58	+92	+170	+370	+820	+1200	+1850							
					+400	+920	+1350	+2000							
0	0	+68	+110	+195	+440	+1000	+1500	+2300							
					+460	+1100	+1650	+2500							
0	0	+76	+135	+240	+550	+1250	+1900	+2900							
					+580	+1400	+2100	+3200							

TABLE 12.20 NUMERICAL VALUES OF FUNDAMENTAL DEVIATION FOR HOLES

Basic dim. mm		Lower deviation limit EI																	Values of	
Over	up to	All tolerance grades												IT6	IT7	IT8	Up to IT8	Over IT8	Up to IT8	Over IT8
		A 1)	B 1)	C	CD	D	E	EF	F	FG	G	H	JS 2)	J			K 3)		M 3)4)	
–	3 1)5)	+270	+140	+60	+34	+20	+14	+10	+6	+4	+2	0		+2	+4	+6	0	0	–2	–2
3	6	+270	+140	+70	+46	+30	+20	+14	+10	+6	+4	0	Deviation limit = $\pm \frac{ITn}{2}$ where n is the IT grade number	+5	+6	+10	–1+Δ		–4+Δ	–4
6	10	+280	+150	+80	+56	+40	+25	+18	+13	+8	+5	0		+5	+8	+12	–1+Δ		–6+Δ	–6
10	14	+290	+150	+95		+50	+32		+16		+6	0		+6	+10	+15	–1+Δ		–7+Δ	–7
14	18																			
18	24	+300	+160	+110		+65	+40		+20		+7	0		+8	+12	+20	–3+Δ		–8+Δ	–8
24	30																			
30	40	+310	+170	+120		+80	+50		+25		+9	0		+10	+14	+24	–2+Δ		–9+Δ	–9
40	50	+320	+180	+130																
50	65	+340	+190	+140		+100	+60		+30		+10	0		+13	+18	+28	–2+Δ		–11+Δ	–13
65	80	+360	+200	+150																
80	100	+380	+220	+170		+120	+72		+36		+12	0		+16	+22	+34	–3+Δ		–13+Δ	–13
100	120	+410	+240	+180																
120	140	+460	+260	+200		+145	+85		+43		+14	0		+18	+26	+41	–3+Δ		–15+Δ	–15
140	160	+520	+280	+210																
160	180	+580	+310	+230																
180	200	+660	+340	+240		+170	+100		+50		+15	0		+22	+30	+47	–4+Δ		–17+Δ	–17
200	225	+740	+380	+260																
225	250	+820	+420	+280																
250	280	+920	+480	+300		+190	+110		+56		+17	0		+25	+36	+55	–4+Δ		–20+Δ	–20
280	315	+1050	+540	+330																
315	355	+1200	+600	+360		+210	+125		+62		+18	0		+29	+39	+60	–4+Δ		–21+Δ	–21
355	400	+1350	+680	+400																
400	450	+1500	+760	+440		+230	+135		+68		+20	0		+33	+43	+66	–5+Δ		–23+Δ	–23
450	500	+1650	+840	+480																
500	560					+260	+145		+76		+22	0					0		–26	
560	630																			
630	710					+290	+160		+80		+24	0					0		–30	
710	800																			
800	900					+320	+170		+86		+26	0					0		–34	
900	1000																			
1000	1120					+350	+195		+98		+28	0					0		–40	
1120	1250																			
1250	1400					+390	+220		+110		+30	0					0		–48	
1400	1600																			
1600	1800					+430	+240		+120		+32	0					0		–58	
1800	2000																			
2000	2240					+480	+260		+130		+34	0					0		–68	
2240	2500																			
2500	2800					+520	+290		+145		+38	0					0		–76	
2800	3150																			

1) The fundamental deviation for tolerances A and B shall not be used for basic dimensions less than or equal to 1 mm.

2) For tolerances JS7 up to JS11: If the IT grade, n, is an odd number, it can be rounded off to the nearest lower even number so that the deviation limits, i.e., $\pm \frac{ITn}{2}$ can be expressed in full micrometres.

3) To determine the values of tolerances K, M and N for tolerance grades up to IT8 and deviation limits for tolerances P up to ZC for tolerance grades up IT7, take the Δ values from the right-hand column.

CONTINUATION OF TABLE 12.20

Values of fundamental deviation in micrometres

Fundamental deviation		Upper deviation limit ES														Values of Δ					
up to IT8	over IT8	up to IT7	Tolerance grades over IT7													Tolerance grades					
N [3)5)]		PtoZC [3)]	P	R	S	T	U	V	X	Y	Z	ZA	ZB	ZC	IT3	IT4	IT5	IT6	IT7	IT8
–4	–4	Values of tolerance grades over IT7 increased by Δ	–6	–10	–14		–18		–20		–26	–32	–40	–60	0	0	0	0	0	0
–8+Δ	0		–12	–15	–19		–23		–28		–35	–42	–50	–80	1	1.5	1	3	4	6
–10+Δ	0		–15	–19	–23		–28		–34		–42	–52	–67	–97	1	1.5	2	3	6	7
–12+Δ	0		–18	–23	–28		–33		–40		–50	–64	–90	–130	1	2	3	3	7	9
								–39	–45		–60	–77	–108	–150						
–15+Δ	0		–22	–28	–35		–41	–47	–54	–63	–73	–98	–136	–188	1.5	2	3	4	8	12
						–41	–48	–55	–64	–75	–88	–118	–160	–218						
–17+Δ	0		–26	–34	–43	–48	–60	–68	–80	–94	–112	–148	–200	–274	1.5	3	4	5	9	14
						–54	–70	–81	–97	–114	–136	–180	–242	–325						
–20+Δ	0		–32	–41	–53	–66	–87	–102	–122	–144	–172	–226	–300	–405	2	3	5	6	11	16
				–43	–59	–75	–102	–120	–146	–174	–210	–274	–360	–480						
–23+Δ	0		–37	–51	–71	–91	–124	–146	–178	–214	–258	–335	–445	–585	2	4	5	7	13	19
				–54	–79	–104	–144	–172	–210	–254	–310	–400	–525	–690						
–27+Δ	0		–43	–63	–92	–122	–170	–202	–248	–300	–365	–470	–620	–800	3	4	6	7	15	23
				–65	–100	–134	–190	–228	–280	–340	–415	–535	–700	–900						
				–68	–108	–146	–210	–252	–310	–380	–465	–600	–780	–1000						
–31+Δ	0		–50	–77	–122	–166	–236	–284	–350	–425	–520	–670	–880	–1150	3	4	6	9	17	26
				–80	–130	–180	–258	–310	–385	–470	–575	–740	–960	–1250						
				–84	–140	–196	–284	–340	–425	–520	–640	–820	–1050	–1350						
–34+Δ	0		–56	–94	–158	–218	–315	–385	–475	–580	–710	–920	–1200	–1550	4	4	7	9	20	29
				–98	–170	–240	–350	–425	–525	–650	–790	–1000	–1300	–1700						
–37+Δ	0		–62	–108	–190	–268	–390	–475	–590	–730	–900	–1150	–1500	–1900	4	5	7	11	21	32
				–114	–208	–294	–435	–530	–660	–820	–1000	–1300	–1650	–2100						
–40+Δ	0		–68	–126	–232	–330	–490	–595	–740	–920	–110	–1450	–1850	–2400	5	5	7	13	23	34
				–132	–252	–360	–540	–660	–820	–1000	–1250	–1600	–2100	–2600						
–44			–78	–150	–280	–400	–600													
				–155	–310	–450	–660													
–50			–88	–175	–340	–500	–740													
				–185	–380	–560	–840													
–56			–100	–210	–430	–620	–940													
				–220	–470	–680	–1050													
–66			–120	–250	–520	–780	–1150													
				–260	–580	–840	–1300													
–78			–140	–300	–640	–960	–1450													
				–330	–720	–1050	–1600													
–92			–170	–370	–820	–1200	–1850													
				–400	–920	–1350	–2000													
–110			–195	–440	–1000	–1500	–2300													
				–460	–1100	–1650	–2500													
–135			–240	–550	–1250	–1900	–2900													
				–580	–1400	–2100	–3200													

3) (Continued) Example:
K7 in the range of 18 – 30 mm: Δ = 8 μm, therefore
ES = -2 + 8 = + 6 μm
S6 in the range of 18 – 30 mm: Δ = 4 μm, therefore
ES = -35 + 4 = -31 μm

4) Exception for tolerance M6 in the range of 250 – 315 mm, ES = -9 μm (instead of -11 μm).

5) Fundamental deviation for tolerance N for tolerance grades over IT8 shall not be used for basic dimensions equal to 1 mm or less.

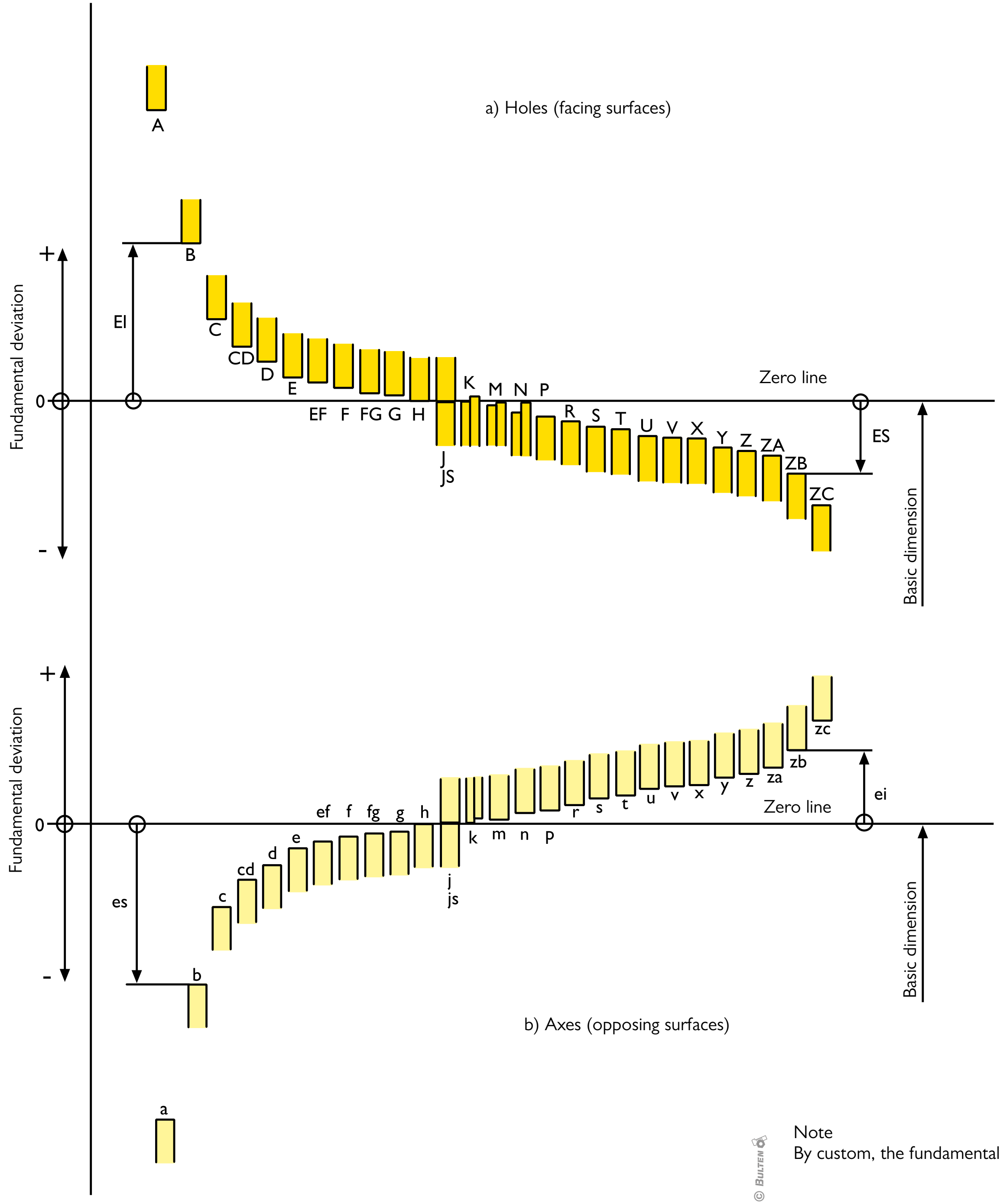
TABLE 12.21 TOLERANCE RANGES OF AXES (FROM SS-ISO 4759-1)

Nominal dimension		Tolerance range								
over	up to	h13	h14	h15	h16	h17	js14	js15	js16	js17
	3	0 –0.14	0 –0.25	0 –0.40	0 –0.60	0 –1.00	±0.125	±0.20	±0.30	±0.50
3	6	0 –0.18	0 –0.30	0 –0.48	0 –0.75	0 –1.20	±0.15	±0.24	±0.375	±0.60
6	10	0 –0.22	0 –0.36	0 –0.58	0 –0.90	0 –1.50	±0.18	±0.29	±0.45	±0.75
10	18	0 –0.27	0 –0.43	0 –0.70	0 –1.10	0 –1.80	±0.215	±0.35	±0.55	±0.90
18	30	0 –0.33	0 –0.52	0 –0.84	0 –1.30	0 –2.10	±0.26	±0.42	±0.65	±1.05
30	50	0 –0.39	0 –0.62	0 –1.00	0 –1.60	0 –2.50	±0.31	±0.50	±0.80	±1.25
50	80	0 –0.46	0 –0.74	0 –1.20	0 –1.90	0 –3.00	±0.37	±0.60	±0.95	±1.50
80	120	0 –0.54	0 –0.87	0 –1.40	0 –2.20	0 –3.50	±0.435	±0.70	±1.10	±1.75
120	180	0 –0.63	0 –1.00	0 –1.60	0 –2.50	0 –4.00	±0.50	±0.80	±1.25	±2.00
180	250	0 –0.72	0 –1.15	0 –1.85	0 –2.90	0 –4.60	±0.575	±0.925	±1.45	±2.30
250	315	0 –0.81	0 –1.30	0 –2.10	0 –3.20	0 –5.20	±0.65	±1.05	±1.60	±2.60
315	400	0 –0.89	0 –1.40	0 –2.30	0 –3.60	0 –5.70	± 0.70	±1.15	±1.80	±2.85
400	500	0 –0.97	0 –1.55	0 –2.50	0 –4.00	0 –6.30	± 0.775	±1.25	±2.00	±3.15

TABLE 12.22 TOLERANCE RANGES OF HOLES (FROM SS-ISO 4759-1)

Nominal dim. Over	up to	Tolerance range C13	C14	D9	D10	D11	D12	EF8	E11	E12	H14	H15	Js9	K9
	3	+0.20 +0.06	+0.31 +0.06	+0.045 +0.020	+0.060 +0.020	+0.080 +0.020	+0.12 +0.02	+0.024 +0.010	+0.074 +0.014	+0.100 +0.014	+0.25 0	+0.40 0	±0.0125	0 –0.025
3	6	+0.24 +0.06	+0.37 +0.07	+0.060 +0.030	+0.078 +0.030	+0.115 +0.030	+0.15 +0.03	+0.028 +0.014	+0.095 +0.020	+0.140 +0.020	+0.30 0	+0.48 0	±0.015	0 –0.030
6	10					+0.130 +0.040	+0.19 +0.04	+0.040 +0.018	+0.115 +0.025	+0.175 +0.025	+0.36 0	+0.58 0	±0.018	0 –0.036
10	18						+0.2 +0.05		+0.142 +0.032	+0.212 +0.032	+0.43 0	+0.70 0		
18	30						+0.275 +0.065				+0.52 0	+0.84		
30	50						+0.33 +0.08				+0.62 0	+1.00 0		
50	80						+0.40 +0.10				+0.74 0	+1.20 0		
80	120						+0.47 +0.12				+0.87 0	+1.40 0		
120	180										+1.00 0	+1.60 0		
180	250										+1.15 0	+1.85 0		
250	315										+1.30 0	+2.10 0		
315	400										+1.40 0	+2.30 0		
400	500										+1.55 0	+2.50 0		

TABLE 12.23 TOLERANCES – SCHEMATIC ILLUSTRATION OF FUNDAMENTAL DEVIATIONS



Note
By custom, the fundamental deviation is the deviation limit closest to the zero line

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