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Costruzione di Motori per Aeromobili
Bolted connection - Chapter 1

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Chapters

1 Prestressed single bolt connections

2 Refinements and special problems

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Section 1 -Bolted connections: scope

This section gives the future engineer some preliminary warning never to be forgotten.

- manufacturing and assembly processes, technologies, materials and lubricants can influence the results of bolting applications; there is more to know than is covered in these slides
- a good engineer will not just consider some design formulas, but all the factors which may influence safety, even if not under his own initial responsibility: this is the philosophy of Machine Design
- bolts treated in these slides are meant for high-performance coupling, specially for machinery parts, where - due to variable loads and/or vibrations – fatigue is always an important factor;
- the german VDI standars are a prominent coherent set of rules available on this matter
- bolting trusses, frames and in general civil engineering steel constructions are under other standards, such as (examples):

Specifications for Joints using High-Strength Bolts, RCSC, www.boltcouncil.org, USA, 2002 on...

Eurocode 3 : Design of steel structures, Part 1.8 Design of Joints, CEN, Brussels, 1993 on...

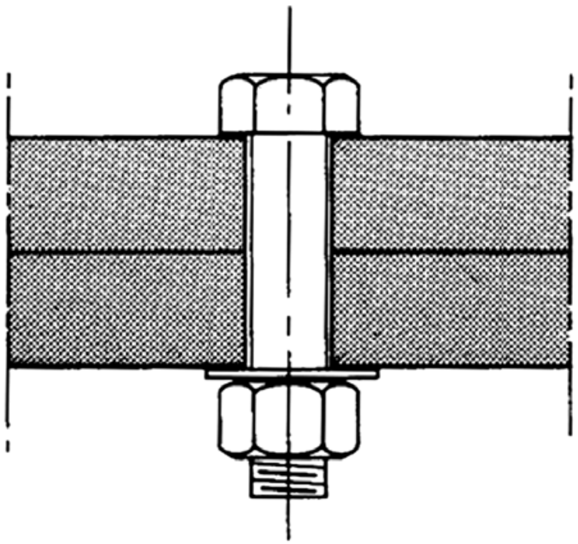
1 - Bolted connections: scope (1/9)

A bolted joint is a detachable connection between two or more parts by means of one or more bolts.

It is intended to transmit forces and moments between the joined parts in a clearly defined position relative to one another.

The bolts are to be designed in such a way that they withstand the working loads which occur and the function of the joint produced can be fulfilled.

G. L. Kulak, J. W. Fisher, J. H. A. Struik, Guide to Design Criteria for Bolted and Riveted Joints, Second Edition, AMERICAN INSTITUTE OF STEEL CONSTRUCTION, Inc., 2001



1 - Bolted connections: scope (2 / 9)

Caveat

Sections 2, 3 and 4 are meant to provide some context ideas about manufacturing, thread types in use, bolt and nut types normally employed.

The would-be engineer is urged to recognize that **much more** should be known in order to appreciate how much manufacturing and assembly processes, technologies, materials and lubricants can influence the results of bolting applications, which are seen in these slides from a limited (although very essential) point of view: the loading and the strength of one-bolt joint.

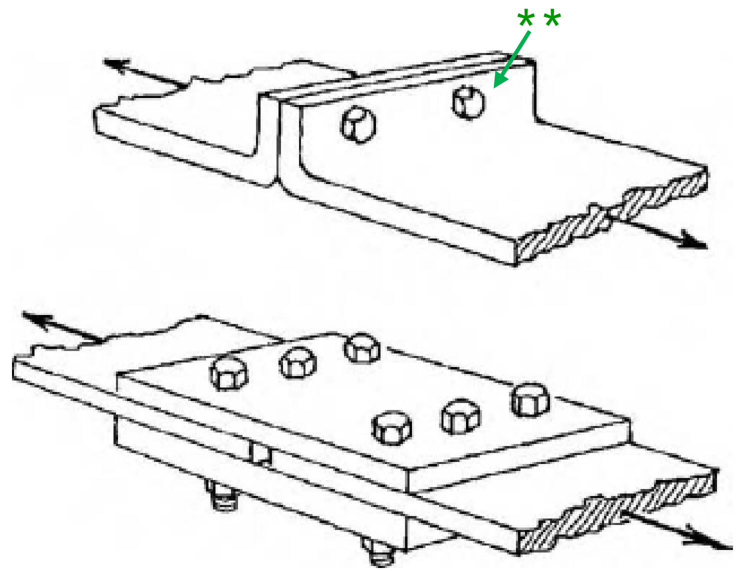
It will be wise to make sure that the limitations of this chapter are recognised and understood. A reference textbook like J. H. Bickford's (see next slide) is a valuable source of information and data, being the result of a lifelong experience of this author.

Bolted joints can be classified in two main types, depending on the direction of the external loads or forces acting on the joint.

When the line of action of the forces on the joint is parallel (or almost parallel) to the axis of the bolt, the joint is loaded in tension and is called a tension or tensile joint.

If the line of action of the load is perpendicular (or mostly perpendicular) to the axis of the bolt, the joint is loaded in shear and is called a shear joint.

The two types differ in the way they respond to loads, the ways in which they fail, the ways in which they are assembled.



From: J. H. Bickford, Introduction to the Design and Behavior of Bolted Joints, Non-Gasketed Joints, CRC Press 2008

** this is a jont of dubious morality: bending of the flanges will introduce bending of the bolts, which is really bad in fatigue

We shall deal here only with "prestressed bolt connections", i.e., these connections are that are loaded with a high axial force (mounting prestressing) during assembly.

This prestressing provides the necessary force bond of contact surfaces of the connected materials and is necessary to insure a correct load sharing between bolt and part during the application of variable external loads.

Pre-stressed bolt connections form the majority of bolt and threaded connections used in practice.

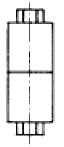
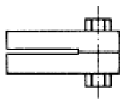
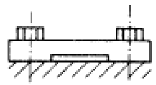
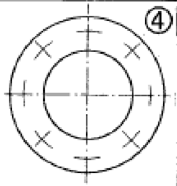
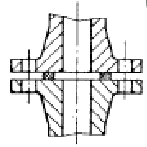
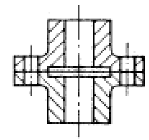
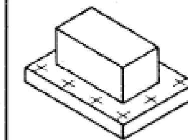
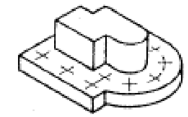
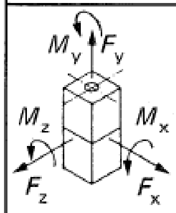
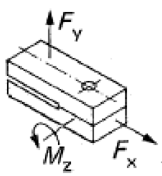
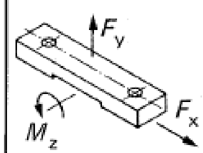
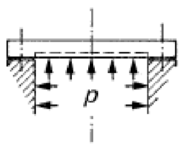
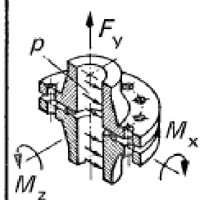
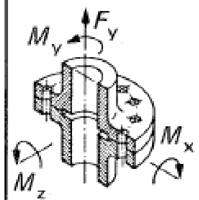
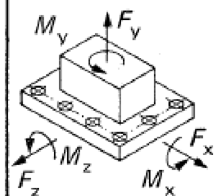
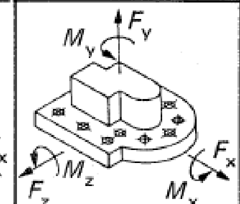
Our main reference will be to a specific standard:
VDI Richtlinien - VDI 2230 Blatt 1 (Part 1) - 2003
Systematische Berechnung hochbeanspruchter Schrauben-verbindungen: Zylindrische Einschraubenverbindungen / Systematic calculation of high duty bolted joints: Joints with one cylindrical bolt
(edition 2015 is available, but must be purchased, not available as free download!)

1 - Bolted connections: scope (5/9)

VDI Richtlinien - VDI 2230 Blatt 1 (Part 1) deals with the calculation of the single-bolted joint, which is the object of this chapter. It is based on the elastic behaviour of the joint in the immediate surroundings of the bolt axis. During assembly and in the service case, this region has a considerable effect on the deformation and thus on the loading of the bolt.

VDI Richtlinien - VDI 2230 Blatt 2 (Part 2) deals with joints coupled by many bolts (multi-bolted joints, see next slide); in this case the problem is the identification of the most loaded bolt and related loads, either through beam or plate theory, or through Finite Elements; this being done, the calculation then proceeds much in the way of Part 1 for each bolt or the most stressed one.

VDI Richtlinien - VDI 2230 Blatt 3 (Part 3) provides detailed recommendations on bolt assembly problems and techniques, and related refinements in calculations.

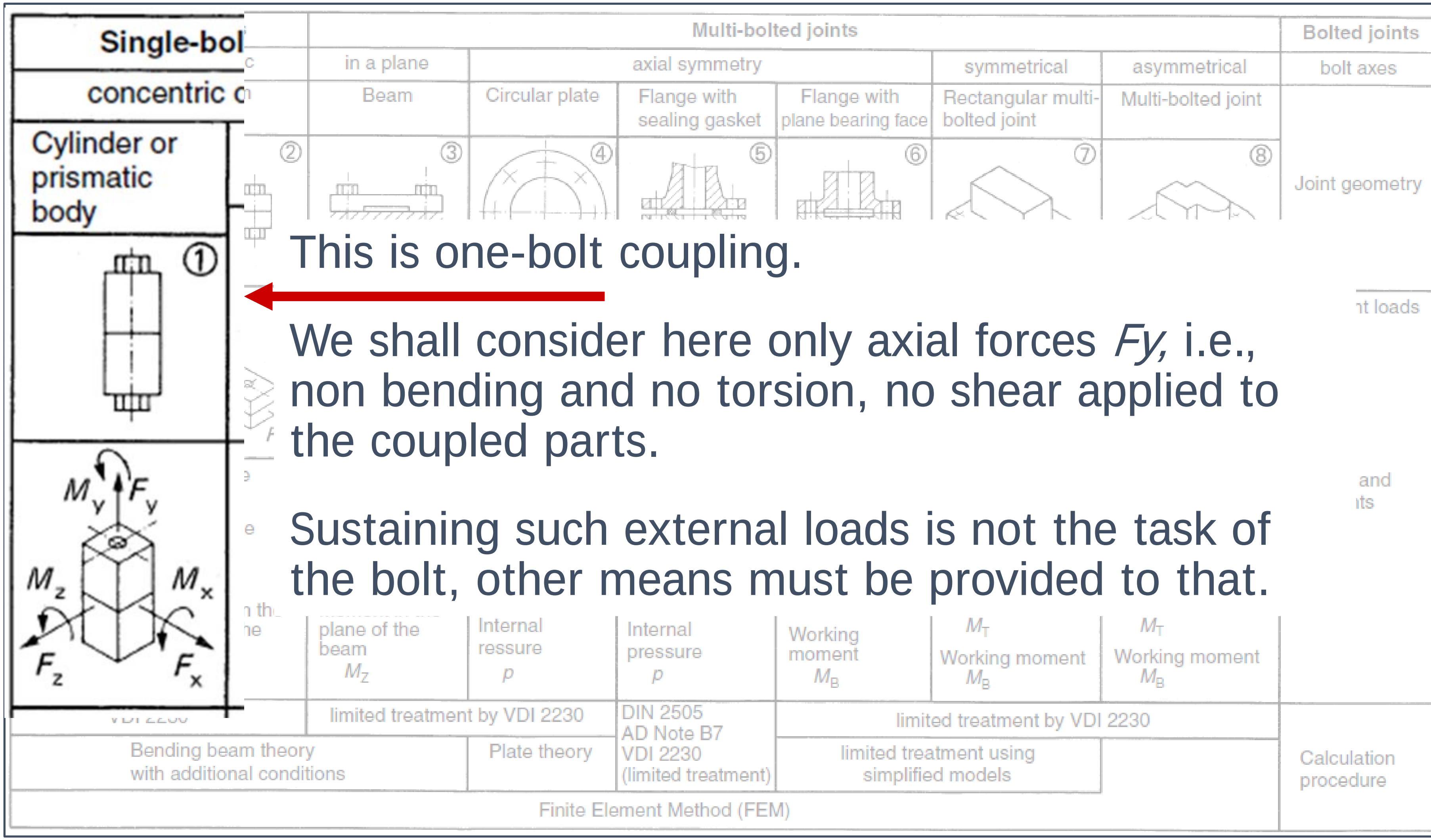
Single-bolted joints		Multi-bolted joints						Bolted joints
concentric or eccentric		in a plane	axial symmetry			symmetrical	asymmetrical	bolt axes
Cylinder or prismatic body	Beam	Beam	Circular plate	Flange with sealing gasket	Flange with plane bearing face	Rectangular multi-bolted joint	Multi-bolted joint	
① 	② 	③ 	④ 	⑤ 	⑥ 	⑦ 	⑧ 	Joint geometry
								Relevant loads
Axial force F_A Transverse force F_Q Working moment M_B	Axial force F_A Transverse force F_Q Moment in the plane of the beam M_Z	Axial force F_A Transverse force F_Q Moment in the plane of the beam M_Z	Internal ressure p	Axial force F_A (Pipe force) Working moment M_B Internal pressure p	Axial force F_A Torsional moment M_T Working moment M_B	Axial force F_A Transverse force F_Q Torsional moment M_T Working moment M_B	Axial force F_A Transverse force F_Q Torsional moment M_T Working moment M_B	Forces and moments
VDI 2230		limited treatment by VDI 2230		DIN 2505 AD Note B7 VDI 2230 (limited treatment)	limited treatment by VDI 2230			Calculation procedure
Bending beam theory with additional conditions			Plate theory		limited treatment using simplified models			
Finite Element Method (FEM)								

Source: VDI Richtlinien - VDI 2230 Blatt 1 / Part 1 – 2003

Adjournements 2018:

VDI 2230Blatt1 - Systematic calculation of highly stressed bolted joints - Joints with one cylindrical bolt. Standard published on 1.11.2015

VDI 2230Blatt2 - Systematic calculation of highly stressed bolted joints - Multi bolted joints. Standard published on 1.12.2014

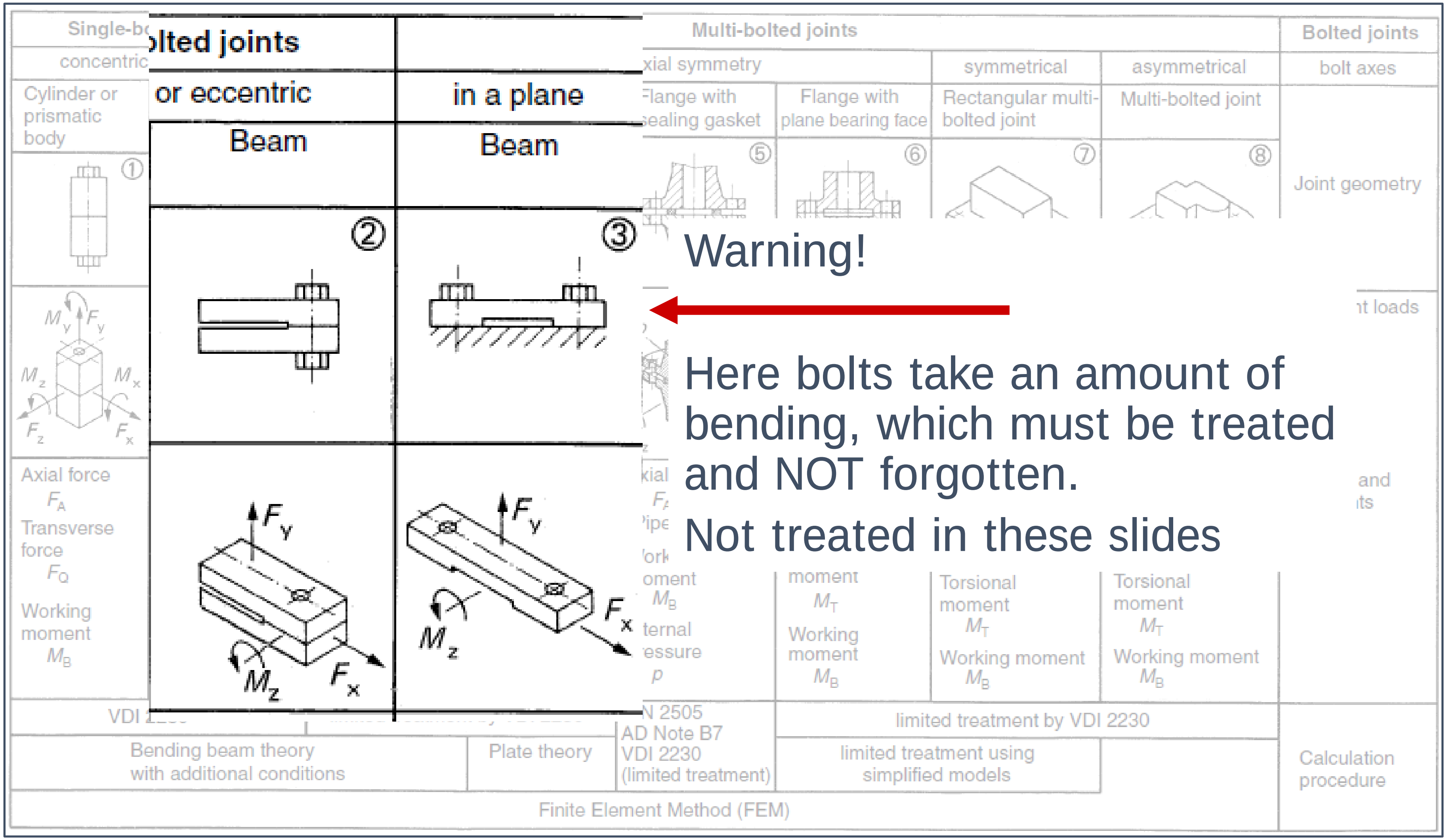


This is one-bolt coupling.

We shall consider here only axial forces *Fy*, i.e., non bending and no torsion, no shear applied to the coupled parts.

Sustaining such external loads is not the task of the bolt, other means must be provided to that.

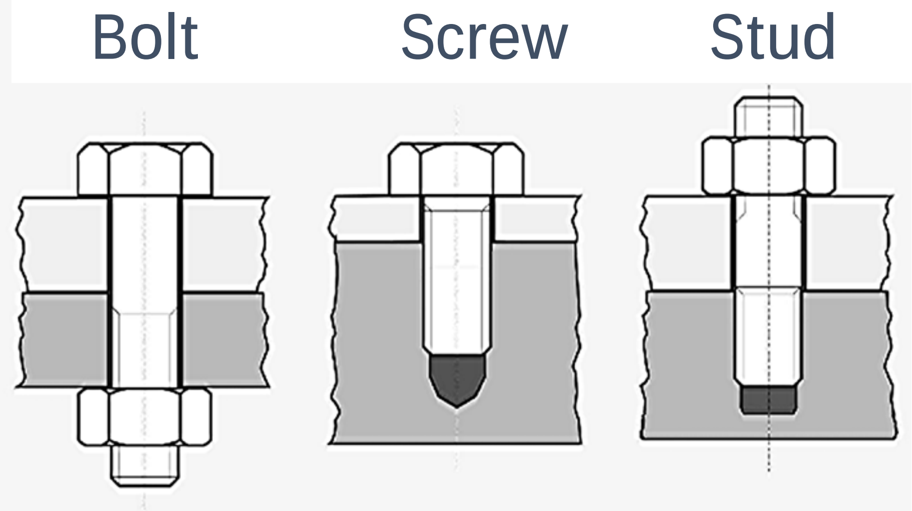
Source: VDI Richtlinien - VDI 2230 Blatt 1 / Part 1 – 2003



Source: VDI Richtlinien - VDI 2230 Blatt 1 / Part 1 – 2003

The definition of screw is an externally threaded headed fastener (the recessed drive socket in a setscrew is considered a head), which is tightened by applying torque to the head, causing it to be threaded into the material it will hold.

A bolt on the other hand is an externally threaded headed fastener, which is used in conjunction with a nut.



To obtain reliable and repeatable fastener torque the bolt / nut combination should always be tightened by holding the bolt head stationary and turning the nut.

Nearly any bolt with a common head can be used as a screw by tightening it by the head into a tapped hole.

However, examples of bolts that cannot be used as screws are:

- carriage bolts and timber bolts

- plow bolts

Sections 2, 3, 4 - Manufacturing, geometry, materials

The purpose of Sections 2, 3, 4 is to give a general information about the manufacturing processes, the geometrical factors and the material choices that have a bearing on the performance of bolts.

It will be made clear later that manufacturing processes influence fatigue strength, due to metallurgical and residual stress factors. However, sufficient preliminary knowledge should have been already gained in general chapter on fatigue.

Section 2 gives an overview of manufacturing technologies.

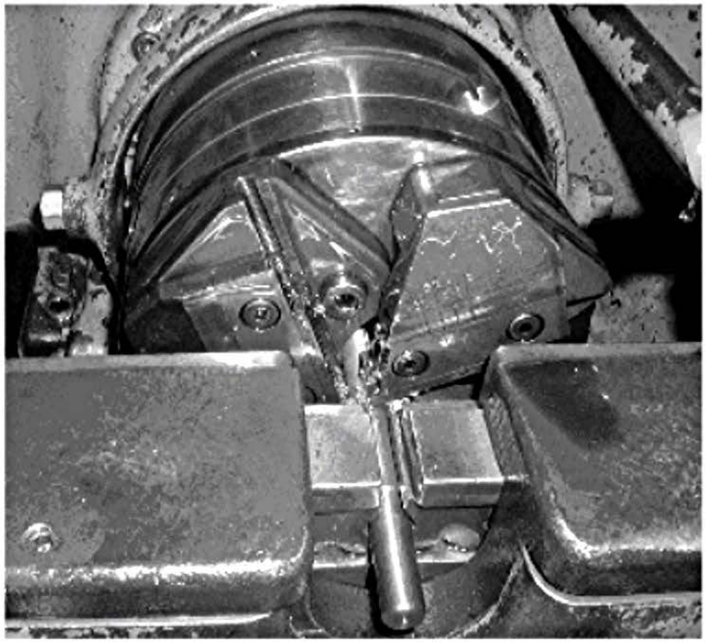
Section 3 reminds the thread geometry fundamentals.

Section 4 gives and overview on materials available challenges in special situations like corrosion, high temperature etc., and on carbon steel and stainless steel properties. Then it shows how bolts are marked according to material class both in the case of carbon steels and stainless steels. The basic strength definitions for bolts and nuts and the main strength tests are finally provided.

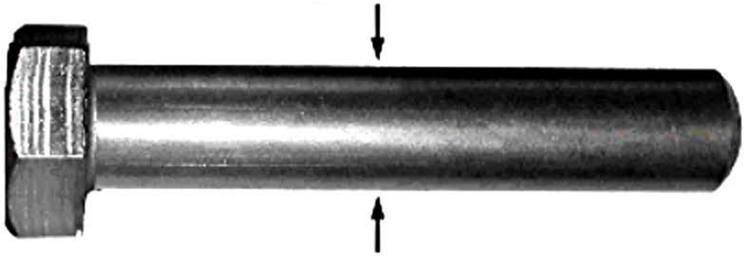
Machining is only used on unique designs or with screws either too small or too large to be made in any other way. The machining process is exact, but too time consuming, wasteful, and expensive.

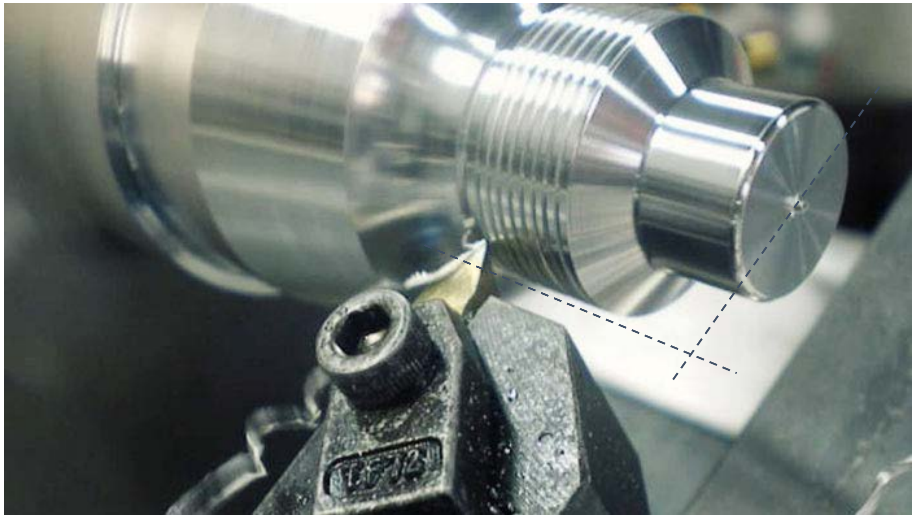
A one-hundred, ninty-three pound bolt, one of 16 used to join sections of the generator shaft of a 75,000 kW generator - Grand Coulee Dam, 1942 (US).

Cut thread on a bolt



Major diameter





Cut thread on a shaft

http://www.fastenal.com/web/services.ex?action=engineering&article=ScrewThreadDesign

Machining is economically advantageous for small quantities and non-standard geometries.

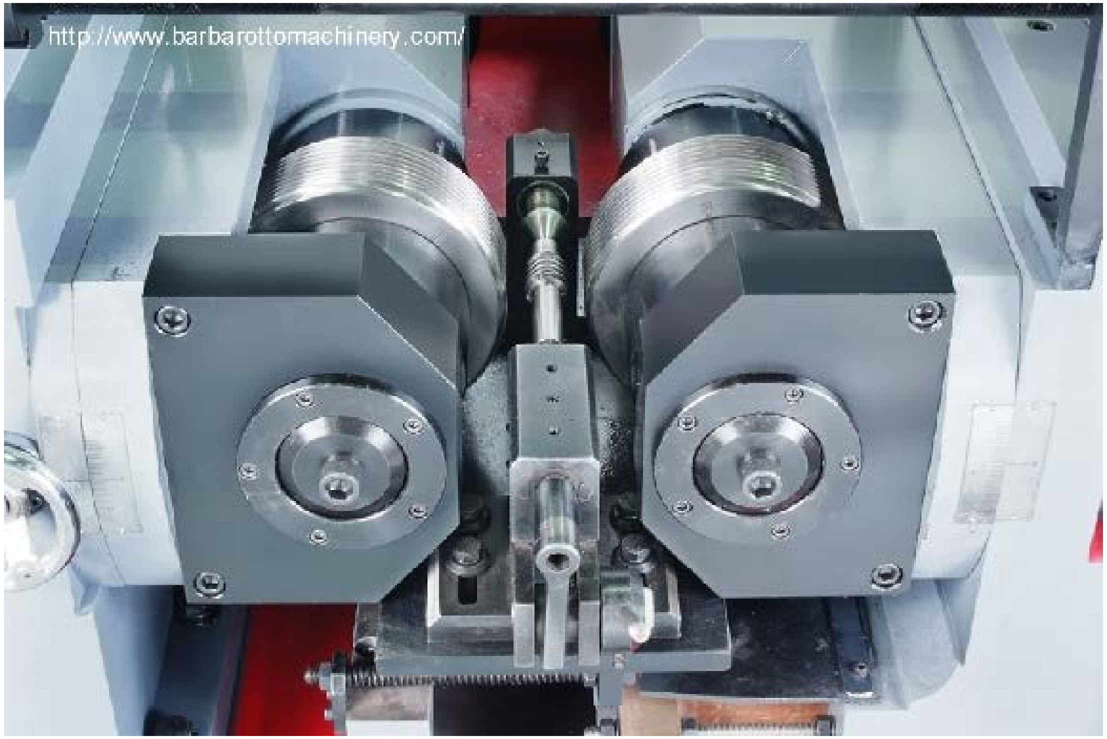
However, defects (e.g., microcracks and grain boundaries) which form at the surface during this process serve as preferred sites for fatigue crack initiation.

Thus, machined threads often exhibit only limited fatigue properties.

2 - Bolt and thread manufacturing (3/10)

Rolling is advantageous for large throughputs.

During the process, grains are being aligned in the rolling direction (mechanical fibering) and beneficial compressive residual stresses are introduced into the material.



Consequently, both the initiation and propagation of fatigue cracks are hindered.

Fatigue properties are markedly enhanced by rolling the threads after, instead of before, heat treatment (see fatigue at the end of this chapter).

2 - Bolt and thread manufacturing (4/10)

https://ceprofs.civil.tamu.edu/llowery/cven446/photos/bolt%20photos/bolts.htm

The bulk of all screws are mass manufactured using the thread rolling method. A wire or a bar is fed to the machine that automatically cuts them at a designated length and forms the head (see next slide).

1 - Take a blank, then form a head on one end, then form the end for the threads.

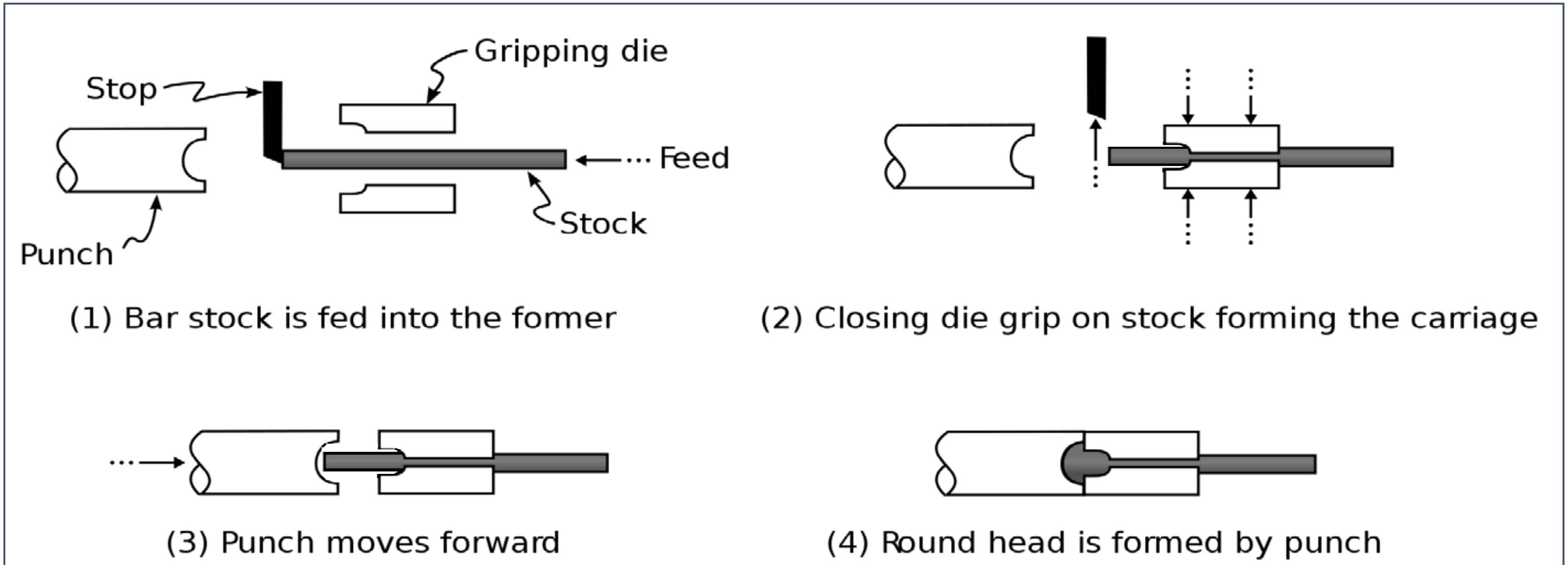
2 - Bolt and thread manufacturing (5/10)

2 - Then shear a hex off of the round head.

3 - Then roll the threads into the shank.

However, larger sizes are manufactured starting from forged blanks.

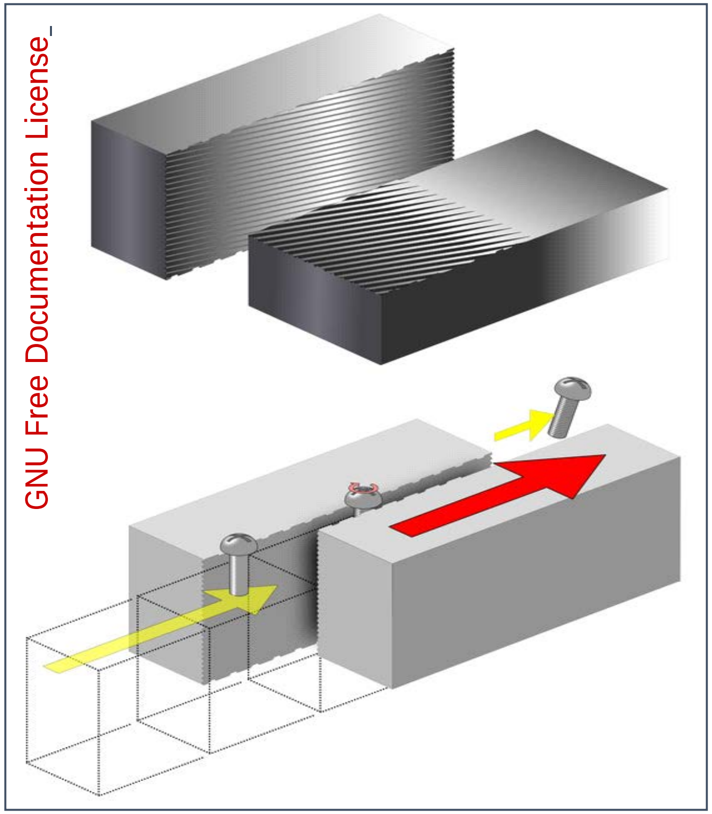
D.H. Herring, R.D. Sisson, Testing of Heat-Treated Fasteners, Fastener Technology International - February 2010

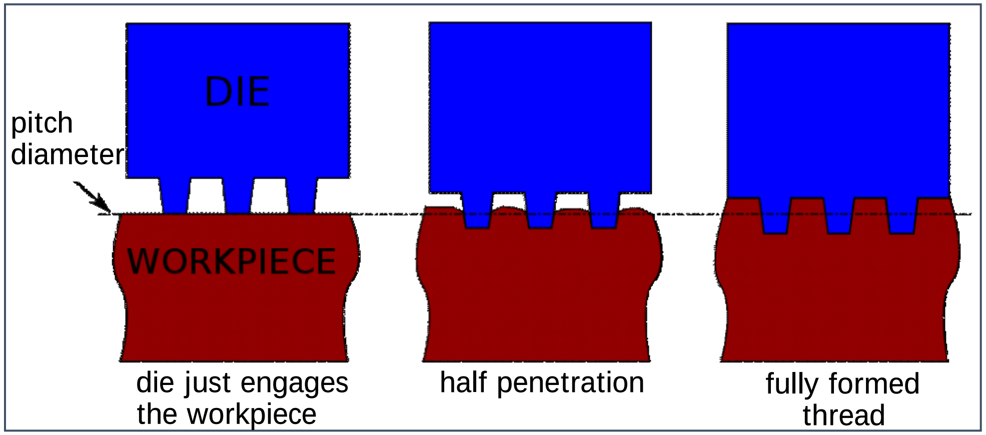


http://www.enotes.com/topic/Bolt_manufacturing_process

The bar stock is fed into a machine, called a "former". The bar stock is stopped, and then gripped by a gripping die. The gripping die is shaped in such a way that it forms a carriage along the neck of the stock when its full gripping force is applied. An indented punch, the shape of a rounded head, moves forward, colliding with the stock and causing the metal to flow at right angles to the ram force provided by the punch, increasing the diameter of the workpiece, and reducing its overall length, thus forming a "rounded" head.
The work billet is then sheared to the desired length.

The blank is then threaded by rolling (as shown on the right) using one of three techniques:



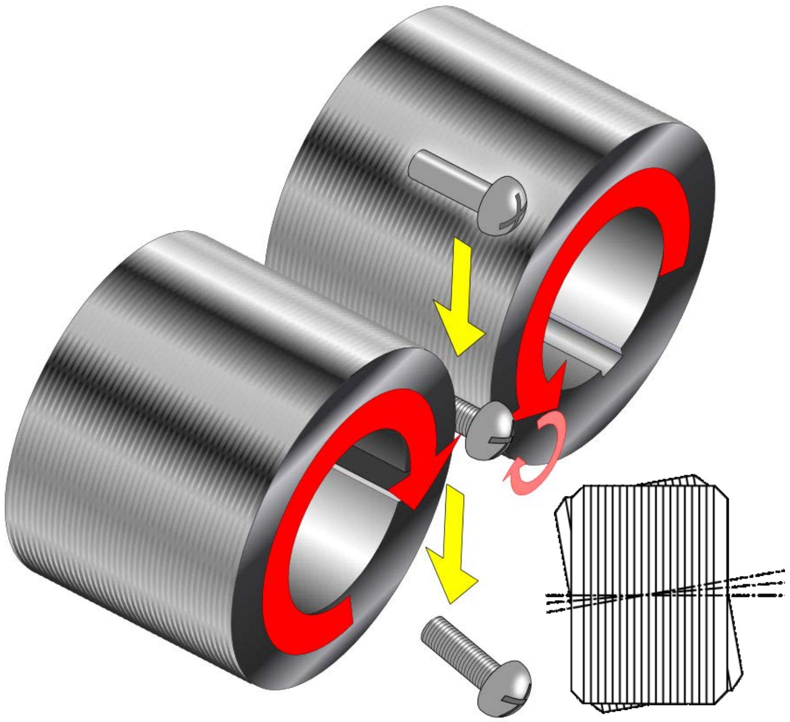


Thread forming and rolling concept - GNU Free Documentation License

1 - In the reciprocating die, two flat dies are used to roll the screw thread.

One die is stationary, while the other moves in a reciprocating manner, and the screw blank is rolled between the two.

2 - In the cylindrical die rolling the screw blank is rolled between two (centerless, drawing on the right) or three round dies in order to create the finished thread.



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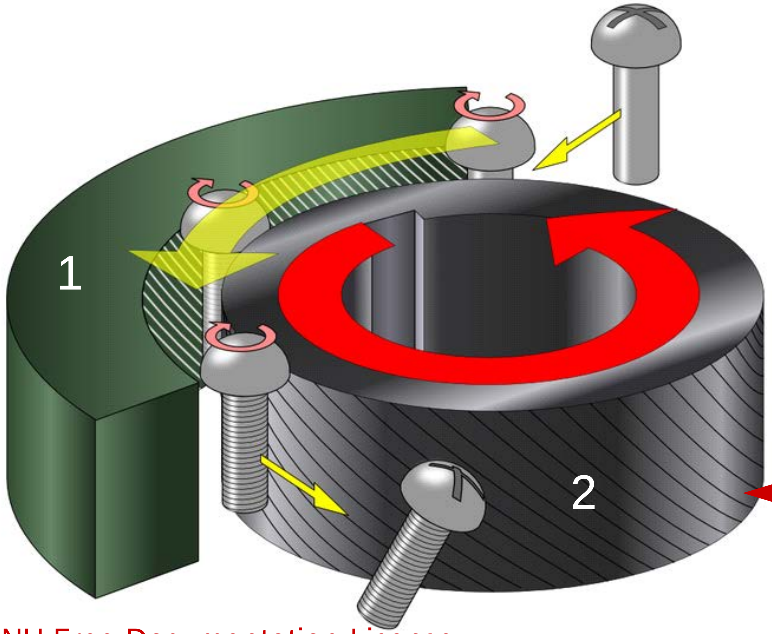

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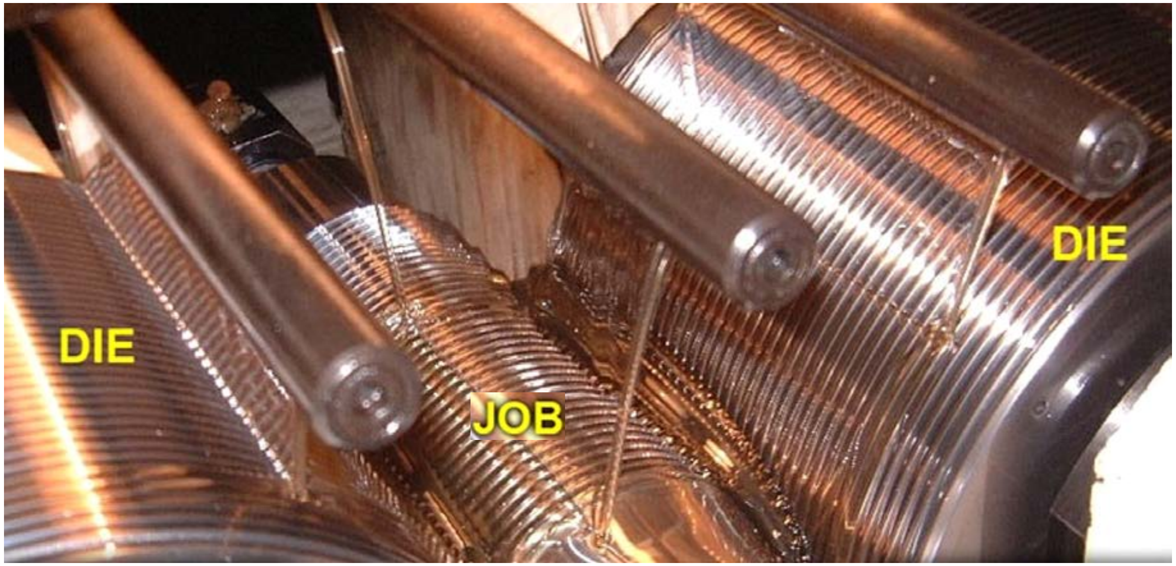
3 - The planetary rotary die using one stationary segment die (1) and one rotating circular die (2).

All three rolling methods create higher strength screws than the machine-cut variety. This is because the thread is not literally cut into the blank during the thread-rolling process, rather it is impressed into the blank.



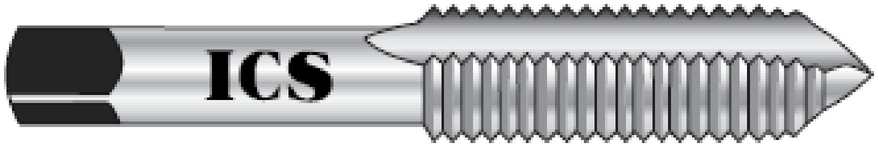
Thus, no metal material is lost, and weakness in the metal is avoided (residual stresses and metal structure are more favourable, in particular to fatigue).

On average, the cold heading machine produces 100 to 550 screw blanks per minute.

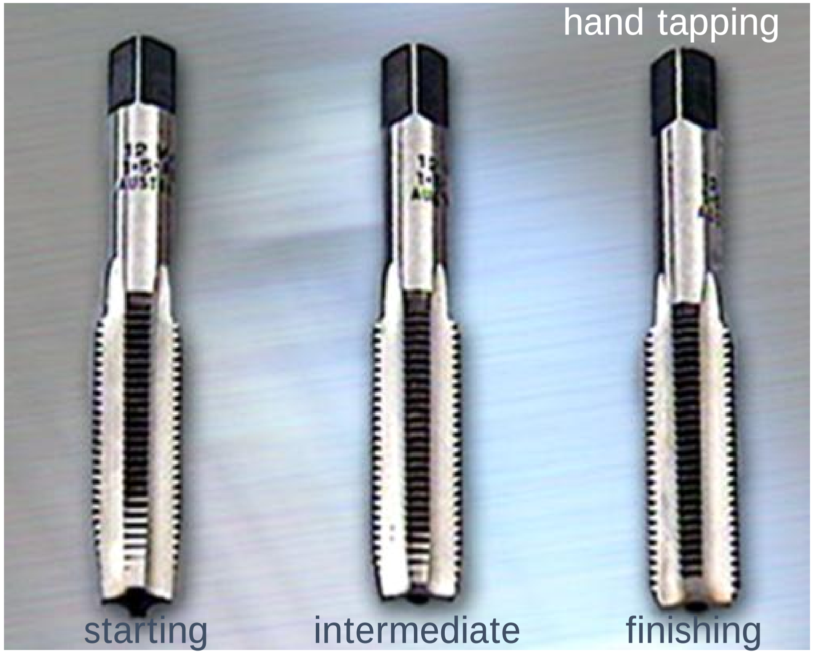
The more productive of the thread-rolling techniques is by far the planetary rotary die, which creates screws at a speed up to 2,000 parts per minute, however in the case of smaller bolts up to 1/8 inch, 3 mm; down to 100÷400 parts per minute for bolts up to ½ inch.

Holes are threaded by hand or machine tapping.

First, hole is drilled with a suitable diameter and suitable depth, next the tap is inserted and turned to cut the threads.



machine tapping



A starting tap has a well tapered end, which is why it is sometimes called a "taper" tap. This allows the tap gradually to cut deeper threads as it passes through the job. It can be used to cut a thread in work that has a thin enough section to allow the tap to pass through it. It is also used to perform the first cut in a blind hole.

An intermediate, or second tap is used for the second cut in a blind hole. It has fewer tapers than a tapered tap, which allows the threads to be cut near the bottom of the hole.

A finishing, bottoming or plug tap is designed to cut the final thread into a blind hole. It has almost no taper, so the threads it cuts extend to the bottom of the hole.

Sections 2, 3, 4 - Manufacturing, geometry, materials

The purpose of Sections 2, 3, 4 is to give a general information about the manufacturing processes, the geometrical factors and the material choices that have a bearing on the performance of bolts.

It will be made clear later that manufacturing processes influence fatigue strength, due to metallurgical and residual stress factors. However, sufficient preliminary knowledge should have been already gained in general chapter on fatigue.

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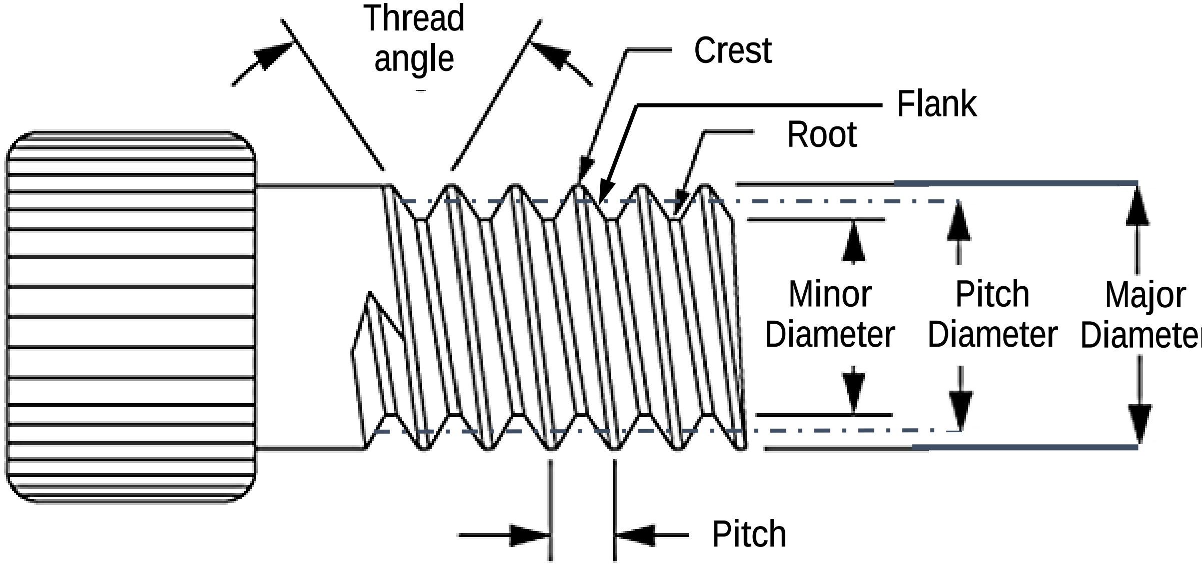
A helical thread used in screws, or bolts, is characterized by a helical ridge, known as “external thread”, wrapped around a cylinder.



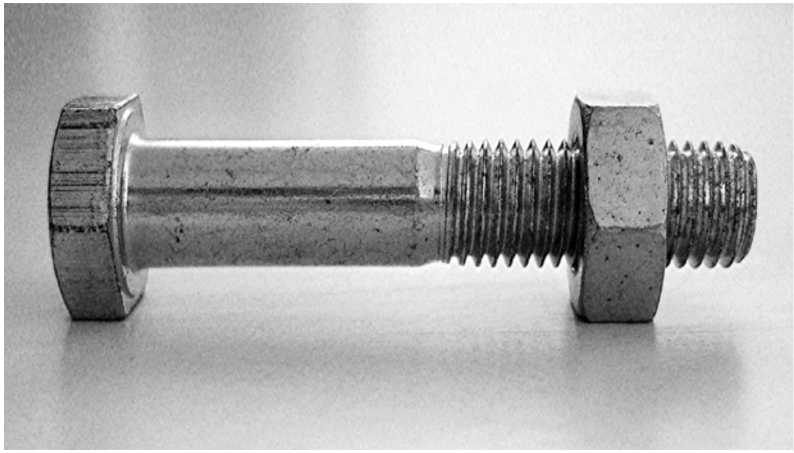
Major diameter is the largest diameter of the thread. For a male thread, this means "outside diameter", but in careful usage the better term is "major diameter", since the underlying physical property being referred to is independent of the male - female context. On a female thread, the major diameter is not on the "outside". The terms "inside" and "outside" invite confusion, whereas the terms "major" and "minor" are always unambiguous.

Minor diameter is the smallest diameter of the thread.

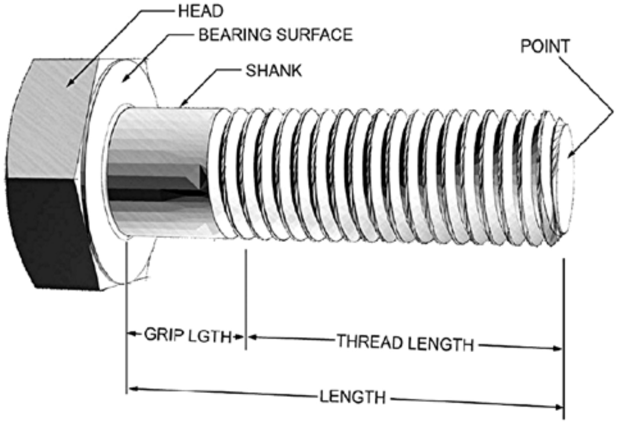
Pitch diameter is the diameter at which each pitch is equally divided between the mating male and female threads (corrected for allowances).

DESIGNATION EXAMPLES OF VARIOUS THREADS			
ORIGIN	COARSE THREAD	FINE THREAD	EXTRA FINE THREAD
METRIC	M 8	M 8 X 1.0	M 8 X 0.75
U.S.A.	3/8 - 16 UNC	3/8 - 24 UNF	3/8 - 32 UNEF
BRITISH	1/2 BSW	1/2 BSF	NONE
ORIGIN	MINIATURE THREAD	PIPE TAPERED THREAD	PIPE PARALLEL THREAD
METRIC	M 1.0	M 12 X 1.5T (TAPER)	M12 X 1.5
U.S.A.	1.0 UNM	1/8 - 27 NPT	1/8 - 27 NPS
BRITISH	10 BA	1/2 BSPT	1/2 BSPF

source: MARYLAND METRICS http://mdmetric.com/tech/tict.htm

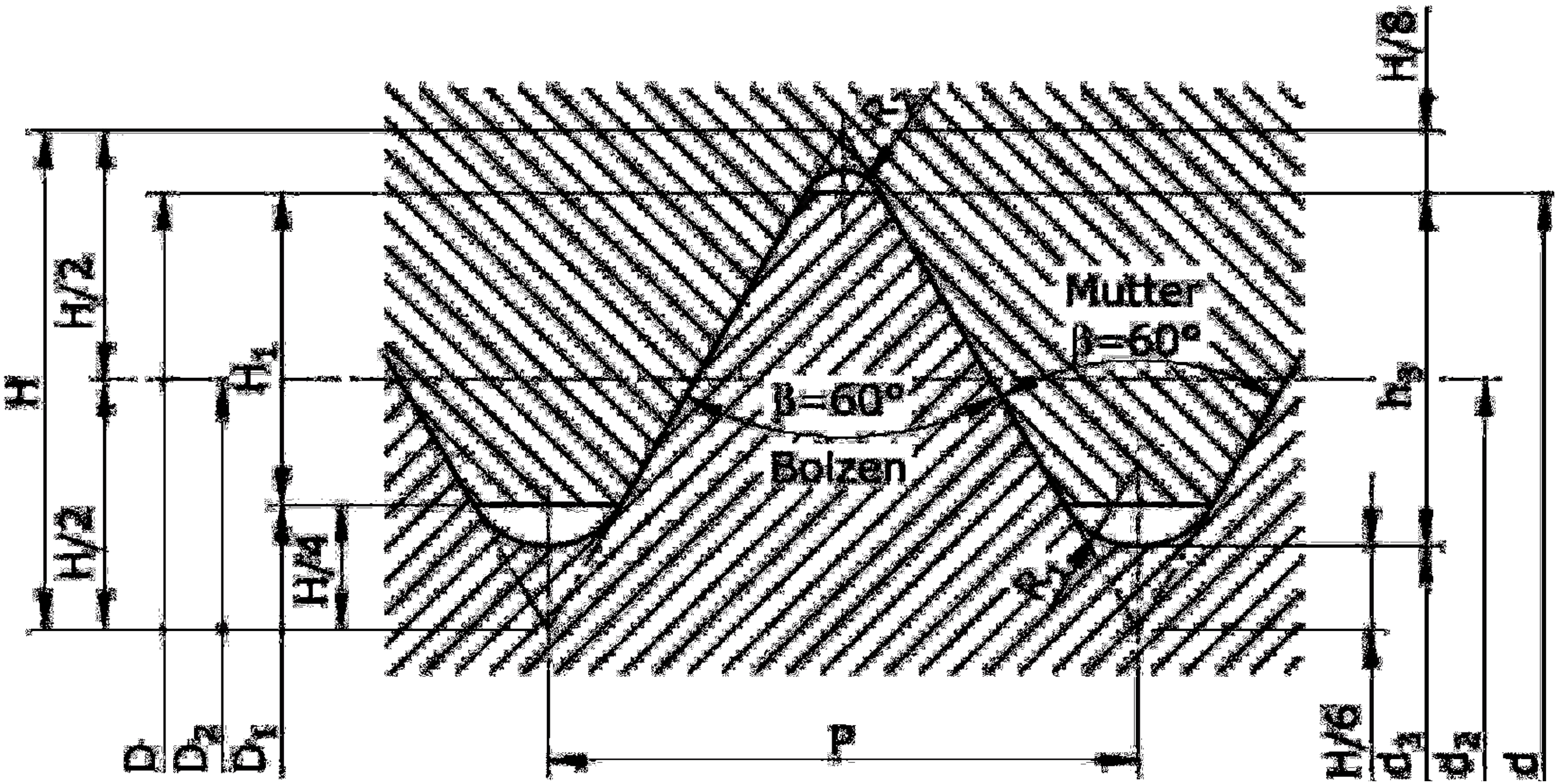


1. The metric thread designation uses pitch in place of the more familiar U.S.A. method of threads per inch. pitch is the distance in mm from any one point on a thread to a corresponding point on the next thread, when measured parallel to its axis.
2. Metric coarse thread does not have to have pitch specified. the absence of the pitch specification indicates that the thread will be from the coarse thread series.
3. The nominal diameter of metric pipe threads, both tapered and parallel is the same as the major diameter (od) of the thread. However, the U.S.A., British and Japanese nominal pipe thread diameters correspond to the approximate inside diameter (id) of the pipe or tube. This places the U.S.A., British and Japanese pipe threads among thread diameters which are larger than their nominal size.
4. Most metric countries in Europe and Asia use the British based inch system (ISO) for measuring non-hydraulic/pneumatic pipe and tubing threads.
5. In the British system there is no need to indicate the number of threads per inch. Each of the British thread series have only one number of threads per inch allocated within its series and nominal diameter.
6. The U.S.A. miniature thread series (UNM) is interchangeable with the corresponding sizes of ISO metric threads where the nominal diameters are equal.



7. A few additional hints to thread system identification are as follows: generally, bolt heads with numerical markings such as 8.8, 9.8, 10.9, 12.9, will probably be metric; and bolt heads with line/slash markings will probably be from the U.S.A. grade marking system.

In these slides reference will be made only to the Metric threads, screws, nuts and bolts.



Metric thread dimensions:

d : nominal (bolt crest) diameter $\quad$ d_3 : minor diameter

d_2 : pitch (mean) diameter (of the imaginary cylinder passing through the thread in a manner as to equalize the width of thread ridge and groove)

Bolt size	Pitch	Pitch diameter	Stress cross section	Cross section at minor diameter
	P	d_2	A_S	A_{d3}
	mm	mm	mm^2	mm^2
Metric standard (coarse) threads				
M 4	0,7	3,545	8,78	7,749
M 5	0,8	4,480	14,2	12,69
M 6	1	5,350	20,1	17,89
M 7	1	6,350	28,9	26,18
M 8	1,25	7,188	36,6	32,84
M 10	1,5	9,026	58,0	52,30
M 12	1,75	10,863	84,3	76,25
M 14	2	12,701	115	104,7
M 16	2	14,701	157	144,1
M 18	2,5	16,376	193	175,1
M 20	2,5	18,376	245	225,2
M 22	2,5	20,376	303	281,5
M 24	3	22,051	353	324,3
M 27	3	25,051	459	427,1
M 30	3,5	27,727	561	519,0
M 33	3,5	30,727	694	647,2
M 36	4	33,402	817	759,3
M 39	4	36,402	976	913,0

Nominal values for pitch, pitch diameter, stress cross section and cross section at minor diameter, for bolts with metric standard threads (pitch according to DIN 13-1 and -28).

Stress cross section A_S or "stress area" based on stress diameter d_S:

$$d_S = \frac{d_2 + d_3}{2}$$

according to DIN 13-28.

Source: VDI Richtlinien
VDI 2230 Blatt 1 / Part 1 – 2003

Bolt size	Pitch	Minor diameter	Reduced shank diameter	Reduced shank cross section
	P	d_3	$d_T = 0{,}9 \cdot d_3$	$A_T = \pi/4\ d_T^{\ 2}$
	mm	mm	mm	mm^2
Metric standard (coarse) threads				
M 4	0,7	3,141	2,83	6,28
M 5	0,8	4,019	3,62	10,3
M 6	1	4,773	4,30	14,5
M 7	1	5,773	5,20	21,2
M 8	1,25	6,466	5,82	26,6
M 10	1,5	8,160	7,34	42,4
M 12	1,75	9,853	8,87	61,8
M 14	2	11,546	10,4	84,8
M 16	2	13,546	12,2	117
M 18	2,5	14,933	13,4	142
M 20	2,5	16,933	15,2	182
M 22	2,5	18,933	17,0	228
M 24	3	20,319	18,3	263
M 27	3	23,319	21,0	346
M 30	3,5	25,706	23,1	420
M 33	3,5	28,706	25,8	524
M 36	4	31,093	28,0	615
M 39	4	34,093	30,7	739

Nominal values for pitch, minor diameter, reduced-shank diameter, reduced-shank cross section and for necked-down (or waisted or reduced shank) bolts with metric standard threads (pitch and minor diameter according to DIN 13-1, -5 to -8)

Source: VDI Richtlinien
VDI 2230 Blatt 1 / Part 1 – 2003

3 - Thread geometry fundamentals (6 / 9)

Bolt size	Pitch	Pitch diameter	Stress cross section	Cross section at minor diameter
	P	d_2	A_S	A_{d3}
	mm	mm	mm^2	mm^2
Metric fine threads				
M 8 x 1	1	7,350	39,2	36,03
M 9 x 1	1	8,350	51,0	47,45
M 10 x 1	1	9,350	64,5	60,45
M 10 x 1,25	1,25	9,188	61,2	56,29
M 12 x 1,25	1,25	11,188	92,1	86,03
M 12 x 1,5	1,5	11,026	88,1	81,07
M 14 x 1,5	1,5	13,026	125	116,1
M 16 x 1,5	1,5	15,026	167	157,5
M 18 x 1,5	1,5	17,026	216	205,1
M 18 x 2	2	16,701	204	189,8
M 20 x 1,5	1,5	19,026	272	259,0
M 22 x 1,5	1,5	21,026	333	319,2
M 24 x 1,5	1,5	23,026	401	385,7
M 24 x 2	2	22,701	384	364,6
M 27 x 1,5	1,5	26,026	514	497,2
M 27 x 2	2	25,701	496	473,2
M 30 x 1,5	1,5	29,026	642	622,8
M 30 x 2	2	28,701	621	596,0

The same for metric fine threads.

Source: VDI Richtlinien
VDI 2230 Blatt 1 / Part 1 – 2003

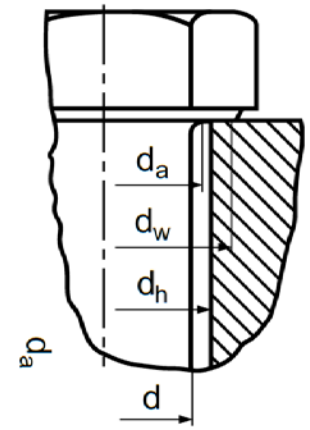
Bolt size	Pitch	Minor diameter	Reduced shank diameter	Reduced shank cross section
	P	d_3	$d_T = 0{,}9 \cdot d_3$	$A_T = \pi/4\ d_T{}^2$
	mm	mm	mm	mm^2
Metric fine threads				
M 8 x 1	1	6,773	6,10	29,2
M 9 x 1	1	7,773	7,00	38,4
M 10 x 1	1	8,773	7,90	49,0
M 10 x 1,25	1,25	8,466	7,62	45,6
M 12 x 1,25	1,25	10,466	9,42	69,7
M 12 x 1,5	1,5	10,160	9,14	65,7
M 14 x 1,5	1,5	12,160	10,94	94,1
M 16 x 1,5	1,5	14,160	12,74	128
M 18 x 1,5	1,5	16,160	14,54	166
M 18 x 2	2	15,546	13,99	154
M 20 x 1,5	1,5	18,160	16,34	210
M 22 x 1,5	1,5	20,160	18,14	259
M 24 x 1,5	1,5	22,160	19,94	312
M 24 x 2	2	21,546	19,39	295
M 27 x 1,5	1,5	25,160	22,64	403
M 27 x 2	2	24,546	22,09	383
M 30 x 1,5	1,5	28,160	25,34	504
M 30 x 2	2	27,546	24,79	483

The same for metric fine threads.

Source: VDI Richtlinien
VDI 2230 Blatt 1 / Part 1 – 2003

Nominal thread Ø d	Width across flats s max. (mm)	Ø of the bearing surface dw min. (mm)	Through hole (ISO 273) d_h (mm)	Bearing surface A_p (mm²)	Stressed cross-section A_s (mm²)	Surface pressure under the head [1] $\left[\frac{N}{mm^2}\right]$ 8.8	10.9	12.9
M 4	7	5,9	4,5	11,4	8,78	385	568	665
M 5	8	6,9	5,5	13,6	14,2	528	777	909
M 6	10	8,9	6,6	28	20,1	364	532	625
M 8	13	11,6	9	42,1	36,6	442	649	761
M10	16	14,63	11	73,1	58	405	594	695
M10	17	15,6	11	96,1	58	308	452	529
M12	18	16,63	13,5	74,1	84,3	580	853	999
M12	19	17,4	13,5	94,6	84,3	454	668	782
M14	21	19,64	15,5	114,3	115	517	759	888
M14	22	20,5	15,5	141,4	115	418	613	718
M16	24	22,5	17,5	157,1	157	515	756	885
M18	27	25,3	20	188,6	192	541	769	901
M20	30	28,2	22	244,4	245	532	761	888
M22	34	31,71	24	337,3	303	480	685	803
M22	32	30	24	254,5	303	637	908	1065
M24	36	33,6	26	355,8	353	528	750	880
M27	41	38	30	427,3	459	576	821	960
M30	46	42,7	33	576,7	561	520	740	865

Source: Technical Information – Bossard AG, CH 6301 Zu

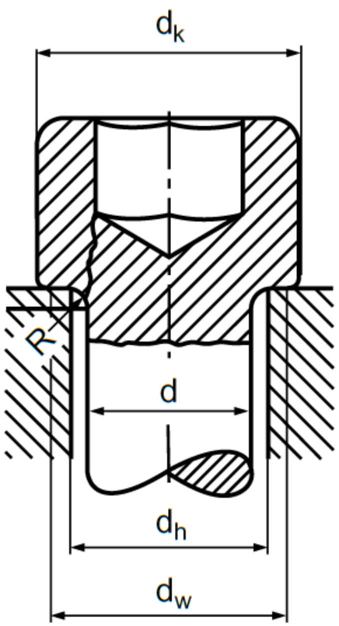


$d_h > d_a$

Surface pressure under the head of a hexagon screw – coarse threads

Source: Technical Information – Bossard AG, CH 6301 Zug, 2012.06

Nominal thread Ø d	Ø of head d_K (mm)	Ø of the bearing surface $d_{w\,min.}$ (mm)	Through hole (ISO 273) d_h (mm)	Bearing surface A_p (mm^2)	Stressed cross-section A_s (mm^2)	Surface pressure under the head [1] $\left[\frac{N}{mm^2}\right]$ 8.8	10.9	12.9
M 4	7	6,53	4,5	17,6	8,79	250	370	432
M 5	8,5	8,03	5,5	26,9	14,2	268	394	461
M 6	10	9,38	6,6	34,9	20,1	292	427	502
M 8	13	12,33	9	55,8	36,6	333	489	574
M10	16	15,33	11	89,5	58	331	485	567
M12	18	17,23	13,5	90	84,3	478	702	822
M14	21	20,17	15,5	130,8	115	452	663	776
M16	24	23,17	17,5	181,1	157	447	656	767
M18	27	25,87	20	211,5	192	482	686	804
M20	30	28,87	22	274,5	245	474	678	791
M22	33	31,81	24	342,3	303	473	675	792
M24	36	34,81	26	420,8	353	447	635	744
M27	40	38,61	30	464	459	530	756	884
M30	45	43,61	33	638,4	561	470	669	782



Surface pressure under the head of a socket head cap screw

1) The values shown in the table for surface pressures are for a 90% utilisation of the yield strength $R_{p0,2}$ of the screw and μ_G=0,12

The purpose of Sections 2, 3, 4 is to give a general information about the manufacturing processes, the geometrical factors and the material choices that have a bearing on the performance of bolts.

It will be made clear later that manufacturing processes influence fatigue strength, due to metallurgical and residual stress factors. However, sufficient preliminary knowledge should have been already gained in general chapter on fatigue.

Section 2 gives an overview of manufacturing technologies.

Section 3 reminds the thread geometry fundamentals.

Section 4 gives and overview on materials available challenges in special situations like corrosion, high temperature etc., and on carbon steel and stainless steel properties. Then it shows how bolts are marked according to material class both in the case of carbon steels and stainless steels. The basic strength definitions for bolts and nuts and the main strength tests are finally provided.

Carbon steels and corrosion-resistant materials - 1

Carbon steels, coated

Low carbon, medium carbon, and low-alloy steels can be made more resistant to atmospheric corrosion by coating or by plating them.

Stainless steels

Description by a combination of letters and figures, stamped on bolt head or nut; example:

Composition group: A, C, F

A2-70

Abbreviation of property class: 50, 70, 80 (1/10 of tensile strenght in MPa)

Abbreviation of chemical composition

A - Austenitic stainless steels

Most common of the stainless steels and more corrosion-resistant than the three other types listed below. Non-magnetic. Cannot be heat-treated but can be cold-worked. Good high & low temperature properties (chemical compositions: 1, 2, 3, 4, 5)

C - Martensitic stainless steels UTS 70–180 ksi

Heat treatable, magnetic. Can experience stress corrosion if not properly treated.

F - Ferritic stainless steels

Cannot be heat-treated or cold-worked. Magnetic.

Carbon steels and corrosion-resistant materials - 2

Hastelloy B

Highly corrosion-resistant nickel-molybdenum alloy with excellent resistance against reducing media (all concentrations of hydrochloric acid up to boiling point, sulphuric and phosphoric acid, alkaline solutions). Applications: components under strong chemical action, turbo superchargers for jet engines ... ...

Hastelloy C

Highly corrosion-resistant nickel-chrome-molybdenum alloy with particularly high resistance against aggressive, oxidizing and reducing media. Applications: components under strong chemical action in chemical processes and plants, exhaust cleaning systems, production of fibers and paper, waste disposal ... ...

Hastelloy G

Highly corrosion-resistant nickel-chrome-iron alloy with excellent resistance to corrosion in oxidizing media. Applications: in chemical process engineering, production of phosporic acid and nitric acid, desulphurization plants ... ...

Inconel

Nickel-chrome alloy with good industrial properties at high temperatures up to and above 1000 °C, excellent resistance to oxidation. Applications: heat treatment plants, nuclear energy technology, gas turbines, linings, chemical industry

Monel

Nickel-copper alloy with high strength and toughness over a wide range of temperatures. Excellent resistance to corrosion of salt water, many acids and alkaline solutions.

Source: Technical Information – Bossard AG, CH 6301 Zug, 2012.06

Carbon steels and corrosion-resistant materials - 3

Nimonic

A family of nickel-chrome alloys with particularly high fatigue strength and resistance to oxidation. For high mechanical stresses at temperatures up to 1000°C. Hardening methods allow to control relaxation and creep.
Applications: rotating components at high temperatures, springs, fasteners, combustion chamber components, blades ... ...

Titanium grades 1, 2, 3, 4

High strength in relation to low density. Excellent resistance to corrosion under chloride. Applications: components with weight saving and high strength, subject to strong oxidizing conditions specially chlorides, chemical industry, seawater desalination, power station technology, medical technology ... ...

Titanium grade 5

Titanium alloy with a high specific strength. Applications: components for aero-space industry, chemical processing technology, rotating components, fasteners, ...

Titanium grades 7, 11

Pure titanium alloyed with palladium, high resistance to corrosion specially in moist media containing chlorine. Grade 11 has increased deformation properties.
Applications: chemical and petrochemical plants, ...

Source: Technical Information – Bossard AG, CH 6301 Zug, 2012.06

Carbon steels – DIN EN 10269

Source: VDI 2230-2003 Blatt 1 / Part 1

Werkstoff/Material		Mechanische Eigenschaften/Mechanical properties (in MPa)														
Nr. No.	Kurzname nach DIN 17 006 Symbol according to DIN 17 006	R_m	$R_{p0,2}$	Warmstreckgrenze $R_{p0,2T}$ High-temperature yield point $R_{p0,2T}$ für/for $d \leq 100$ bei/at T in °C						Zeitdehngrenze $R_{p0,2/10\,000}$ Creep limit $R_{p0,2/10\,000}$ bei/at T in °C						
		min.	min.	200	300	400	500	600	700	200	300	400	500	600	700	800
1.5511	35B2	500	300	229	192	173	–	–	–		208[4])	147	35	–	–	–
1.7218	25CrMo4	600	440	412	363	304	235	–	–			274[5])	147	64[6])	–	–
1.7709	21CrMoV5-7	700	550	500	460	410	350	–	–			429[5])	238	116[6])	–	–
1.7711	40CrMoV4-7	850	700	631	593	554	470	293	–			361[7])	242	138		
1.4301	X5CrNi18-10	500	190	127	110	98	92	90[1])	–				121[6])	94	35	–
1.4923	X21CrMoNiV12-1	800	600	530	480	420	335	280[1])	–			436[7])	289	79	–	–
1.4980	X6NiCrTiMoVB25-15-2	900	635	560	540	520	490	430	≈ 310[2])				580	320	190[8])	–
2.4952	NiCr20TiAl (Nimonic 80a)	1000	600	568	560	540	520	500	≈ 450[3])				624	398	173	58

1) at T = 550 °C
2) 380 at T = 650 °C
3) 480 at T = 650 °C
4) at T = 350 °C
5) at T = 420 °C
6) at T = 550 °C
7) at T = 450 °C
8) at T = 650 °C

continues ...

Carbon steels – DIN EN 10269

Source: VDI 2230-2003 Blatt 1 / Part 1

		E 10^3 MPa	λ W/(mK)	Elastizitätsmodul E_T (in 10^3 MPa) Young's modulus E_T (in 10^3 MPa) bei/at T in °C						Wärmeausdehnungskoeffizient α_T (in 10^{-6} K^{-1}) Coefficient of thermal expansion α_T (in 10^{-6} K^{-1}) bei/at T in °C von 20 bis						
		bei 20 °C:		200	300	400	500	600	700	200	300	400	500	600	700	800
1.5511	35B2	211	42	196	186	177	164	127	–	12,1	12,9	13,5	13,9	14,1	–	–
1.7218	25CrMo4															
1.7709	21CrMoV5-7		33													
1.7711	40CrMoV4-7		33													
1.4301	X5CrNi18-10	200	15	186	179	172	165		–	16,5	17,0	17,5	18,0		–	–
1.4923	X21CrMoNiV12-1	216	24	200	190	179	167	127	–	11,0	11,5	12,0	12,3	12,5	–	–
1.4980	X6NiCrTiMoVB25-15-2	211	12	200	192	183	173	162		17,5	18,7	18,0	18,2	18,5	18	–
2.4952	NiCr20TiAl	216	13	208	202	196	189	179	161	12,6	13,1	13,5	13,7	14,0		

The next two tables will show chemical and mechanical parameters for all bolt "classes", from 3.6 to 12.9.

For "property class" definition of screws see sl. 13, for nuts see sl. 14 of this Section.

Screws in carbon steel - EN ISO 898-1

Property class	Material and heat treatment	Chemical composition limits (check 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]		—	0,55	0,05	0,06	0,003	—
4.6[2]							
5.6		0,13	0,55	0,05	0,06	0,003	—
5.8[2]		—	0,55	0,05	0,06		
6.8[2]							
8.8[3]	Carbon steel with additives (e.g. Boron, Mn or Cr), quenched and tempered	0,15[4]	0,40	0,035	0,035	0,003	425
	or Carbon steel, quenched and tempered	0,25	0,55	0,035	0,035		
9.8	Carbon steel with additives (e.g. Boron, Mn or Cr), quenched and tempered	0,15[4]	0,35	0,035	0,035	0,003	425
	or Carbon steel, quenched and tempered	0,25	0,55	0,035	0,035		
10.9[5, 6]	Carbon steel with additives (e.g. Boron, Mn oder Cr), quenched and tempered	0,15[4]	0,35	0,035	0,035	0,003	340
10.9[6]	Carbon steel, quenched and tempered	0,25	0,55	0,035	0,035	0,003	425
	or Carbon steel with additives (e.g. Boron, Mn or Cr), quenched and tempered	0,20[4]	0,55	0,035	0,035		
	or Alloyed steel, quenched and tempered[7]	0,20	0,55	0,035	0,035		
12.9[6, 8, 9]	Alloyed steel, quenched and tempered[7]	0,28	0,50	0,035	0,035	0,003	380

Source: Technical Information – Bossard AG, CH 6301 Zug, 2012.06

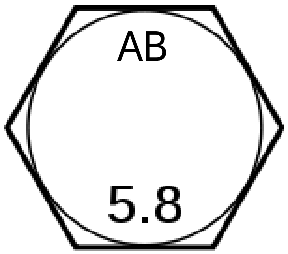
Screws in carbon steel - EN ISO 898-1

Sub-clause number	Mechanical and physical property		Property class 3.6	4.6	4.8	5.6	5.8	6.8	8.8[1] $d \le$ 16mm[3]	8.8[1] $d >$ 16mm[3]	9.8[2]	10.9	12.9
5.1 und 5.2	Tensile strength Rm in N/mm² [4), 5)]	nominal value	300	400		500		600	800	800	900	1000	1200
		min.	330	400	420	500	520	600	800	830	900	1040	1220
5.3	Vickers hardness HV $F \ge 98$ N	min.	95	120	130	155	160	190	250	255	290	320	385
		max.	220[6)]					250	320	335	360	380	435
5.4	Brinell hardness HB $F = 30\ D^2$	min.	90	114	124	147	152	181	238	242	276	304	366
		max.	209[6)]					238	304	318	342	361	414
5.5	Rockwell hardness 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
5.6	Surface hardness HV 0,3	max.	—						7)				
5.7	lower yield stress R_{eL}[8)] in N/mm²	nominal value	180	240	320	300	400	480	—	—	—	—	—
		min.	190	240	340	300	420	480	—	—	—	—	—
5.8	Stress at 0,2% non-proportional elongation $R_{p0,2}$[9)] in N/mm²	nominal value	—					—	640	640	720	900	1080
		min.	—					—	640	660	720	940	1100
5.9	Stress under proofing load S_p	S_p / R_{eL} or $S_p / R_{p0,2}$	0,94	0,94	0,91	0,93	0,9	0,92	0,91	0,91	0,9	0,88	0,88
		N/mm²	180	225	310	280	380	440	580	600	650	830	970
5.10	Breaking torque, M_B Nm min.		—						see ISO 898-7				
5.11	Percent elongation after fracture *A* in %	min.	25	22	—	20	—	—	12	12	10	9	8
5.12	Reduction area after fracture Z	% min.	—						52		48	48	44
5.13	Strength under wedge loading[5)]		The values for full size bolts and screws (not studs) shall not be smaller than the minimum values for tensile strength shown in 5.2										
5.14	Impact strength, *KU* in J	J min.	—			25	—		30	30	25	20	15

Source: Technical Information – Bossard AG, CH 6301 Zug, 2012.06

Marking of screws (carbon steels)

The property "class" of the screw is identified with a permanent marking on the head. The manufacturer's identification code (example: AB) may be added



Carbon steel screw head markings

Stainless steel screw head markings

(see sl. 1 of this section)

Property classes for carbon steel bolts, screws and studs

The screw "property class" code is composed of two numbers separated by a dot. The number to the left multiplied by 100 provides an indication of the ultimate stress R_m (MPa), while ".x" - where x is the number to the right - multiplied by the ultimate stress gives the stress at 0,2% permanent set $R_{p0,2}$ (MPa).

Example: 8.8 i.e. $R_m = 800$ MPa , $R_e = 800 \cdot 0,8 = 640$ MPa

Tensile strength and yield strength of a screw

Tensile strength is the axial stress in the bolt or screw at fracture, i.e. when reaching the minimum value of the "ultimate stress" or "tensile strength" $\mathbf{R_m}$.

The breaking load of a fastener is determined by the formula: $\mathbf{F = A_s\, R_m}$.

• **F**= Breaking load (lbs , N) • $\mathbf{R_m}$= Tensile strength (psi , MPa) • $\mathbf{A_s}$= Stress area (in^2, mm^2)

Yield strength is the axial stress in the bolt or screw at yield, i.e. when reaching the minimum value of "yield stress" $\mathbf{R_{p0,2}}$. Only axial load is applied, i.e. with no torsional stress due to a torque.

The yield load of a fastener is determined by the formula: $\mathbf{F_{0,2} = A_s\, R_{p0,2}}$.

• $\mathbf{F_{0,2}}$ = Yield load (lbs , N) • $\mathbf{R_m}$= Yield stress (psi , MPa) • $\mathbf{A_s}$= Stress area (in^2, mm^2)

Nominal **yield strength** should not be confused with the **yield strength in operation** (either in assembly tightening or in operation) where the axial stress due to axial load produces a yield stress in the bolt in combination with and shear stresses due to torque.

This will explored later in the Sections devoted to bolt design.

Proof load of a screw

The general definition of proof load is: a predetermined test load, greater than the service load, to which a specimen is subjected before acceptance for use.

By definition, the **proof load** is an applied tensile load that the screw must take without any permanent set, i.e., the bolt returns to its original shape (within a certain tolerance, see next slide) once the load is removed.

The proof load of a fastener is determined by the formula: $\mathbf{F_p = A_s\, S_p}$.

$\mathbf{F_p}$= Proof load (lbs , N) $\mathbf{S_p}$= Proof stress (psi , MPa) $\mathbf{A_s}$= Stress area (in^2, mm^2)

In the case of bolts and screws the VDI 2230 standards define the proof load as the bolt load at the minimum value of the "proportionality stress " $\mathbf{R_{p0,2}}$ which is considered a "proof stress " (the VDI 2230 define explicitly: $\mathbf{R_{p0,2}}$ = 0,2 % proof stress of the bolt according to DIN EN ISO 898-1). As an example, a M20 coarse pitch screw, class 8.8, $\mathbf{A_s}$=157 mm^2, has a "Load at the minimum yield point" $\mathbf{F_{0,2min}} = \mathbf{R_{p0,2min}}\, \mathbf{A_S}$ =100000 N (ref. VDI 2230/2003 Tab. A11).

This value diverges from the one in DIN EN ISO 898-1:1999, where (Tab. 7, see also sl. 7 of this Section):

- Stress under proof load: $\mathbf{S_p}=0,91\, \mathbf{R_{p0,2min}} = 0,91 \cdot 640=580$ MPa
- Proof load: $\mathbf{F_p = A_s\, S_p} = 157 \cdot 580=91000$ N

Tensile strenght testing of a screw

This tensile test on a screw is carried on similarly to specimen tensile test.

The picture on the right refers to a tensile test carried out on full-size bolts.

When carrying out this test, a minimum free threaded length equal to one diameter (1d) is subjected to the tensile load.

In order to carry out a valid test, the fracture must occur in the shank or the free threaded length of the bolt and not at the junction of the head and the shank.

Where bolts and studs cannot be tested full size, tests are conducted using test specimens machined from the bolt, i.e., on the machined specimen shown in sl. 12 for yield testing.

Test specifications are found for both cases in EN ISO 898-1 and in ASTM F-606 / F-606M.

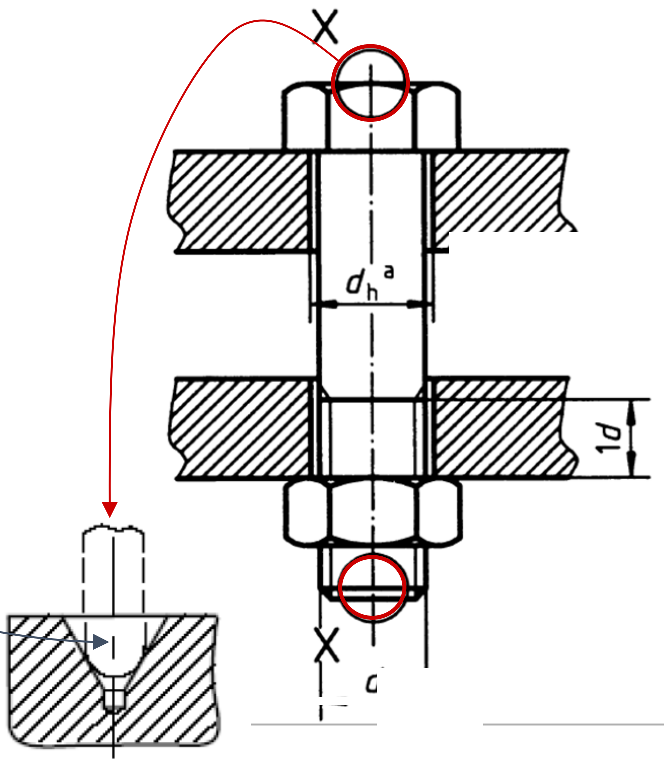
Proof load $\Rightarrow$ based on a stress test on a screw

It consists of two steps:

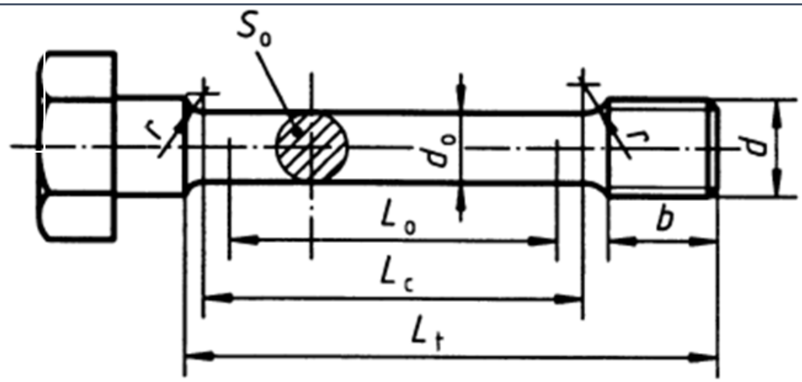
- applying a tensile load
- measuring the eventual permanent elongation due to that tensile load.

The screw length must be measured before and after the test by means of an instrument on two spherical contacts set in conical "dimples" at the bolt ends (as indicated in the figure below).

Proof load is the maximum axial force for which the screw length is the same before and after the test with a tolerance of ± 12.5 μm to allow for measurement errors.



Yield stress testing of screw materials



Key

d = nominal diameter

d_o = diameter of test piece (d_o < minor diameter of thread)

b = threaded length ($b \geqslant d$)

$L_o = 5\, d_o$ or ($5{,}65\sqrt{S_o}$): original gauge length for determination of elongation

$L_o \geqslant 3\, d_o$: original gauge length for determination of reduction of area

L_c = length of straight portion ($L_o + d_o$)

L_t = total length of test piece ($L_c + 2r + b$)

L_u = final gauge length (see ISO 6892:1998)

S_o = cross-sectional area before tensile test

S_u = cross-sectional area after fracture

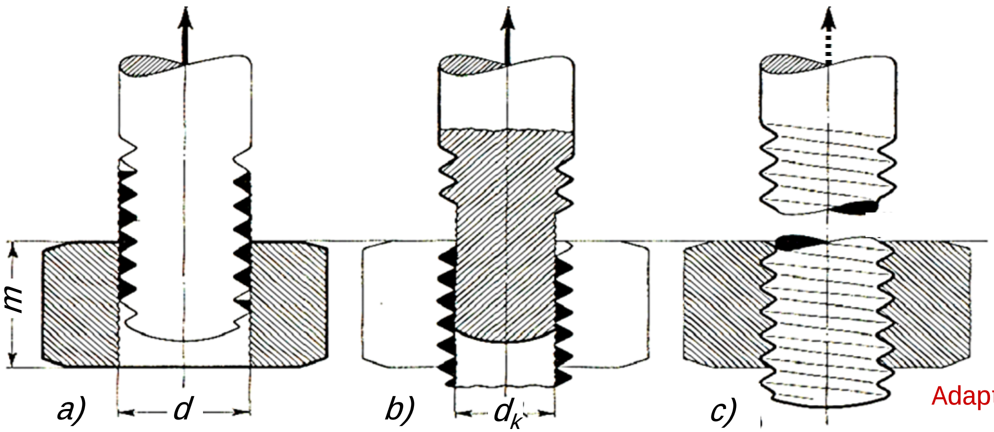
r = fillet radius ($r \geqslant 4$ mm)

This is “Figure 1 - Machined test piece for tensile testing” from EN ISO 898-1:1999. Remark the difference from proof load testing.

Note from EN ISO 898: when machining the test piece, the reduction of the shank diameter of the heat-treated bolts and screws with d > 16 mm shall not exceed 25 % of the original diameter (about 44 % of the initial cross-sectional area) of the test piece.

Stripping strength of a nut

The desired mode of failure of a bolted joint is the complete fracturing of the bolt or screw instead (i.e. before) of the stripping of the thread in the nut or in the internally threaded component.



These figures show:
a) stripping of the nut
b) stripping of the screw
c) fracturing of the screw

Adapted: Karl-Heinz Decker , Maschinenelemente, Carl Hanser Verlag , 1985

The reasoning behind this principle is simple. When a bolt or screw breaks during initial assembly, the joint failure is obvious. When this happens, the broken bolt or screw can be replaced before more serious damage occurs in the assembly.

Shank fracture is sudden and therefore easily noticed. Stripping is gradual and therefore difficult to detect and this introduces the danger of partly failed fasteners being left in assemblies.This can create very dangerous situations in which the assembly may completely come apart and fail when the assembly begins its function and is subjected to its operating loads.

Proof testing of a nut

The following "principle" rule should be followed when nuts are selected for use with bolts or screws:

a nut should have a “proof load capacity” equal to or greater than the “ultimate tensile strength” of the bolt or screw with which it will be used.

It is customary to define either directly a proof load $\mathbf{F_p}$ that a nut shall withstand without stripping or the tensile “(nut) proof stress” $\mathbf{S_p}$ acting in the stress area of the bolt which is coupled to the nut, according to the formula: $\mathbf{F_p = A_s\, S_p}$,

with: • $\mathbf{F_p}$= nut Proof load • $\mathbf{S_p}$= nut Proof stress (in the screw) • $\mathbf{A_s}$= (screw) Stress area

A simple conclusion would be that minimum nut proof stress S_p should be equal to R_m of the screw. The effective situation is quite complex. See: Alexander, E.M., Analysis and design of threaded assemblies. 1977 *SAE Transactions*, Paper No. 770420 and the further considerations at sl. 21-22 of this Section.

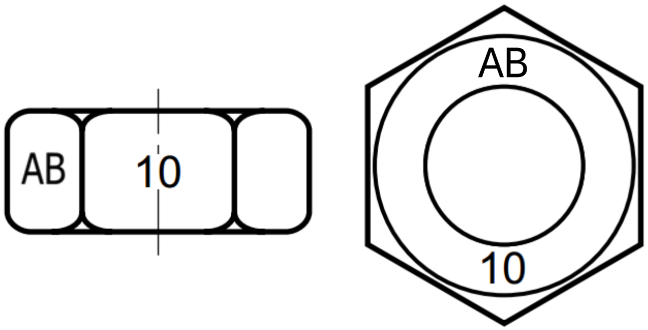
Property class designation for nuts

The choice of nut material and height guarantees that the nut is appropriate to the screw.

To this purpose a "Property class" of the nut is defined. The corresponding code for nuts is a single number and is derived from the left hand number of the property class of the bolt, screw or stud. For the bolt of property class 8.8 a nut of property class 8 is required.

Marking of nuts (carbon steels)

Identification of the nut property class is mandatory for hexagon nuts with thread diameter d $\geq$ 5 mm. The class number is marked by an indentation on the bearing surface or on the side. The manufacturer’s identification code may be added.



Property class of nut	Mating bolts		Nuts	
			Style 1	Style 2
	Property class	Thread range	Thread ranges	
4	3,6; 4,6; 4,8	> M16	> M16	—
5	3,6; 4,6; 4,8	≤ M16	≤ M39	—
	5,6; 5,8	≤ 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

On the left the designation system for nuts with nominal heights ≥ 0,8D, EN ISO 898-1.

There are two styles of nut, style 2 being approximately 10% higher than style 1.

Style 1 height is intended for property classes 4, 5, 6, 8, 10 and 12 (up to M16) with appropriate mechanical properties, while style 2 nuts are intended for use with classes 8, 9 and 12.

A bolt or screw of a particular property class assembled with the equivalent property class of nut, in accordance with the Table in this page, is intended to provide an assembly capable of being tightened to achieve a bolt tension equivalent to the bolt proof load or yield load without stripping.

Thread		Property class 8										Property class 10				
		Stress under proof load Sp	Vickers hardness HV		Nut		Stress under proof load Sp	Vickers hardness HV		Nut		Stress under proof load Sp	Vickers hardness HV		Nut	
greater than	less than or equal to	N/mm²	min.	max.	state	style	N/mm²	min.	max.	state	style	N/mm²	min.	max.	state	style
—	M4	800	180									1 040				
M4	M7	855		302	NQT		—	—	—	—	—	1 040				
M7	M10	870	200			1						1 040	272	353	QT	1
M10	M16	880										1 050				
M16	M39	920	233	353	QT		890	180	302	NQT	2	1 060				

Thread		Property class 12									
		Stress under proof load Sp	Vickers hardness HV		Nut		Stress under proof load Sp	Vickers hardness HV		Nut	
greater than	less than or equal to	N/mm²	min.	max.	state	style	N/mm²	min.	max.	state	style
—	M4	1 140					1 150				
M4	M7	1 140	295	353	QT	1	1 150				
M7	M10	1 140					1 160	272	353	QT	2
M10	M16	1 170					1 190				
M16	M39	—	—	—	—	—	1 200				

The table excerpts from ISO 898-2/1992 show nut proof stress values for property classes 8 to 12.

The more recent ISO 898-2:2012 give only the proof loads.

NQT = Not quenched or tempered.
QT = Quenched and tempered.

Nut proof load - test method

The nut is assembled on a hardened and threaded test mandrel as shown in the figure on the right.

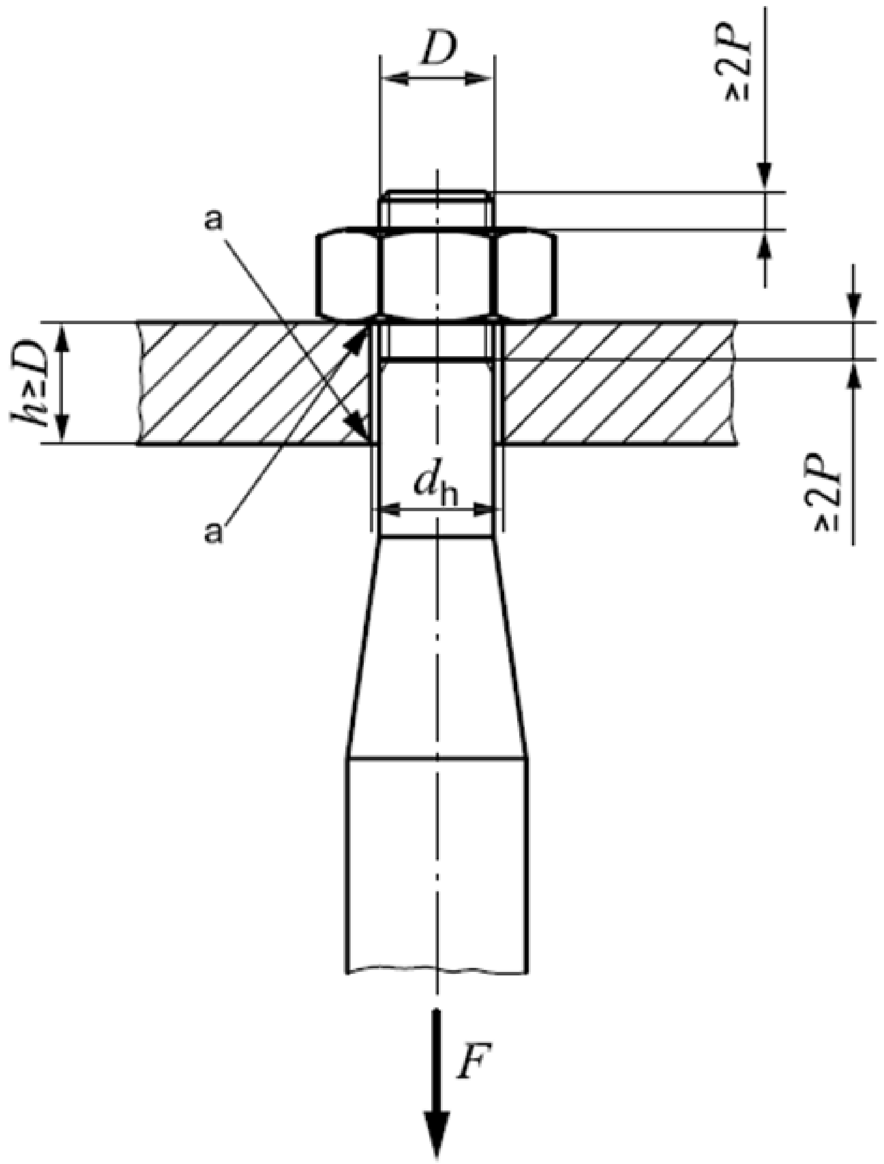
The hardness of the test mandrel must be 45 HRC minimum.

The specified proof load is applied against the nut in an axial direction, and shall be held for 15 s.

The nut must resist the load without failure by stripping or rupture, and must be removable by the fingers after the load is released.

If the thread of the mandrel is damaged during the test, the test is discarded.

It may be necessary to use a manual wrench to start the nut in motion. Such wrenching is permissible provided that it is restricted to one half turn and that the nut is then removable by the fingers.



Further considerations on screw proof load

The apparent conflict between VDI 2230 and DIN EN ISO 898-1 put forward in sl. 10 of this Section is clarified by the following excerpts from ASTM A606-07 *:

3.2.1 Proof Load - The proof-load test consists of stressing the product with a specified load that the product must withstand without measurable permanent set. The proof-load test consists of stressing the product with a specified load that the product must withstand without measurable permanent set. Alternative tests for determining the ability of a fastener to pass the proof-load test are the yield strength test and the uniform hardness test.

Either Method 1 (3.2.3), Method 2 (3.2.4), or Method 3 (3.2.5) may be used, but Method 1 shall be the arbitration method in case of any dispute as to acceptance of the product .

*3.2.3 Method 1, Length Measurement - To ensure consistent and repetitive length measurements of the fastener, the threaded end and top of the bolt head shall have conical depressions made at the approximate axis or center line of the fastener. ... The measuring instrument shall have pointed anvils which mate with the center line depressions and be capable of measuring changes in length of 0.0001 in. with an accuracy of 0.0001 in. in any 0.001 in. range*** ... ...*

*ASTM A606-07 Standard Test Methods for Determining the Mechanical Properties of Externally and Internally Threaded Fasteners, Washers, and Rivets

**Compare with sl. 11 of this Section

... axially load the fastener to the proof load value specified in the product specification ... The measurement shall show no permanent elongation.

3.2.4 Method 2, Yield Strength - Assemble the product in the testing equipment ... As the load is applied, measure and record the total elongation of the product or any part of it that includes the exposed threads to produce a load-elongation diagram. Determine the load or stress at an offset equal to 0.2 % of the length of bolt ...

An example of results is taken from ASTM A490M-04 *:

Nominal Bolt Diameter mm	min Tensile Strength MPa	max Tensile Strength MPa	Proof Stress Length Meas. Method MPa	min Yield Strength Yield Strength Method MPa
M12 to M36 inclusive	1040	1210	830	940

*ASTM A490M-04 Standard Specification for High-Strength Steel Bolts, Classes 10.9 and 10.9.3, for Structural Steel Joints [Metric]

Further considerations on nut proof load

We said in sl. 14 that a simple rule would be to assume a minimum nut proof stress S_p equal to R_m of the screw. In fact, the situation is complex, and we might ask ourselves whether the reference value of R_m ought to be not the minimum guaranteed but rather the maximum allowed.

Before reading this and the next slide, observe that according to the ISO Table on sl. 17 of this section the proof stress $\mathbf{S_p}$ for the M16 nut of class 8 is taken at 880 MPa. Observe further that the Table on sl. 7 shows minimum values for R_e or $R_{p0,2}$ that are equal or greater than their nominal values.

(From BS EN 20898-2:1994) - *Previously it was considered adequate if the nut proof load was designed equal to the bolt minimum ultimate strength, however, the advent of yield point tightening methods and improved understanding of the interaction between nut and bolt threads showed the nuts required re-design to provide greater resistance to stripping of both the internal and external threads.*

For example, consider that the effective tensile strength of a bolt of class 8.8 may be between 800 N/mm^2 and about 965 N/mm^2 (determined from the maximum hardness) in sizes up to M16. Consequently the yield stress may range between 640 N/mm^2 and 772 N/mm^2 for a yield to ultimate stress ratio of 80%. With the use of yield point tightening it will be seen that the tightening stress approaches the proof stress.

4 - Bolt materials and strength fundamentals (23/23)

Recent research has, in addition, shown that a nut tested with a hardened mandrel is capable or sustaining a higher load before stripping than when tested with a bolt of the appropriate property class.

For example, a property class 8 nut when tested with a mandrel of 45 HRC will be capable of approximately 10% higher load than when tested with a property class 8.8 bolt of dimensions similar to the mandrel.

Therefore, a nut that just meets a proof stress of 800 N/mm2 with a hardened mandrel might only be expected to sustain a load of approximately 720 N/mm2 when mated with a property class 8.8 bolt of minimum dimensions.

It will be seen that stripping of the threads may occur when tightening to stresses in excess of this, and from the bolt mechanical properties it will be seen that this could be a frequent occurrence with yield point tightening.

It might be argued, however, that under torque tension loading the tensile strength of the bolt is reduced by about 15 %, but it should also be realized that the stripping strength of the assembly is also reduced by almost the same amount under torque tension loading.

Sections 5, 6, 7 - Bolt tightening

These sections deal with the problem of clamping bolts during the assembly process, and the consequences they have on the effective bolt clamping forces.

Section 5 examines different tightening processes, instruments and problems.

Section 6 develops in full detail the relation between the torque applied to the nut or to the bolt head during the torque-controlled tightening process and the bolt tension, or clamping force, which is produced. The role of friction as the major source of uncertainty is explained. A special attention is given to the way the max tightening torque is specified.

Section 7 draws conclusions about the tightening uncertainty, and gives criteria / coefficients to take it into account in the course of design.

Assembly preload

It will be shown, in later sections, how important the assembly preload is for the correct operation of a bolted joint. It will be shown in this section how it can be obtained, and how much the end result - the magnitude of the preload force exerted by the bolt on the assembled part - can be relied upon.

Preload is produced in three main ways:

- Torque-controlled tightening
- Angle-controlled tightening
- Yield-controlled tightening

Torque-controlled tightening

The nut or the bolt head is rotated through a wrench, the idea is that controlling the torque applied to the nut one controls the tension or preload built-up within the bolt. It is by far the best-known, most common and least expensive way to control preload.

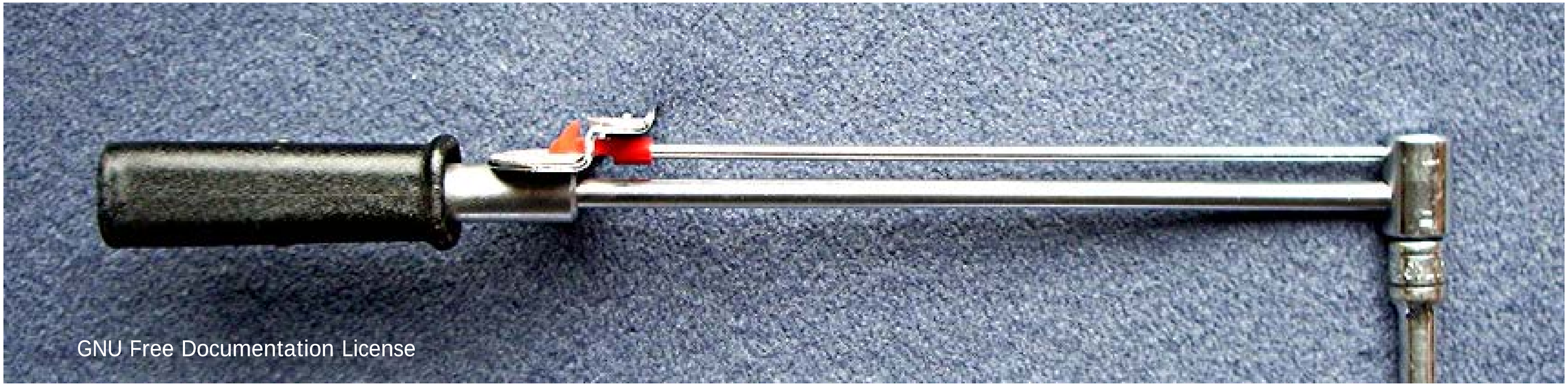
Spanners are used, like the ones shown on the right; many more shapes are available to meet the needs of different applications.

It is clear however that simple spanners do not allow to accurately "control" the torque.

G. Wrigge, I. Gerhardt, 2003; GNU FDL

Torque-controlled tightening

If torque is to be controlled, then it must be measured. Torque (or dynamometric) wrenches, like the one below, make this possible.



Beam type: the simplest form of torque wrench consists of a long lever arm between the handle and the wrench head, made of a material which bends elastically in response to applied torque. Here in a purely mechanical version. Probably outdated, still good.

Torque-controlled tightening

There are other types, as shown here:

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Click type

The torque is preset with a calibrated clutch mechanism. At the point where the desired torque is reached, the clutch slips, signaling the desired torque and preventing additional tightening.

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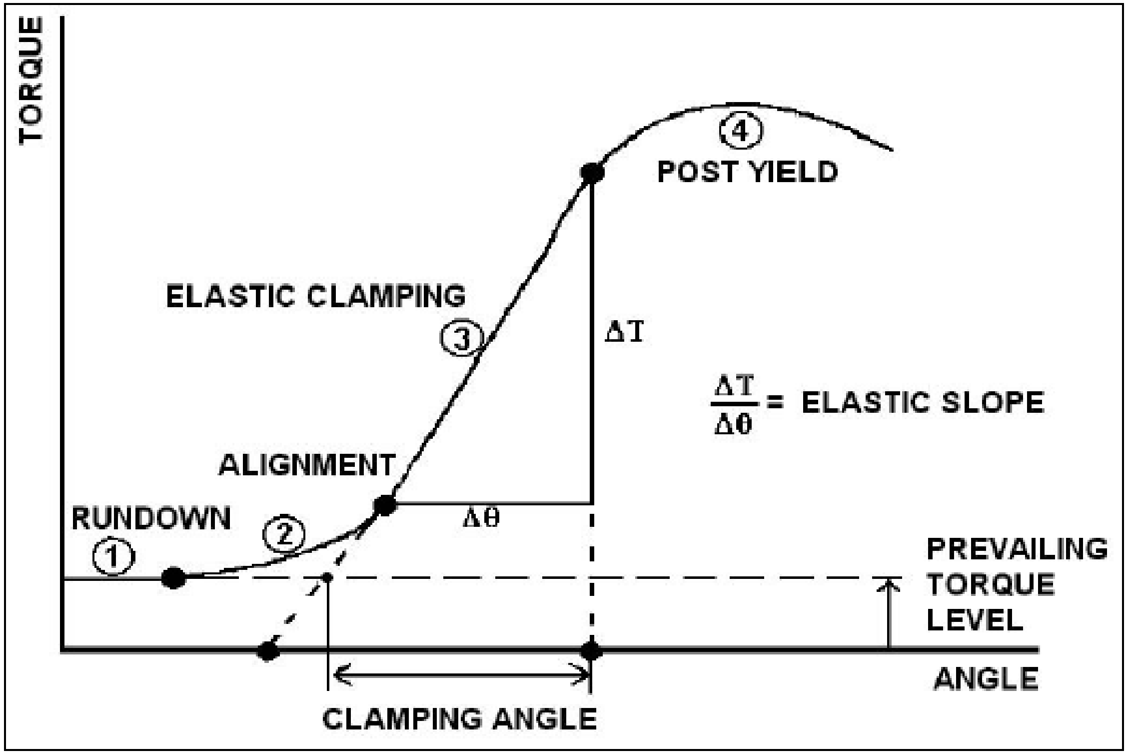
Electronic type

A signal, produced by strain gauge attached to the torsion rod, is converted to the required unit of force (N·m, lbf·ft) and shown on the digital display.

Torque-angle signature

Zone 4: post-yield zone, which begins with an inflection point at the end of the elastic clamping range. Yielding can occur in the bolt or in the joint assembly as a result of underhead embedment or as thread strip in the bolt or mating threads. It can be due to yielding in the joint or gasket, or due to yield of the threads in the nut or clamped components or nut rather than to yield of the fastener.



Source: Engineering Fundamentals of Threaded Fastener Design and Analysis, Ralph S. Shoberg, P.E., RS Technologies, a Division of PCB Load & Torque, Inc.

Zone 3: elastic clamping range, where the slope of the torque-angle curve is *essentially* constant: the tension generated is always proportional to the angle-of-turn from the elastic origin.

Zone 2: alignment or snugging zone, where the fastener and joint mating surfaces are drawn into alignment, or a stable, clamped condition.
Function of drawing together the mating parts, bending of the fastener as a result of non-parallelism of the bearing surface to the fastener underhead surface, plus micro effects such as contact deformations.

Zone 1: the rundown or **prevailing torque** zone that occurs before the fastener or nut contacts the bearing surface. Prevailing torque is due to thread locking features such as nylon inserts or deformed threads.

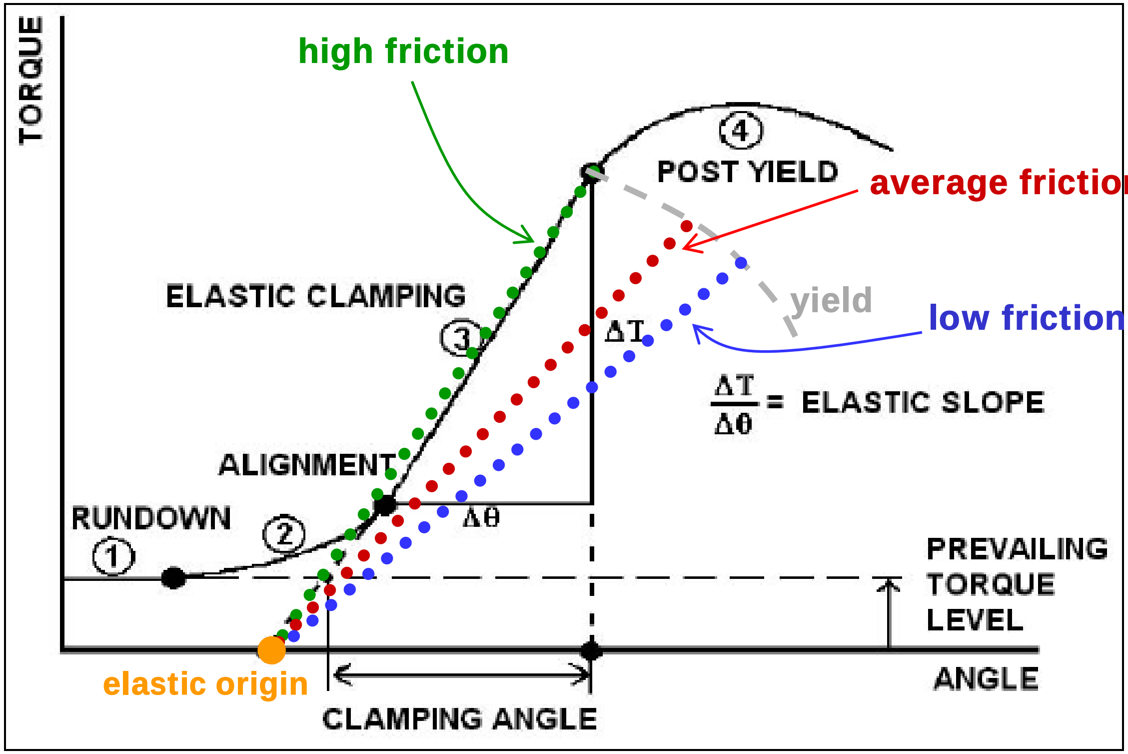
Torque-angle signature

The **elastic origin** is located by projecting a line tangent to the elastic portion of the torque-angle curve backward to zero torque.

An decrease in friction, in the thread or underhead regions, results in a decrease in the slope of the torque-angle signature (**green**, **red**, **blue** dotted lines).

Results for different friction values have been represented after aligning their elastic origins on the same point. The **elastic origin** is their common «zero».

Extensive testing has proved that the tension produced in a bolt clamping given parts is directly proportional to the angle of turn from the elastic origin.



It will be seen in Sect. 8 (sl. 6) that the total angle of turn measured from this point, times the thread pitch, is equal to the interference "i", which is equal to the sum of the compression (shortening) of the clamped components plus the stretch (elongation) of the fastener.

Then the bolt stress, which is proportional to its elongation (and to strain) is independent of the torque necessary to produce bolt or nut rotation, i.e., it is independent of friction and its variability.

These considerations allow to understand the diagrams shown in the next slides, which however require also the knowledge set forth in Sections 6, 8 and 11 (please, read them and come back to this Section) .

Angle-controlled tightening

The figure below shows an experimental torque-angle relation, while the figure above shows the experimental bolt tension-angle relation occurring at the same time.

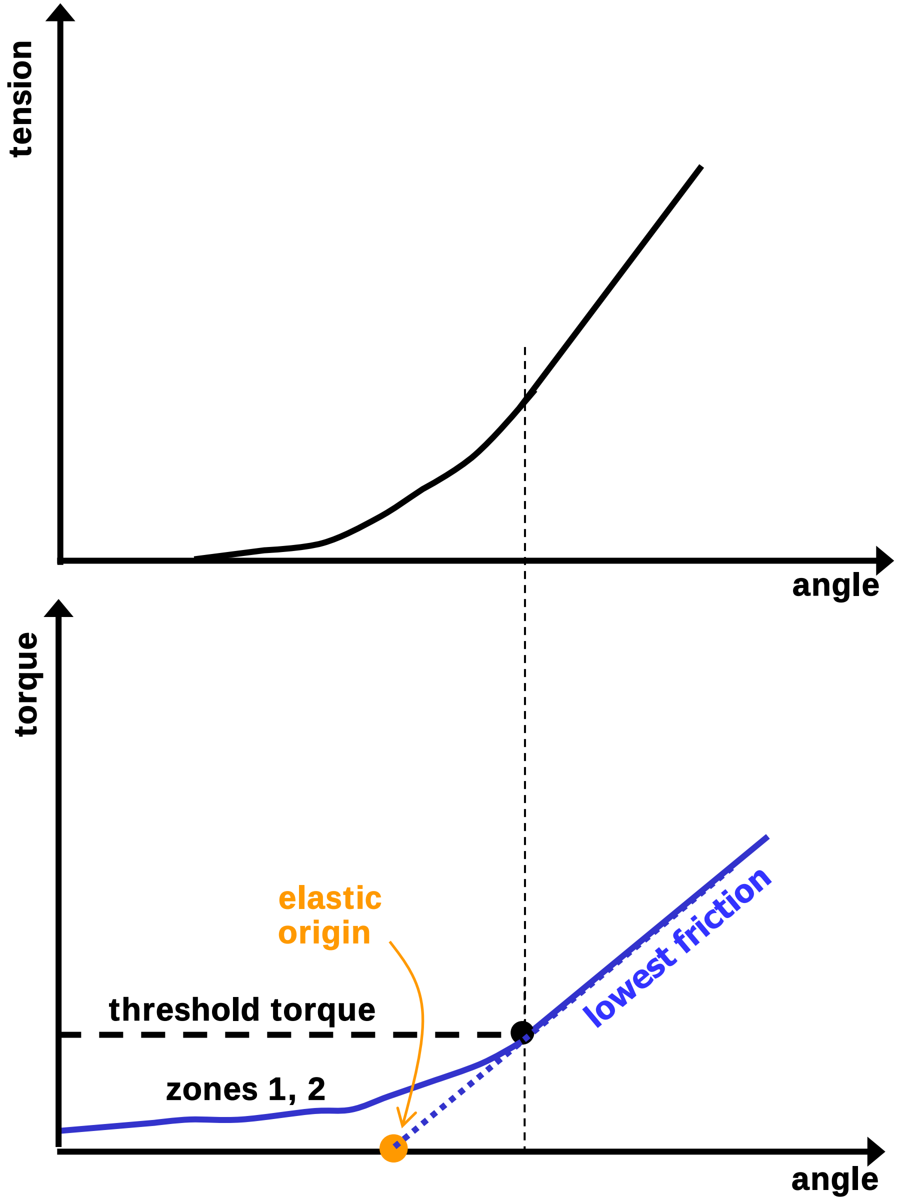
As said before, the tension-angle curve is unique, while the torque-angle curve depends on the friction coefficient.

When the torque-angle diagram becomes linear, at point ● , the torque is the **threshold torque**.

When the torque becomes higher than the threshold torque, the relation between rotation angle of the nut and the elongation of the bolt shaft (i.e. also the tension) becomes linear, i.e., both diagrams are straight lines.

The threshold point is better determined observing where the alignment curve touches the linear line extended back down to torque zero (dotted line).

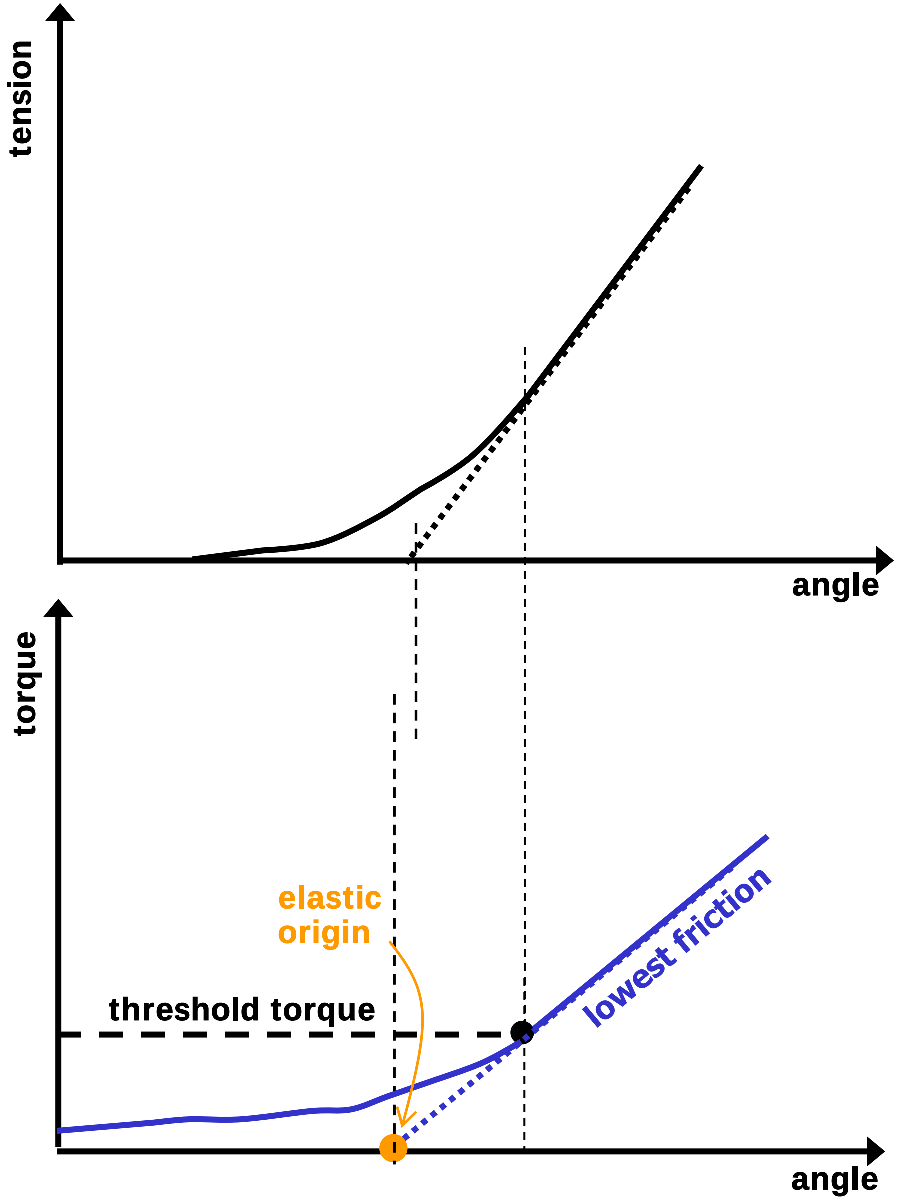
If the rundown & alignment zones had not existed, and the friction contacts (thread and nut) were from the beginning in a regime of kinematical friction, the zero torque would have occurred at the **elastic origin** point ● . i.e., the straight line would have started at this point.



Angle-controlled tightening

If we consider the tension-angle curve, we should find that the elastic origin for the experimental torque-angle diagram is **about** the same also for the tension-angle diagram, within experimental errors,

i.e., when the torque is zero, the tension in the bolt is zero too.

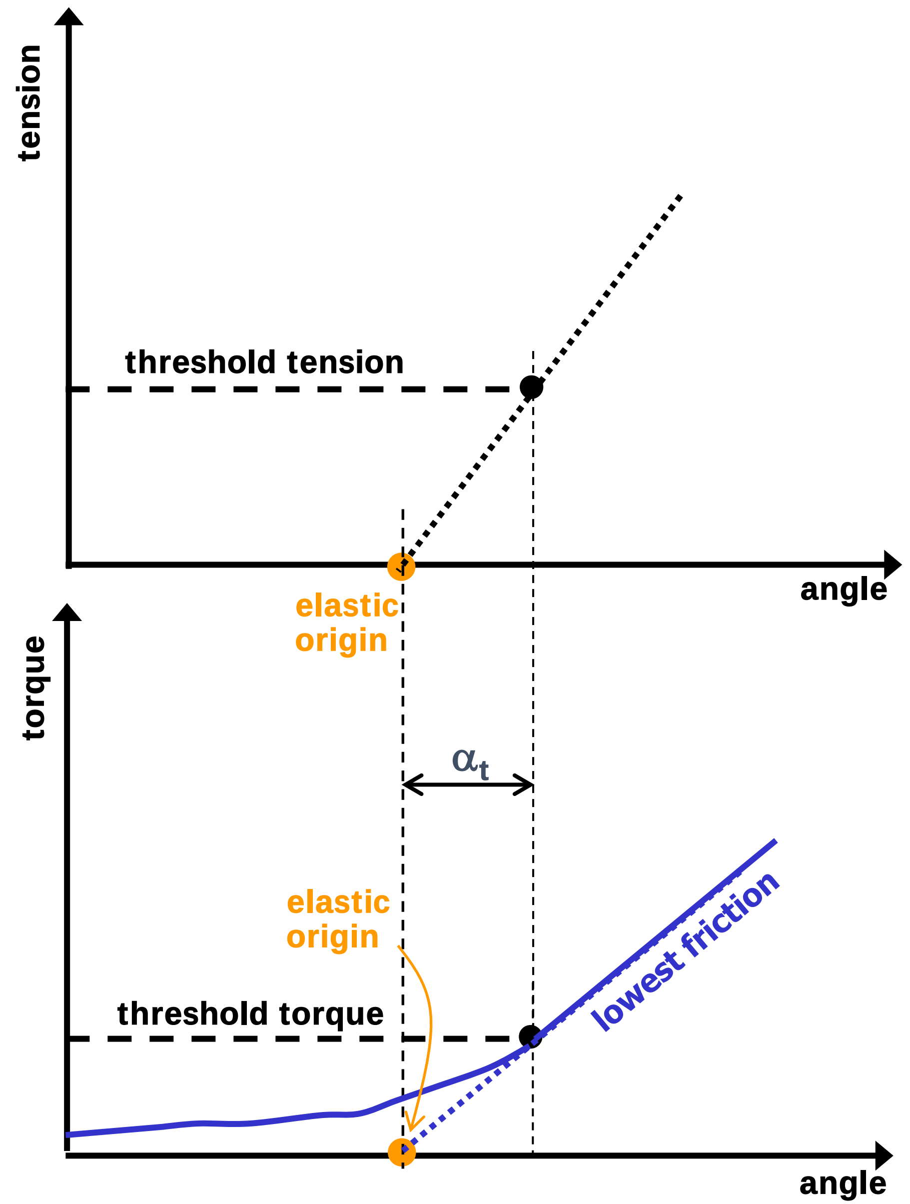


Angle-controlled tightening

However, we normally do not measure experimentally the tension-angle curve, rather we obtain its slope from the elastic theory. (formulas of Sect. 8)

Then we set the origin of this line on the same **elastic origin** angle we found for the torque-angle line.

From now on we consider only the straight elastic lines. The angle on the torque-angle curve at the threshold torque sets also the value of the **threshold tension.**



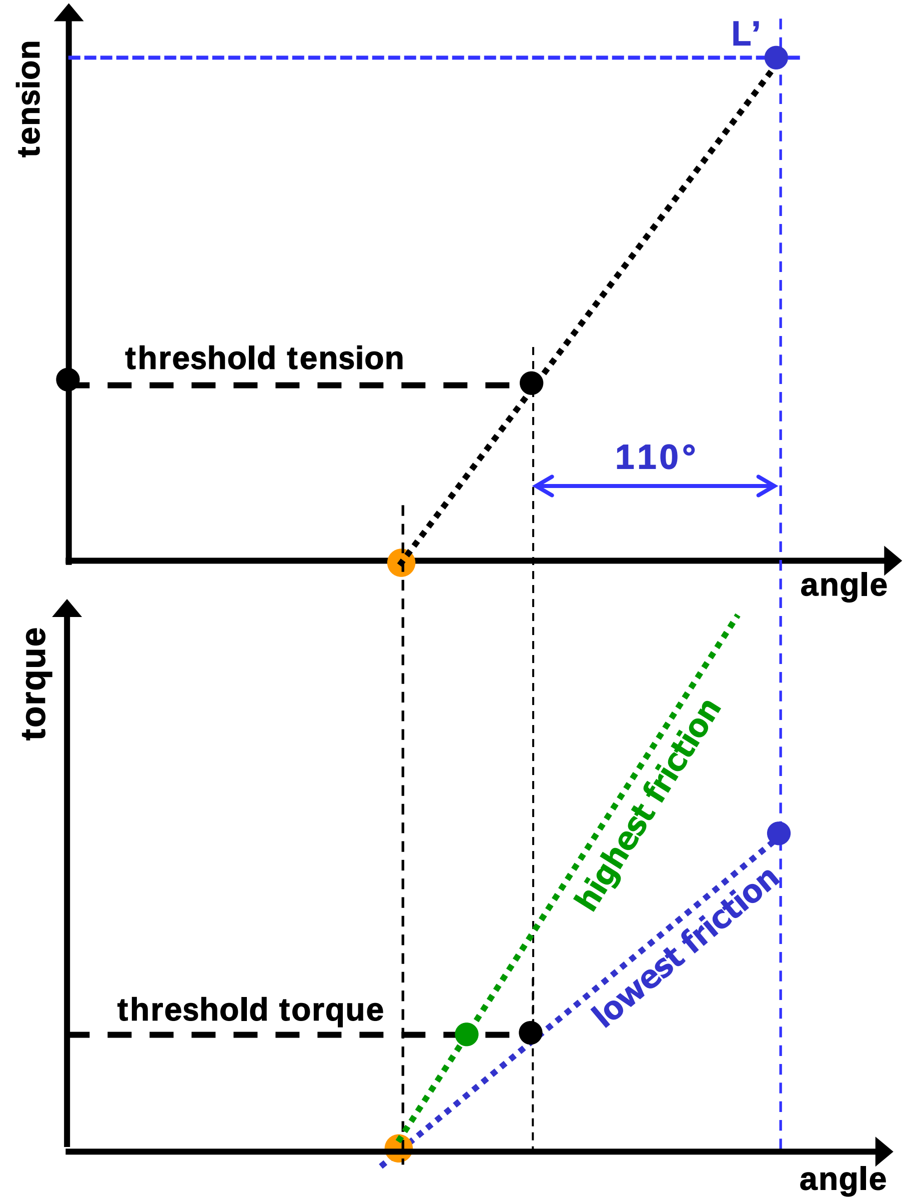
Note that the torque diagram we are considering here is the one obtained experimentally when the bolt is well oiled, i.e., the friction coefficient on thread or nut-face is lowest, so the torque needed for the same tension or angle is lowest.

This is the condition which is easiest to simulate, by properly lubricating the bolt. When bolts are dry, there is a variability due to non controlled conditions.

Angle-controlled tightening

At this point we know how much tension (tensile force) we have in the bolt, ● point on the tension axis, and we know the maximum stress we want to apply, point ● **L'** .

This corresponds to a certain control angle, example 110°, that we add after reaching the threshold torque



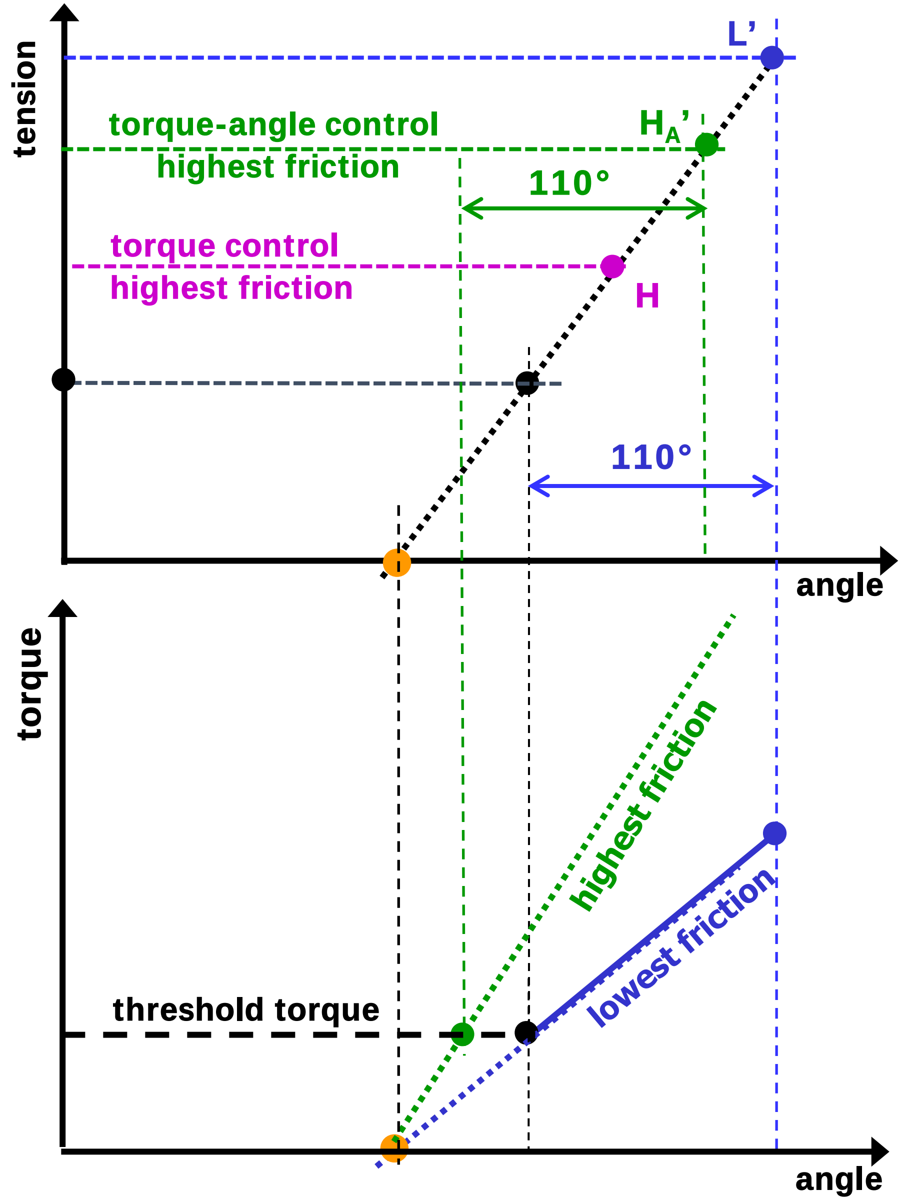
If the friction was higher, e.g., at its highest value, its torque diagram (green curve) would have been higher for the same angles.

Since when we tighten a bolt we do not know the actual friction coefficient (we just know that it could be between a minimum and a maximum), the only thing we can do is to operate with the same **threshold torque** determined before for the **lowest friction**, then for that torque we shall be at the angle marked by point ● .

Angle-controlled tightening

If to this angle (obtained set by the torque value during tightening) we add the same 110° as with the lowest friction, then we get to point ● **H_A'**.

In practical cases, manufacturers of such devices claim that tension in **H_A'** is about 10% lower than tension in **L'** . (*figure is not to scale*)



Finally, we know (see the theory developed further in this chapter) that if we rely on pure torque control, then the tension in the bolt with the highest friction, ● **H** , is lower then the tension in ● **L'** by around 40% .

So, the torque-angle control, although more expensive, is able to much better control the variability in bolts during tightening.

Yield-controlled tightening

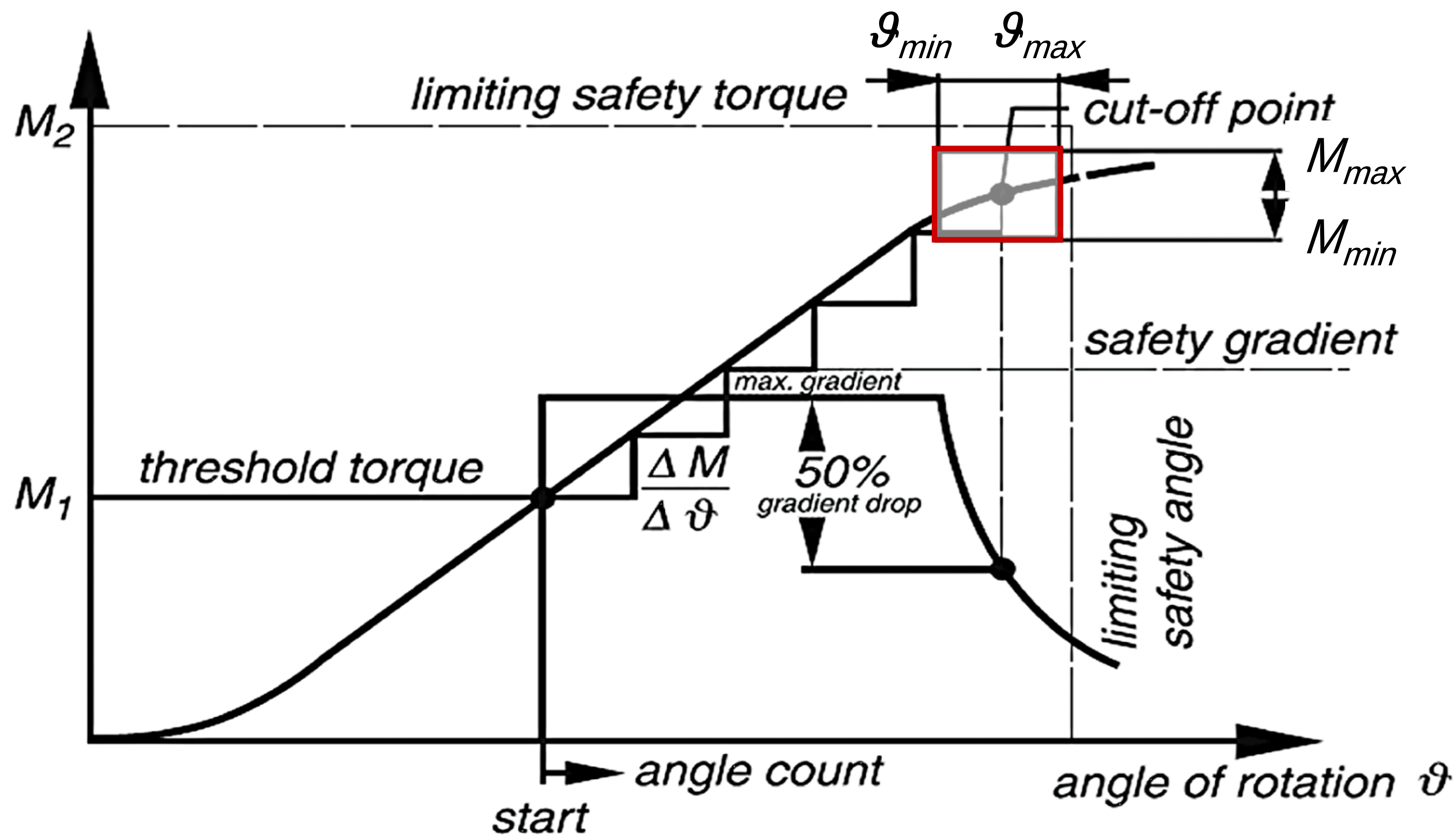
The yield point of the bolt serves as a controlled variable for the assembly preload. Irrespective of the friction at the bearing area (torque values are not used as limits), the bolt is tightened until the yield point or proof stress of the bolt is approximately reached as a result of the combined tensile and torsion stresses. As with angle-controlled tightening, the joint is first of all to be preloaded with a surface setting torque.

The yield point of the bolt is identified by measuring the torque and the angle of rotation during tightening and by determining their difference quotient $dM/d\vartheta$, which is equivalent to the slope of a tangent on the torque/angle curve. Rapid detection of the change in slope of this gradient indicates the yield point has been reached and stops the tightening process.

This is achieved by incorporating sensors to read torque and angle during the tightening process. Very accurate preloads can be achieved by this method because it minimises the influence of friction and of its scatter.

Yield-controlled tightening

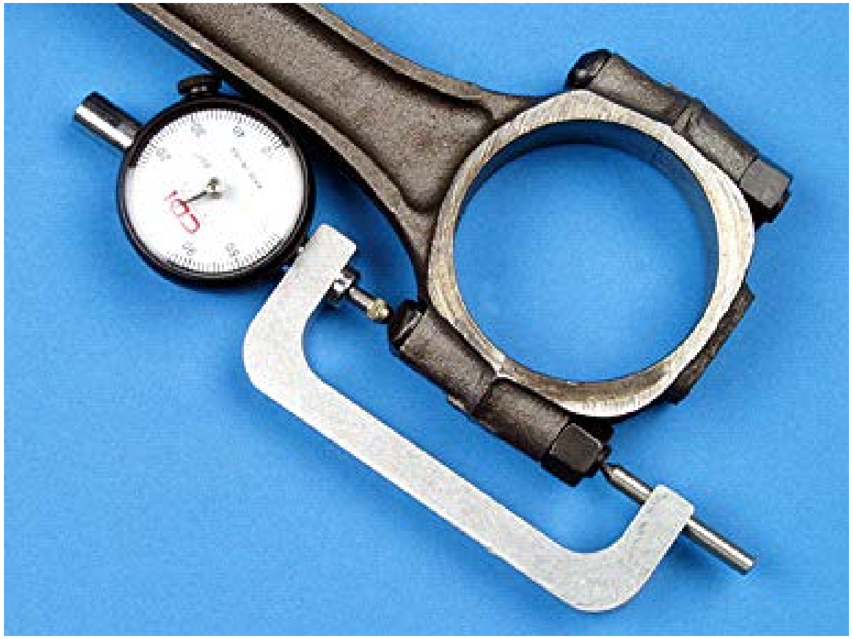
Source: VDI 2230 Blatt 1/ Part 1 - 2003



This procedure only works properly if the observed yield point is due to the bolt yielding. Joints where thread strip or underhead embedment occurs before bolt yield cannot be evaluated by this method.

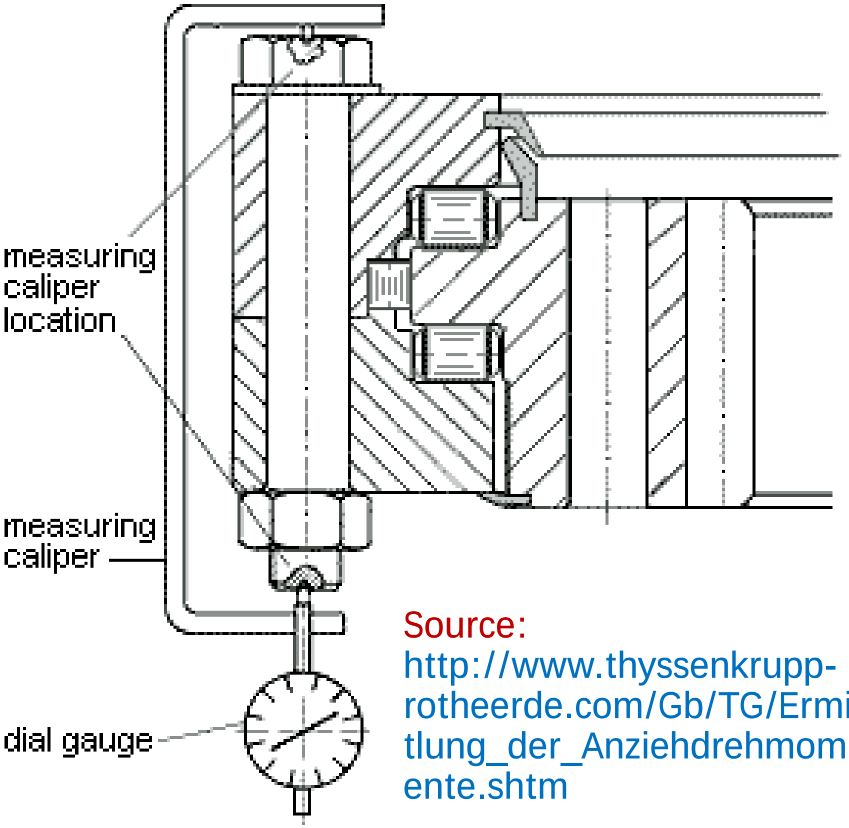
Bolt-elongation (stretch) controlled tightening

In order to measure bolt stretch, first the total rod bolt length is measured (from the head surface to the tip of the shank) in the relaxed state. Then bolt length is monitored during tightening until the specified amount of bolt stretch has been achieved. This can be done either via ultrasound probes or via a mechanical gauge, shown below. The assembly force is independent of friction.



An example of a simple rod bolt stretch gauge. High-performance rod bolts feature a dimple at each end.
This provides a convenient location for a stretch gauge.

Photos Mike Mavrigian - http://www.precisionenginetech.com/tech-explained/2009/06/03/measuring-connecting-rod-bolt-stretch-part-1/



Source: http://www.thyssenkrupp-rotheerde.com/Gb/TG/Ermittlung_der_Anziehdrehmomente.shtm

Bolt elongation measurement can be used to calibrate the tightening torque, however for each specific geometry and material.

Tests and practical experience have shown time and again that the calculated torques for bolts > M 30 or 1¼" are not coinciding with the actual values with adequate precision.

The main influential factor for these differences is thread friction in the bolt and nut contact area, for which to a large extent only empirical or estimated values are available.

The elastic longitudinal elongation at 70 % prestress of the yield point is determined theoretically via the elastic resilience of the bolt.

The bolt is loaded (and torqued) through a spanner until the previously determined bolt elongation is displayed on the dial gauge. This torque is then read off the torque spanner.

To account for any variations, an average value from several measurements should be determined. **Caution**: after a certain operating time the bolt connection must be rechecked for pre-stress and retightened, as required to compensate for any settling phenomena reducing bolt pre-stress, which results in a loss in clamp load.

Sections 5, 6, 7 - Bolt tightening

These sections deal with the problem of clamping bolts during the assembly process, and the consequences they have on the effective bolt clamping forces.

Section 5 examines different tightening processes, instruments and problems.

Section 6 develops in full detail the relation between the torque applied to the nut or to the bolt head during the torque-controlled tightening process and the bolt tension, or clamping force, which is produced. The role of friction as the major source of uncertainty is explained. A special attention is given to the way the max tightening torque is specified.

Section 7 draws conclusions about the tightening uncertainty, and gives criteria / coefficients to take it into account in the course of design.

Torque-controlled tightening

A "preload" force:

F_M: tensile assembly (in german "**M**ontage") force in the bolt shaft is to be generated.

All tightening techniques do not sense the preload produced in the bolt directly but indirectly.

The bolt preload F_M is a function of:

M_G: torsion moment on the bolt shaft (transmitted through the threads, in german "**G**ewinde").

This is produced by the tightening torque applied to the nut or bolt head:

M_A: assembly torsion moment (in german "**A**nziehdrehmoment")

The assembly torque M_A to be applied to nut or head is:

$$M_A = M_K + M_{KZu} + \underbrace{M_G + M_{Ü}}_{M_S}$$

M_S: moment on the bolt shaft, because applied on the thread

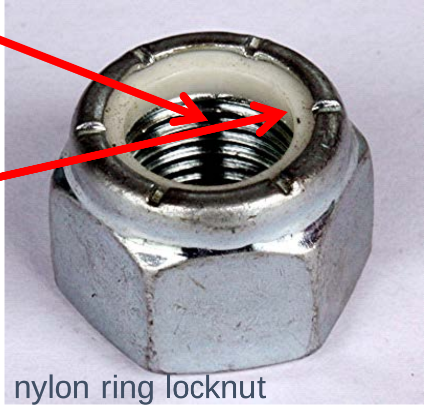
M_K : friction moment in the head (kopf) or nut bearing area

M_{KZu}: additional (**Zu**satz) head (**K**opf) moment, when using elements which prevent slackening (e.g. grip nuts)

M_G : torque acting in the thread ("**G**ewinde")

$M_{Ü}$: overbolting moment when using elements which prevent the bolt from rotating loose (e.g. self locking nuts, or locknuts)

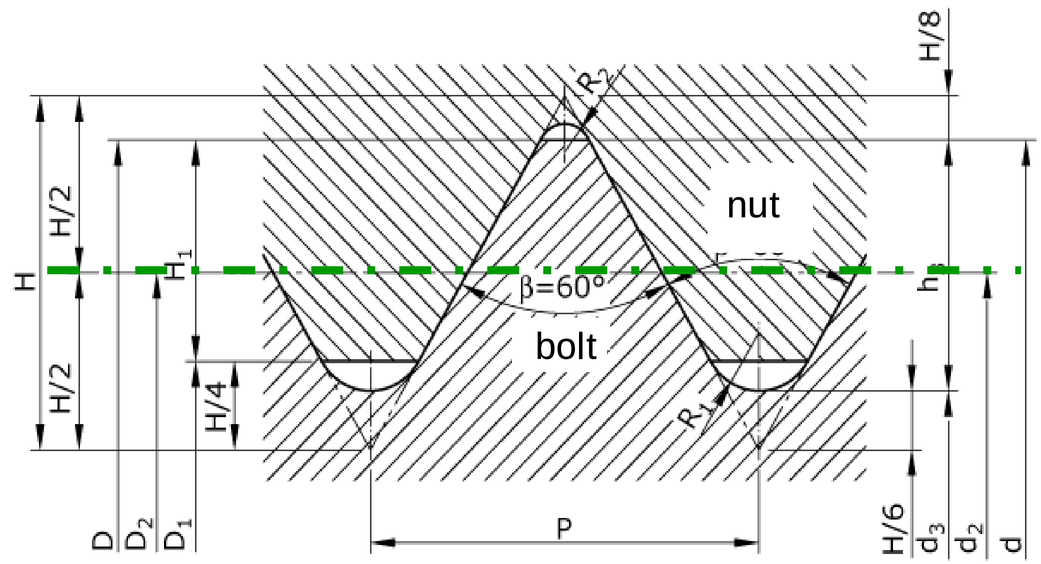
M_{KZu} and $M_{Ü}$ can be ignored in highly preloaded joints

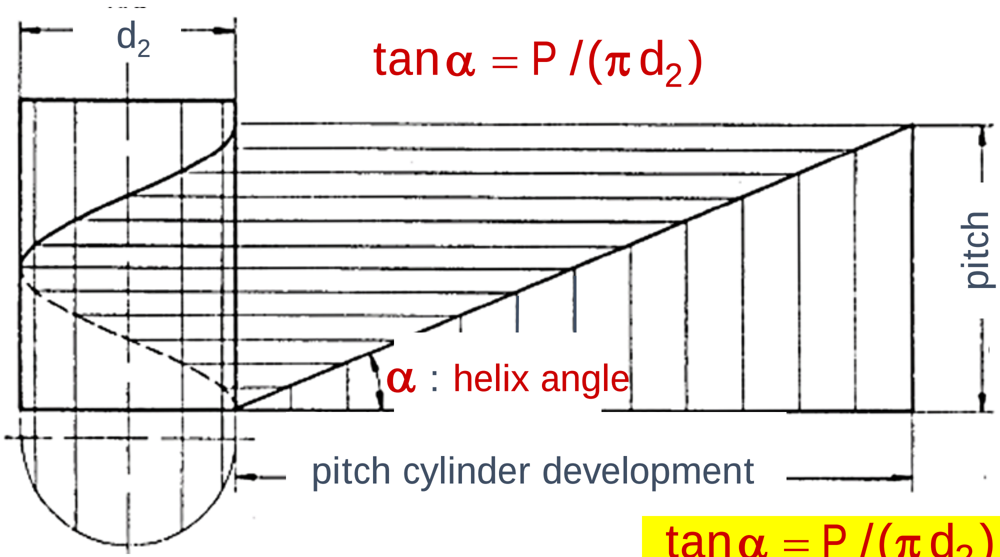


From the equilibrium of the inclined plane, the torsional moment M_G acting in the thread is linked to the bolt preload F_M

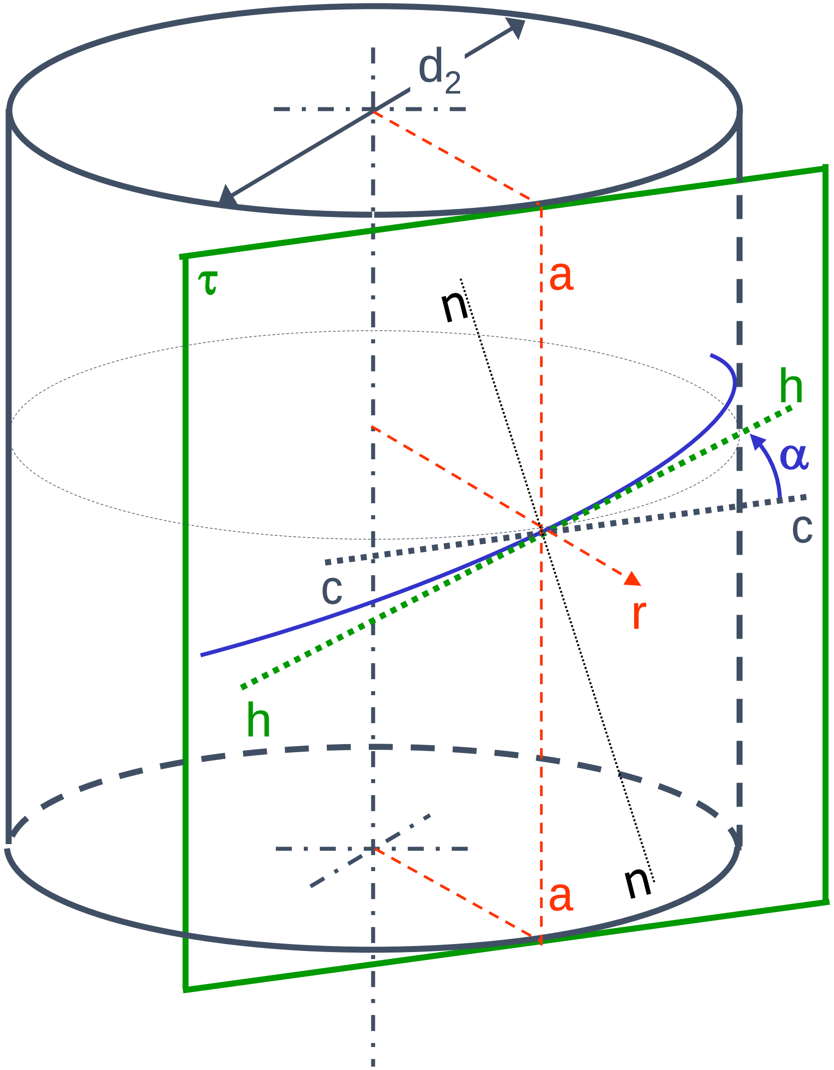
Geometrical parameters:

- d : bolt major i.e. **nominal** diameter
- d_2 : bolt thread **pitch** diameter
- d_3 : bolt thread **minor** diameter
- φ: (equivalent) **friction angle** between the threads
- α: (mean) **helix angle** of the thread





$$\tan\alpha = P/(\pi d_2)$$



The cylinder on the right has the radius of the bolt (and nut) ½ of the **pitch diameter d_2** (see previous slide for symbols).

The figure shows the **plane τ tangent** to the cylinder along the line a-a (parallel to the cylinder axis).

A part of the helix (intersection of the thread surface with the pitch cylinder) is shown.

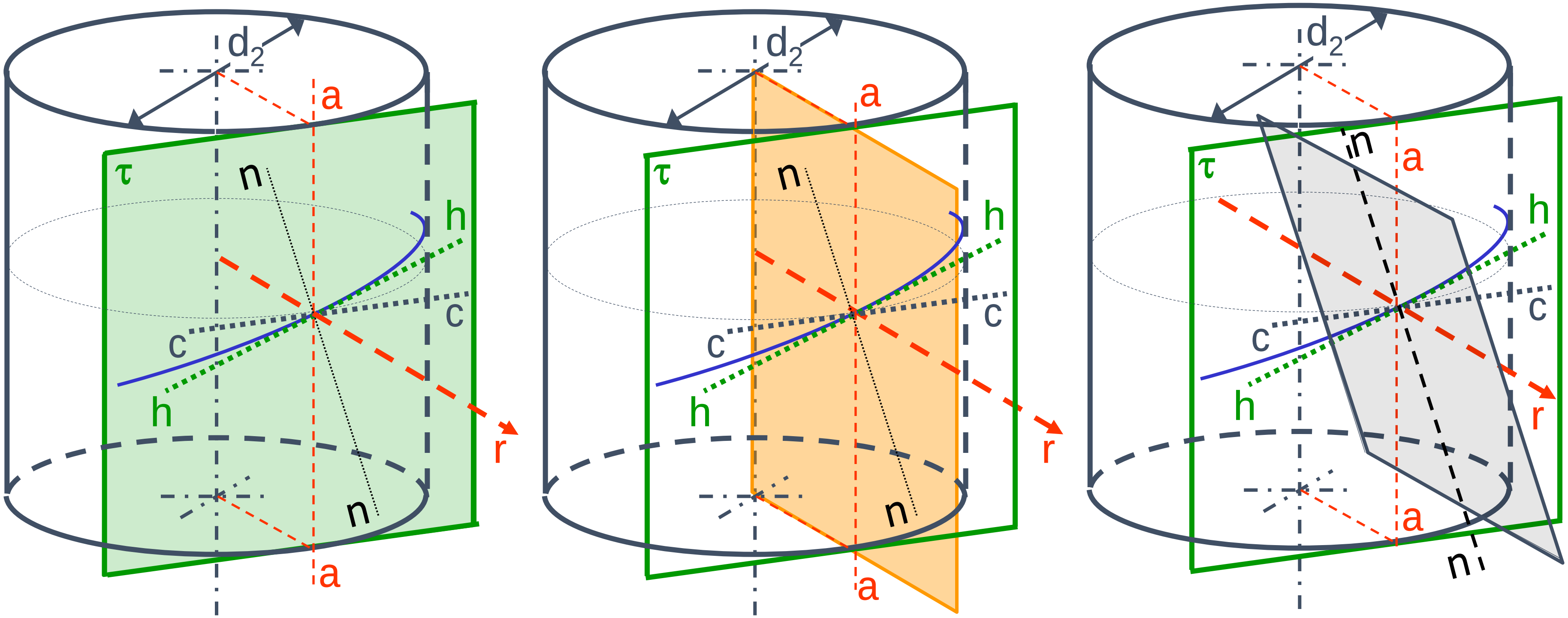
One of its points is crossed here by a radius r.

At this point the line h-h tangent to the helix is shown.

This line is inclined by the helix angle α to the "circumferential" line c-c (orthogonal the radius r and to axis a-a, i.e., intersection of the tangent plane with a plane orthogonal to the cylinder axis).

Line n-n on the τ tangent plane is normal to h-h , i.e. to the helix in the selected point.

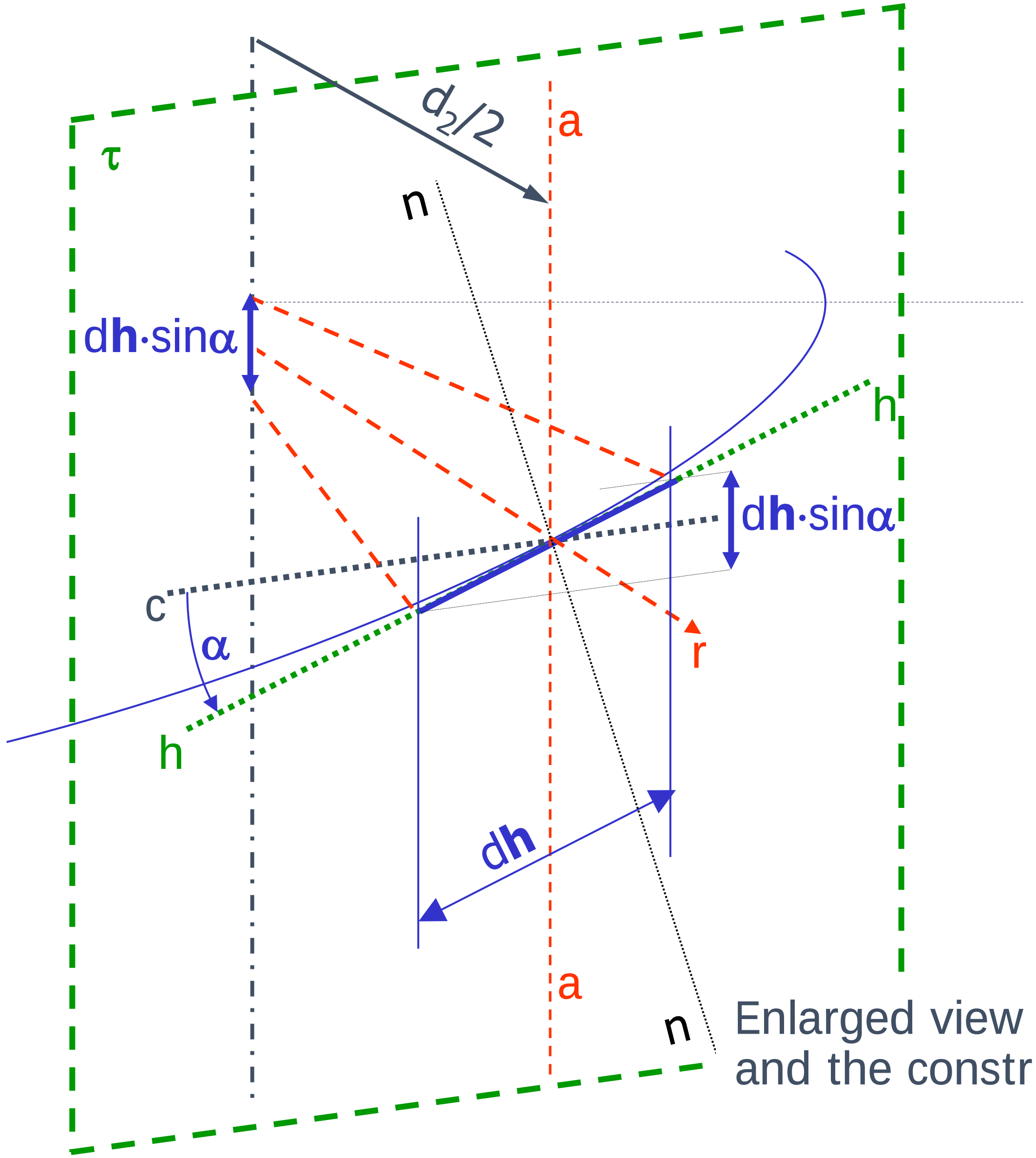
Three important planes are the here defined:

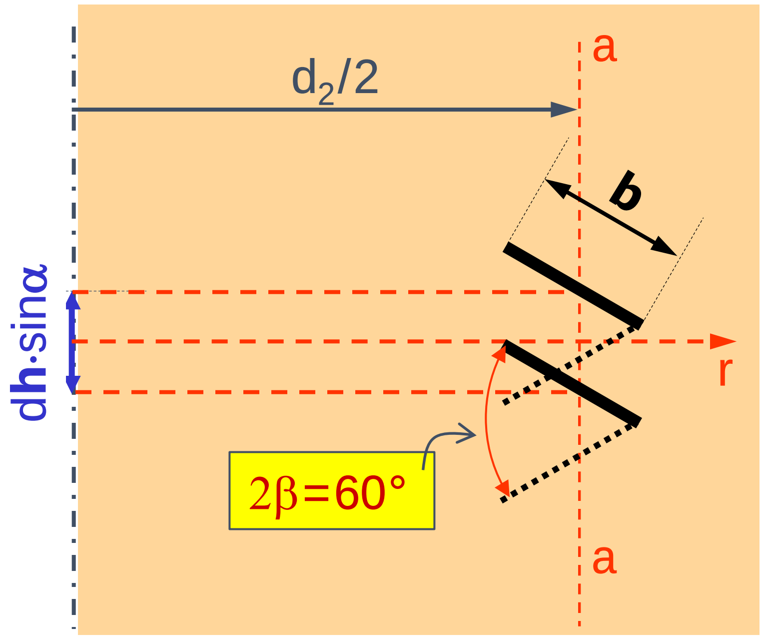


The τ **tangent plane**, containing a-a and h-h normal to r

The **radial plane**, containing a-a and r

The plane **normal to the helix**, containing r and n-n (start with the radial plane and rotate about r)

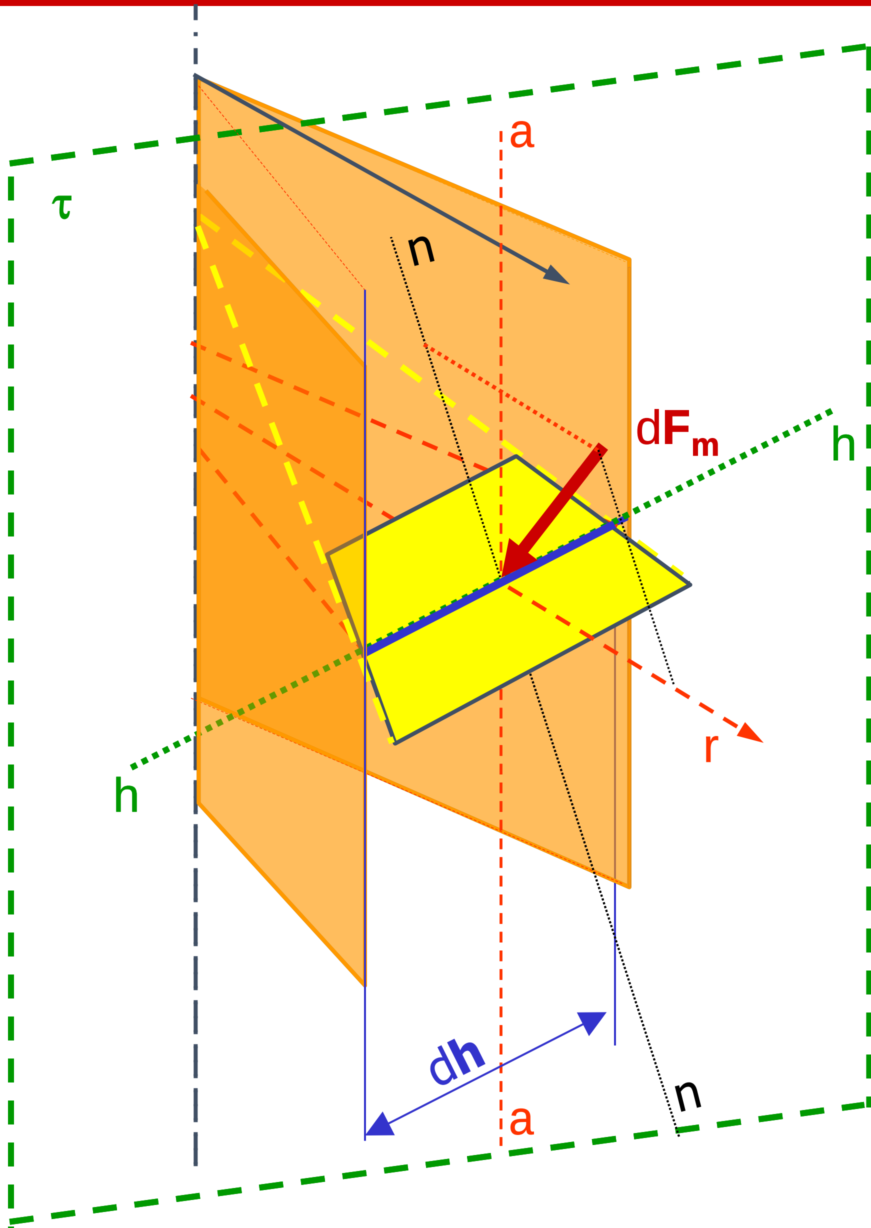


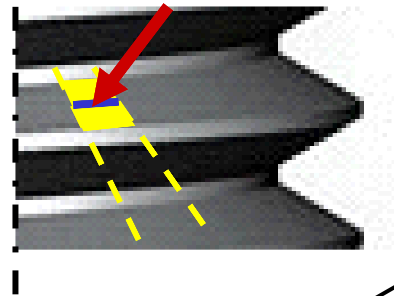


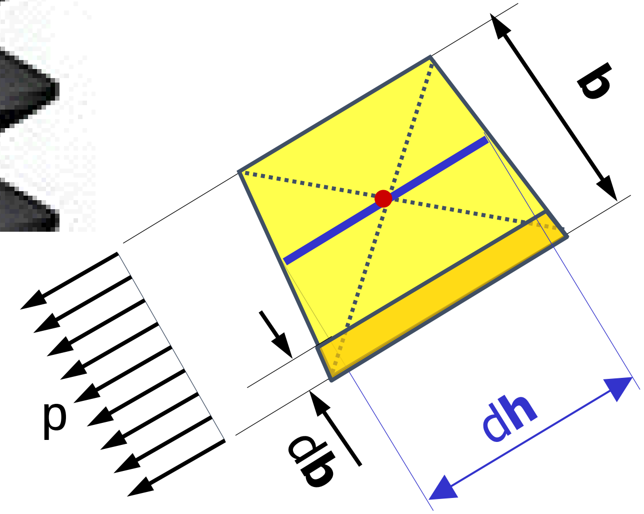
View in the radial plane of the projections of two thread flanks at a distance **dh** along the thread helix; the width of the flank is here named: **b**

Enlarged view of the infinitesimal helix arc **dh** and the construction lines related

6 - Bolt tightening: torque control formulas (7 / 19)







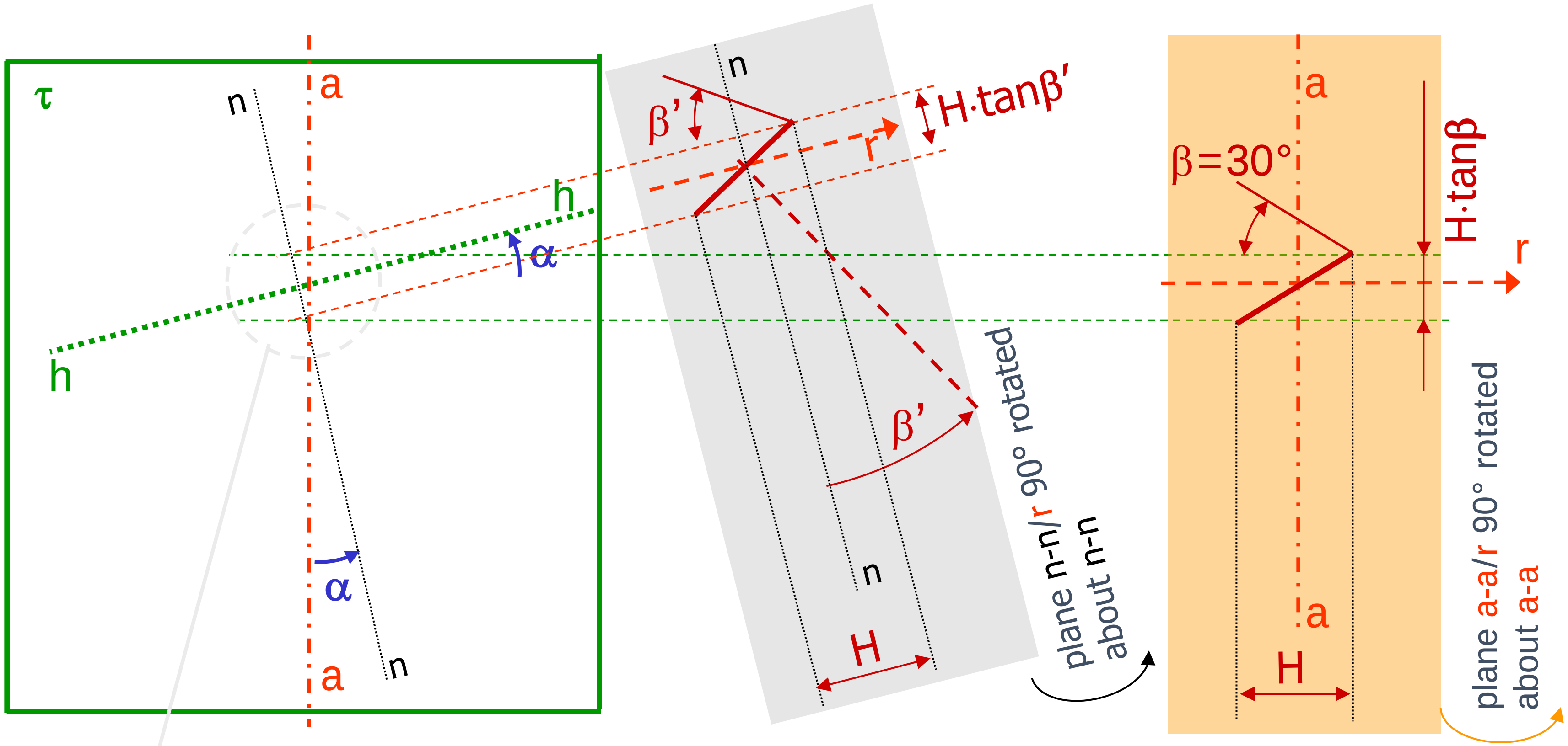
We assume that the normal pressure p is constant along the thread flank width **b** ; then the resultant force due to p applied to the flank's infinitesimal helix arc d**h** is:

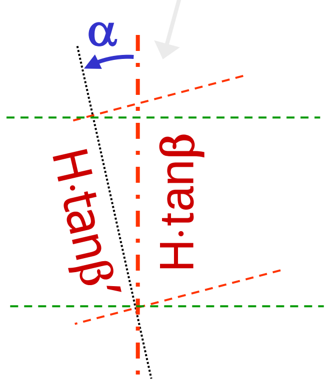
$d\mathbf{F_m} = p \cdot \mathbf{b} \cdot d\mathbf{h}$

normal to the flank surface and applied in its area centroid, approx. at the intersection of r and h-h , i.e., on the helix traced on the pitch cylinder.

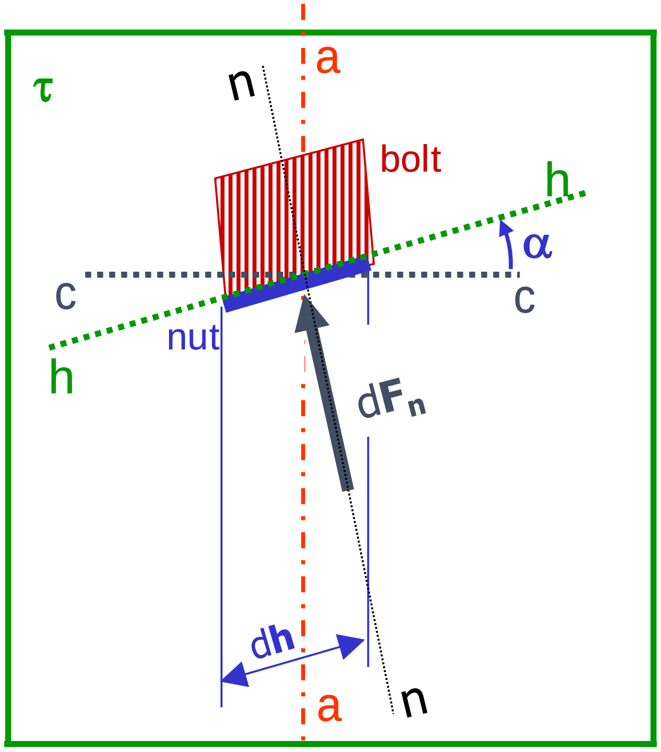
From now on we shall describe the normal force component on the flank as $d\mathbf{F_m}$ applied to the helix on the pitch cyinder.

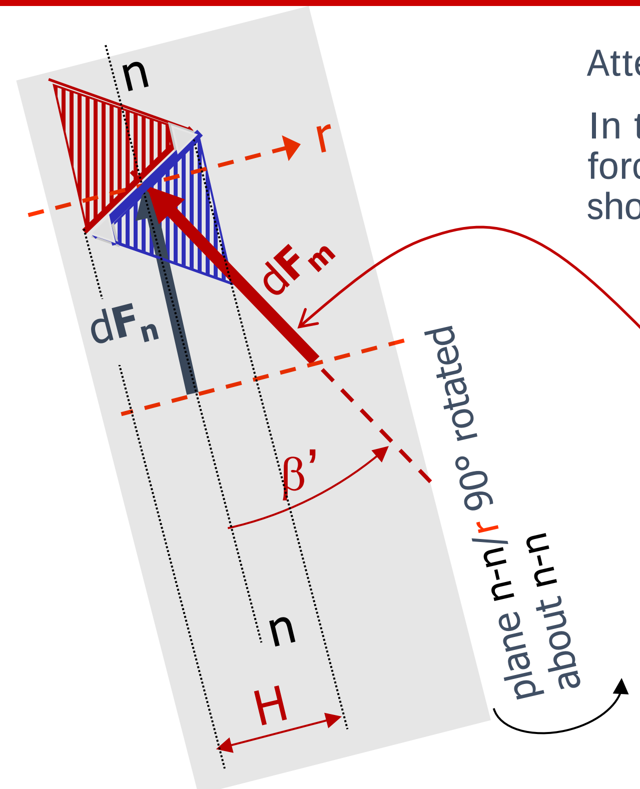
6 - Bolt tightening: torque control formulas (8 / 19)





The thread V-shape height, H , is seen both in the n-n/r plane and in the a-a/r plane. However, the projection along n-n is H·tanβ' , while the projection along a-a is H·tanβ , with β=30° for metric threads. Since H·tanβ = H·tanβ'/cosα , then: tanβ'=tanβ*cosα. However, with α in the range 1÷3 degrees, this correction is negligible. We shall then assume $\beta' \cong 30^\circ$.





Attention!!

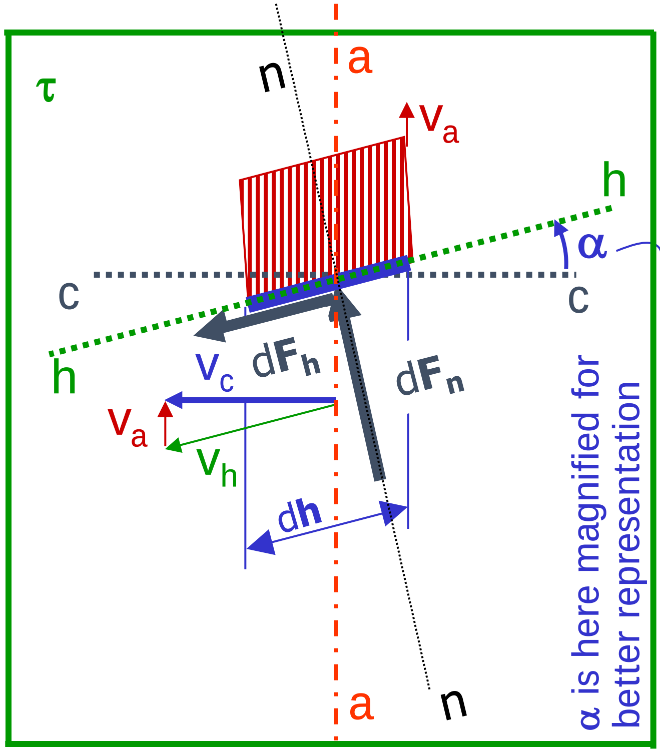
In this figure the nut-to-screw force is the opposite of the one shown in sl. 7 of this Section.

This is the normal force producing the tangential force !!!!

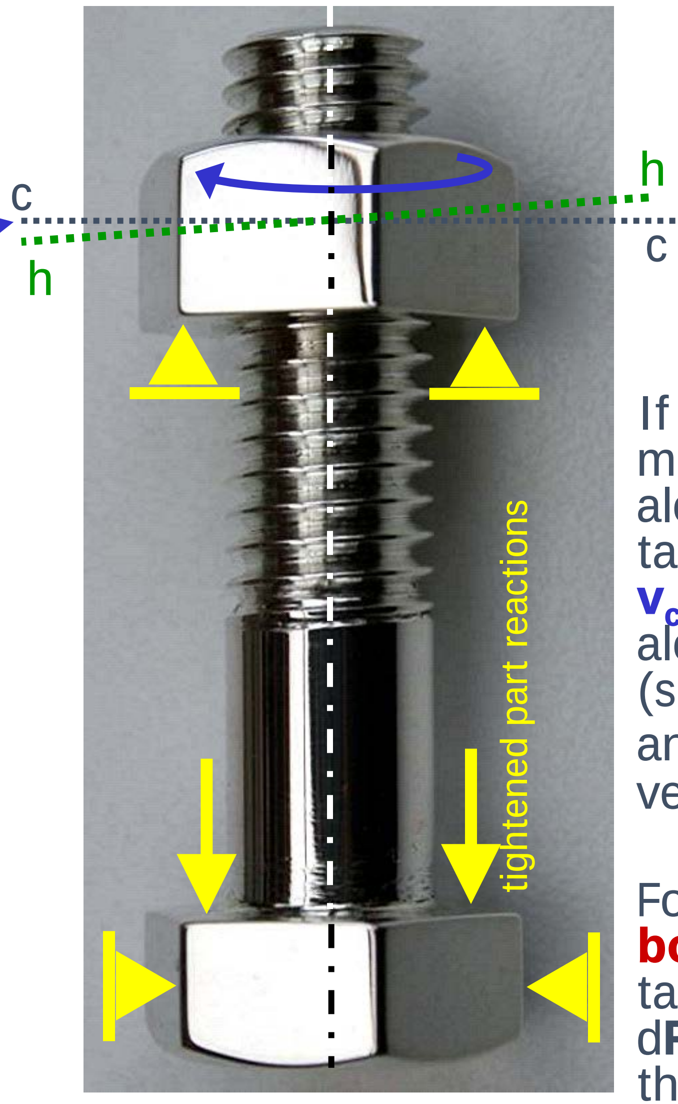
The figure on the left shows schematically an infinitesimal section of the **bolt** thread helix (upper) in contact with the **nut** (lower).

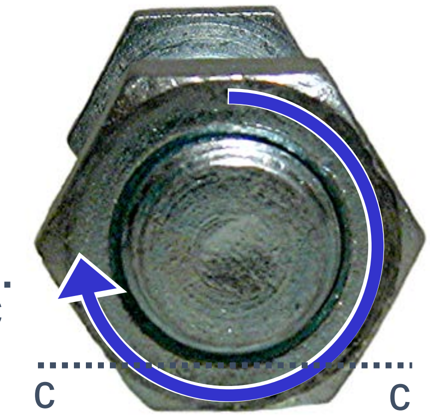
The auxiliary figure on the right, shows the intersection of the threads (nut and bolt) with the plane normal to the helix (containing n-n and r) after a right-wise 90° rotation about n-n . It is evident that force $d\mathbf{F}_n$ along n-n is just the component of the “total” normal force $d\mathbf{F}_m$ orthogonal to the thread flank.

The inclination is “almost” half the thread angle; for metric threads: $\beta' \cong 60° / 2 = 30°$



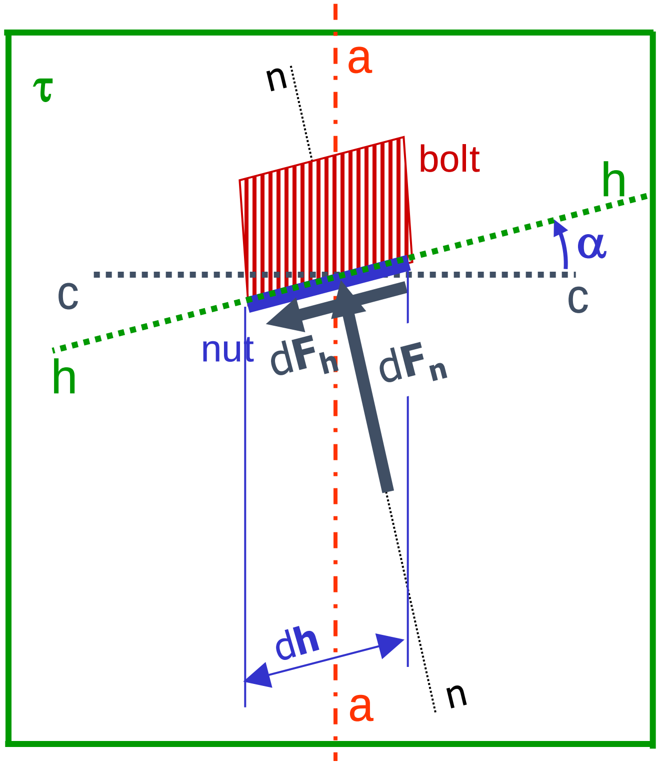
Tangential force on nut thread arc **dh** depends on the relative motion. If the bolt is kept fixed and the nut rotated, the (red) bolt thread element moves elastically up, velocity $\mathbf{v_a}$, the nut (blue) … thread element moves (rigidly) to the left, velocity $\mathbf{v_c}$.

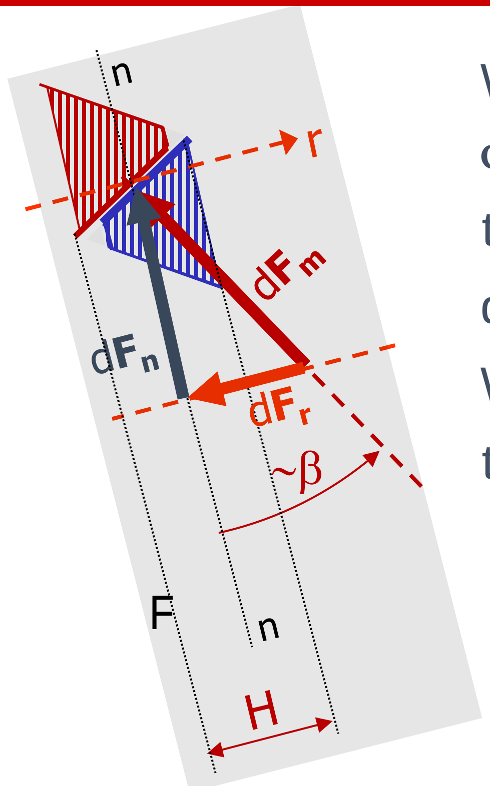




If the **nut**-to-**bolt** motion takes place along the common tangent h-h, the given $\mathbf{v_c}$ is the sum of a $\mathbf{v_h}$ along the thread (sliding velocity) plus an axial $\mathbf{v_a}$ (lifting velocity).

Force from **nut** to **bolt** has then the tangential component d**F**$_h$ along the helix in the same sense of $\mathbf{v_h}$.





With:

φ_G : thread friction angle

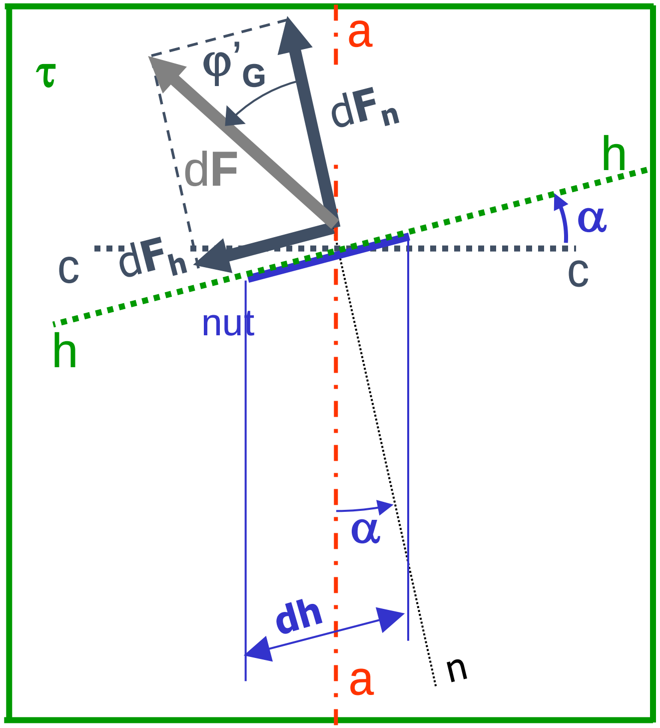
the tangential component:

$d\mathbf{F_h} = d\mathbf{F_m} \tan\varphi_G$

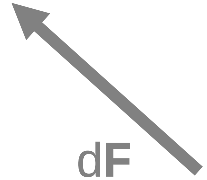
With: $d\mathbf{F_n} \cong d\mathbf{F_m} \cos\beta$

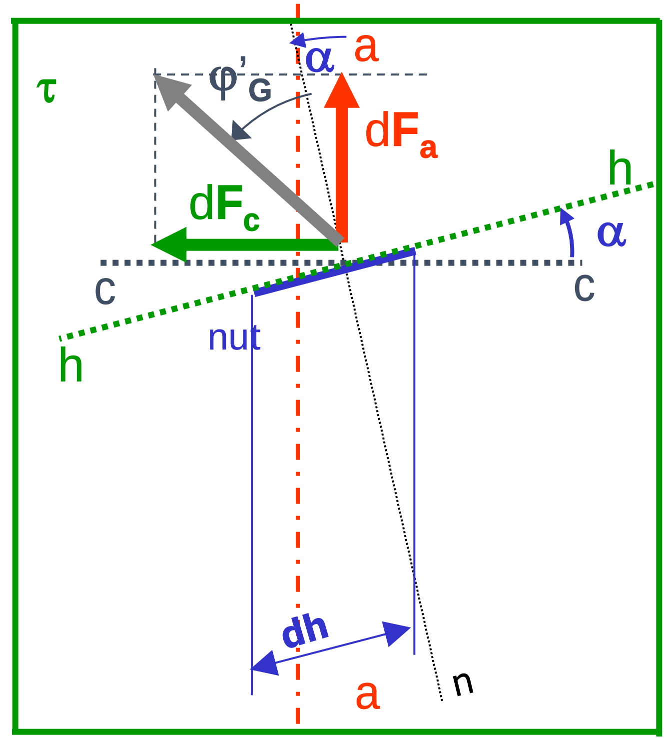
then: (*) $d\mathbf{F_h} = d\mathbf{F_n} \boxed{\dfrac{\tan\varphi_G}{\cos\beta}}$ → $\boxed{\mu'_G}$

The nut-to-bolt radial component component $d\mathbf{F_r}$ is usually neglected, being a radial compressive force that produces relatively low compressive stresses, moreover of a benign nature to fatigue. The two remaining contact force components $d\mathbf{F_n}$ and $d\mathbf{F_h}$, seen on the tangent plane, are linked by equation (*). The tangential (friction) force is linked to $d\mathbf{F_n}$ by a fictitious friction coefficient $\mu'_G \equiv \tan\varphi'_G = \tan\varphi_G/\cos\beta \equiv \mu_G/\cos\beta$.



Decomposition of the total in-plane force dF along the helix and its normal





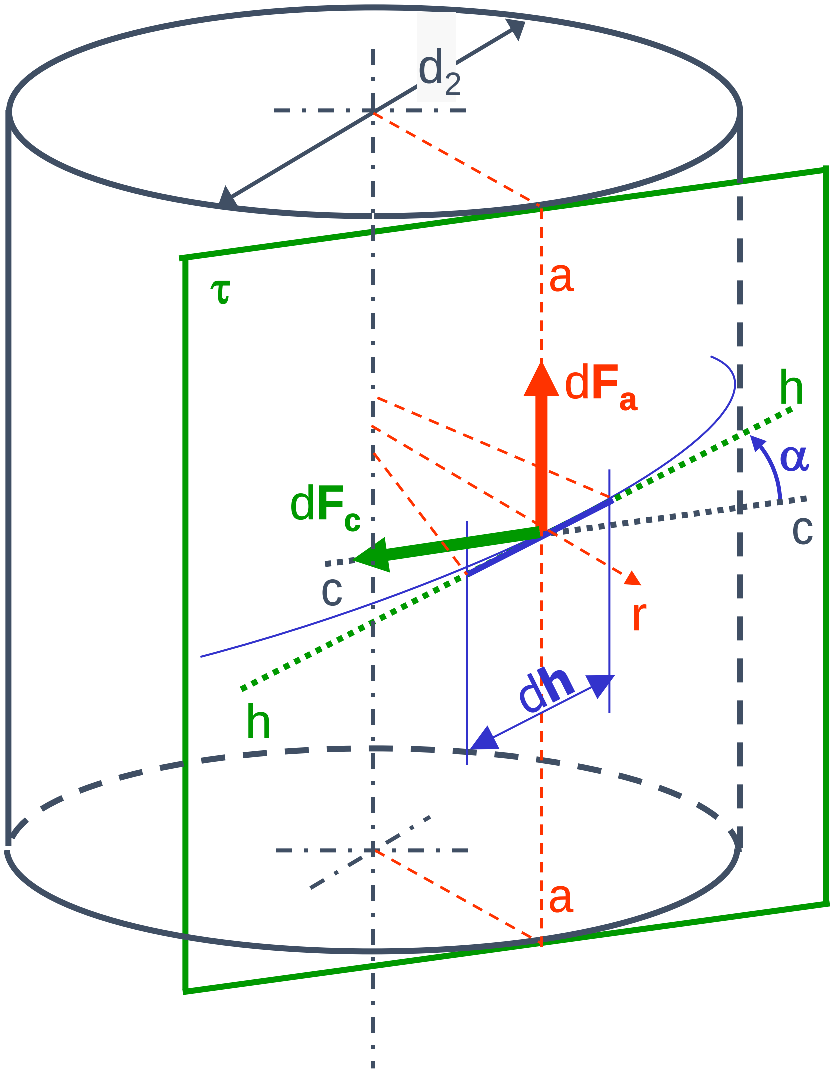
Decomposition of the total in-plane force dF along the bolt axis and along c-c

The total force dF on the plane is inclined of an angle $(\varphi'_G+\alpha)$ to the projection a-a of the cylinder axis. It is convenient to calculate:

dF_a = dF cos $(\varphi'_G+\alpha)$: axial component

dF_c = dF sin $(\varphi'_G+\alpha)$: circumferential component

$$dF_c = dF_a \tan(\varphi'_G+\alpha)$$



The axial and circumferential forces are not constant along the thread. Even without knowing their distribution, we can nevertheless calculate their resultants:

M_G : total torsion moment transmitted by the thread

F_a : total thread axial force

$$F_a = \int_{thread} dF_a$$

$$M_G = \int_{thread} dF_c \frac{d_2}{2} = \frac{d_2}{2} \int_{thread} dF_a \tan(\alpha + \varphi'_G)$$

$$= \frac{d_2}{2} \tan(\alpha + \varphi'_G) \int_{thread} dF_a$$

$$M_G = F_a \frac{d_2}{2} \tan(\alpha + \varphi'_G)$$

The formula: $M_G = F_a \dfrac{d_2}{2} \tan(\alpha + \varphi'_G)$

can be linearised because $\alpha \le \sim 0{,}06$ rad and $\varphi'_G \le \sim 0{,}22$.

$$\tan(\alpha + \varphi'_G) = \frac{\tan\alpha + \tan\varphi'_G}{1 - \tan\alpha \, \tan\varphi'_G}$$

example

$$\tan(0{,}06 + 0{,}22) = \frac{\tan 0{,}06 + \tan 0{,}22}{1 - \tan 0{,}06 \, \tan 0{,}22} = \frac{0{,}284}{1 - 0{,}013} = 0{,}288$$

$$M_G = F_a \frac{d_2}{2} \tan(\alpha + \varphi'_G) \cong F_a \frac{d_2}{2} (\tan\alpha + \tan\varphi'_G) = F_a \frac{d_2}{2} (\tan\alpha + \tan\varphi'_G)$$

with :

$\tan\varphi'_G = \tan\varphi_G / \cos 30°$; $\tan\alpha = P / \pi d_2$; $1 / \cos 30° \cong 1{,}155$

Then the “thread torque”:

$$M_G = F_a \frac{d_2}{2} \left[\frac{P}{\pi d_2} + 1{,}155 \, \mu_G \right] = F_a \left[0{,}16 \, P + 0{,}58 \, d_2 \, \mu_G \right]$$

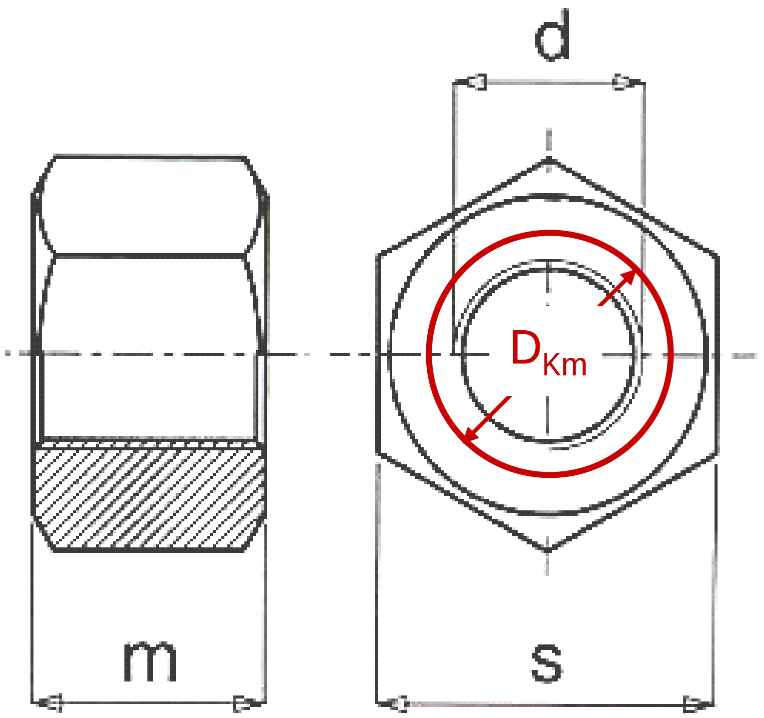
D_{Km} : effective diameter for the friction moment at the bolt head or nut bearing area

The calculation model assumes that the head or nut load (axial) is distributed on a circle at the diameter D_{Km} ; then the friction torsion moment about the bolt axis (head or nutface friction torque):

$$M_K = F_a \frac{D_{Km}}{2} \mu_K$$

with

$\mu_K = \tan \varphi_K$ friction coefficient (head or nut)



D_{Km} , effective diameter for the friction moment at the bolt head or nut bearing area,
is defined as the arithmetic mean of inner and outer effective contact circle on head or nut bearing area.

Friction coefficient class	Range for μ_G and μ_K	Selection of typical examples for	
		materials	lubricants
A	0,04 to 0,10	metallically bright black oxide phosphated galvanic coatings such as Zn, Zn/Fe, Zn/Ni Zinc laminated coatings	solid lubricants, such as MoS2, graphite, PTFE, PA, PE, PI in lubricating varnishes, as top coats or in pastes; liquefied wax wax dispersions
B	0,08 to 0,16	all the above plus Al and Mg alloys	all the above plus greases; oils; delivery state
		hot-galvanized	MoS2; graphite; wax dispersions
		organic coatings	with integrated solid lubricant or wax dispersion
		austenitic steel	

VDI 2230 Blatt 1 / Part 1 - 2003 - Friction coefficient classes with guide values for different materials/surfaces and lubrication states in bolted joints.
Values apply at room temperature. **For classes C, D, E** see the original of the standard.
The aim is to achieve coefficients of friction which fit into the friction coefficient class B in order to apply as high a preload as possible with low scatter.

During assembly, the axial bolt force F_a coincides with the assembly preload F_M (Montagevorspannkraft), so we just change its name.

The final formula for the assembly torsion moment or torque M_A to be applied to the bolt head or to the nut is specified through the torque-tension correlation formula:

$$M_A = F_M \left[\frac{d_2}{2} \left(\frac{p}{\pi d_2} + \frac{\mu_G}{cos(\beta')} \right) + \frac{D_{km}}{2} \mu_k \right]$$

with $tan(\beta') = tan(\beta) cos(\alpha)$

or, in engineer terms

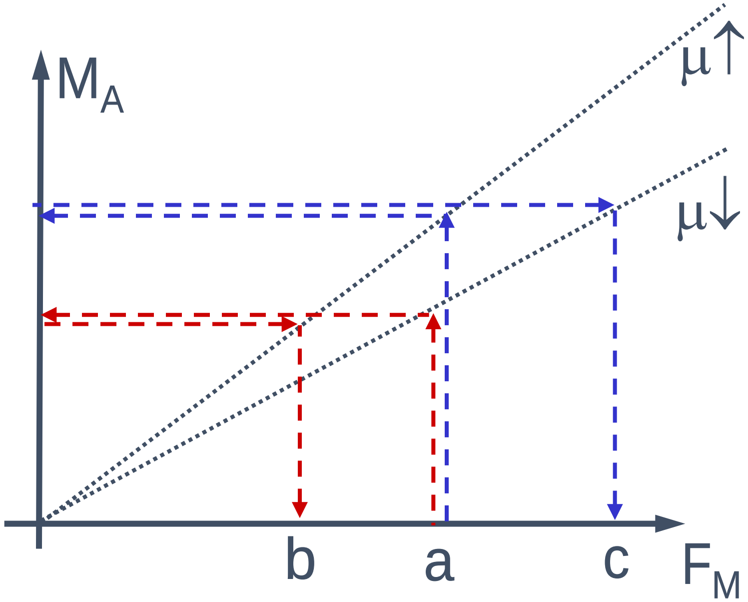
$$M_A = F_M \left[0{,}16\ P + 0{,}58\ d_2\ \mu_G + \frac{D_{Km}}{2} \mu_K \right]$$

Important !!! The value of M_A by means of the formula:

$$M_A = F_M \left[0{,}16p \ + 0{,}58\ d_2\ \mu_G + \frac{D_{Km}}{2} \mu_K \right]$$

must be determined using the lower values of the range for μ_G and μ_K !!!

The figure shows that if this is done, starting from a desired value "a" for F_M , one gets (red lines) a value for M_A , and if the real friction is higher the achieved F_M will be "b"<"a".
This is important to insure that the bolt, which is designed at "a" near yield, will not be overstressed. The inverse choice (blue path) is evidently at risk, producing "c">"a" .



In this way the value of F_M will be always not greater than a safe maximum (a, in the figure).

6 - Bolt tightening: torque control formulas (19/19)

A numerical example: Thread: M16 (coarse) Material class: 8.8

Pitch: $P=2$ mm Stress cross section $A_S = 157 \text{ mm}^2$

Thread friction: $\mu_G=0{,}14$ Head friction: $\mu_K=0{,}1$ Head friction dia.: $D_{Km}=20$ mm

Helix incl. at pitch dia.: $\alpha = 0{,}0433$ rad (2,48 °)

calculate from Sect. 3 sl.8

Bolt tension: $F_M = 79400$ N (at $\sigma_{eq}=90\% \; R_{p0.2}$)

Tightening torque: $M_A = 202$ Nm

$$M_A = 79400\left[0{,}16\,P + 0{,}58\,d_2\,\mu_G + \frac{D_{Km}}{2}\mu_K\right] = 202000\ [\ 0{,}12 + 0{,}48 + 0{,}40\ ]$$

Thread extension torque

Thread friction torque

Nut-face friction torque

Evidently, a fundamental problem with torque tightening is that because the majority of the torque is used to overcome friction (usually between 85% and 95% of the applied torque), slight variations in the frictional conditions can lead to large changes in the bolt preload.

Sections 5, 6, 7 - Bolt tightening

These sections deal with the problem of clamping bolts during the assembly process, and the consequences they have on the effective bolt clamping forces.

Section 5 examines different tightening processes, instruments and problems.

Section 6 develops in full detail the relation between the torque applied to the nut or to the bolt head during the torque-controlled tightening process and the bolt tension, or clamping force, which is produced. The role of friction as the major source of uncertainty is explained. A special attention is given to the way the max tightening torque is specified.

Section 7 draws conclusions about the tightening uncertainty, and gives criteria / coefficients to take it into account in the course of design.

The assembly torque M_A is specified in order to produce an assembly pre-load F_M capable of satisfying a number of conditions; however F_M cannot be unique, because it will vary between $F_{M\ max}$ and $F_{M\ min}$ min due to scatter in friction (see friction coefficient class, Sect. 6 sl. 13) and to ensuing uncertainties due to the tightening procedures and instrumentation (see Sect. 5).

The tightening factor α_A is related to the scatter of the assembly preload (see next slide):

$$\alpha_A = \frac{F_{Mmax}}{F_{Mmin}} \qquad \frac{1}{2}\frac{|\Delta F_M|}{F_{Mm}} = \frac{\alpha_A - 1}{\alpha_A + 1} \qquad \begin{cases} \Delta F_M = F_{max} - F_{min} \\ F_{Mm} = (F_{max} + F_{min})/2 \end{cases}$$

Note that the table in next slide shows only cases where a control and a calibration (or adjustment) is present are considered.

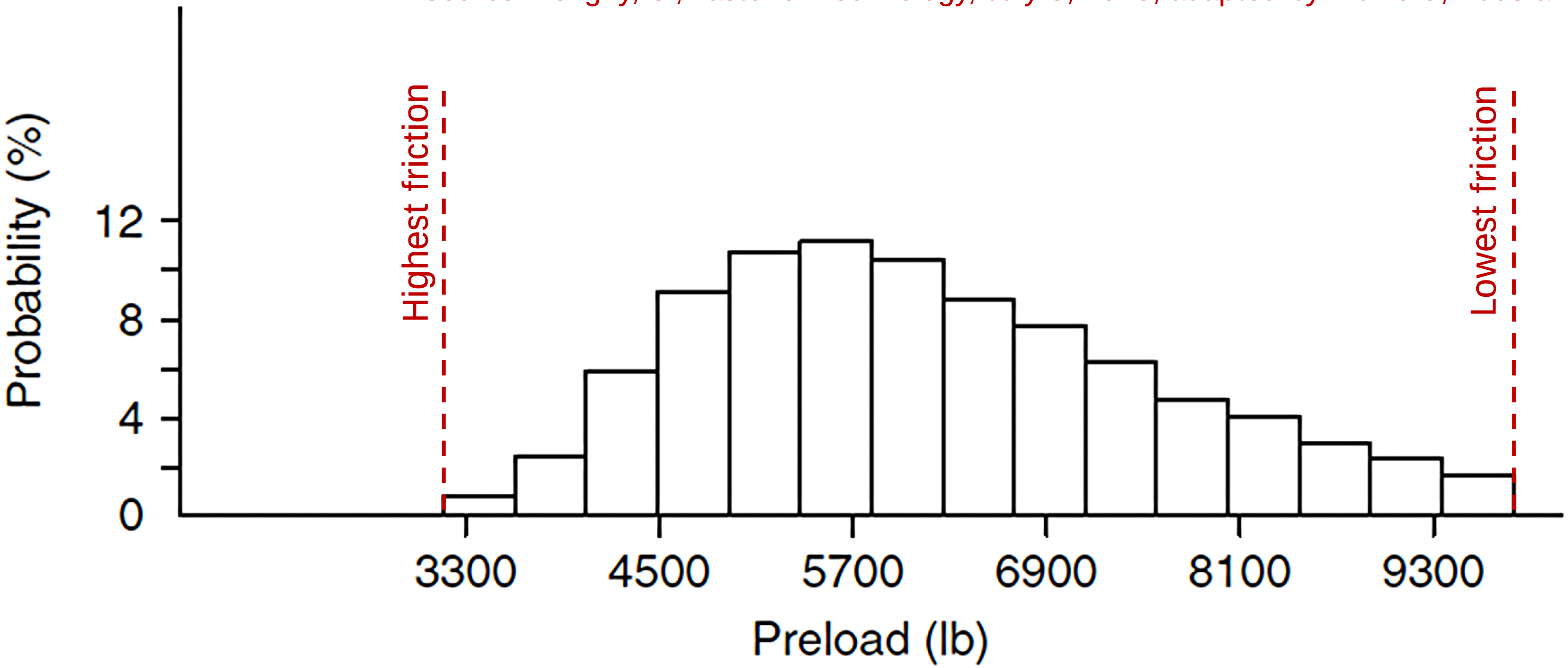
When these are not present, a tightening factor up to α_A=4 may be a wise assumption, however not entirely safe.

7 - Tightening uncertainty (2 / 3)

Tightening factor α_A	Scatter $\frac{1}{2}\frac{\lvert\Delta F_M\rvert}{F_{Mm}} = \frac{\alpha_A - 1}{\alpha_A + 1}$	Tightening technique	Adjusting technique
1,05 to 1,2	2% to 10%	Elongation-controlled tightening with ultrasound measurement	Echo time
1,1 to 1,5	5% to 20%	Elongation-controlled tightening with mech. measurement	Adjustment via longitudinal measurement
1,2 to 1,4	9% to 17%	Yield-controlled tightening, motor or manually operated	Input of the relative torque/rotation-angle coefficient
1,2 to 1,4	9% to 17%	Angle-controlled tightening, motor or manually operated	Experimental determ. of pre-tightening torque and angle of rotation
1,6 to 2,0 (friction coeff. class B) 1,7 to 2,5 (friction coeff. class A)	23% to 33% 26% to 43%	Torque-controlled tightening with torque wrench, indicating wrench, or precision tightening spindle with dynamic torque measurement	Determination of the required tightening torque by estimating the friction coefficient (surface and lubricating conditions)

VDI 2230 Blatt 1 / Part 1 - 2003 - Guide values for the tightening factor α_A (reduced and adapted)

Source: Eshghy, S., Fastener Technology, July 8, 1978, adapted by Bickford, 2003 and MG



Preload scatter after torque control tightening.

This figure shows initial preload results for a large number of fasteners, 5/16–24, SAE Grade 8 fasteners with a 2.3 in. grip length tightened to a uniform torque level. The distribution of preload achieved for a given torque is skewed right in this case. The preload can range approximately from -21% to +35% of the mean.

My comment: this was a very bad case, hovewer it can happen!

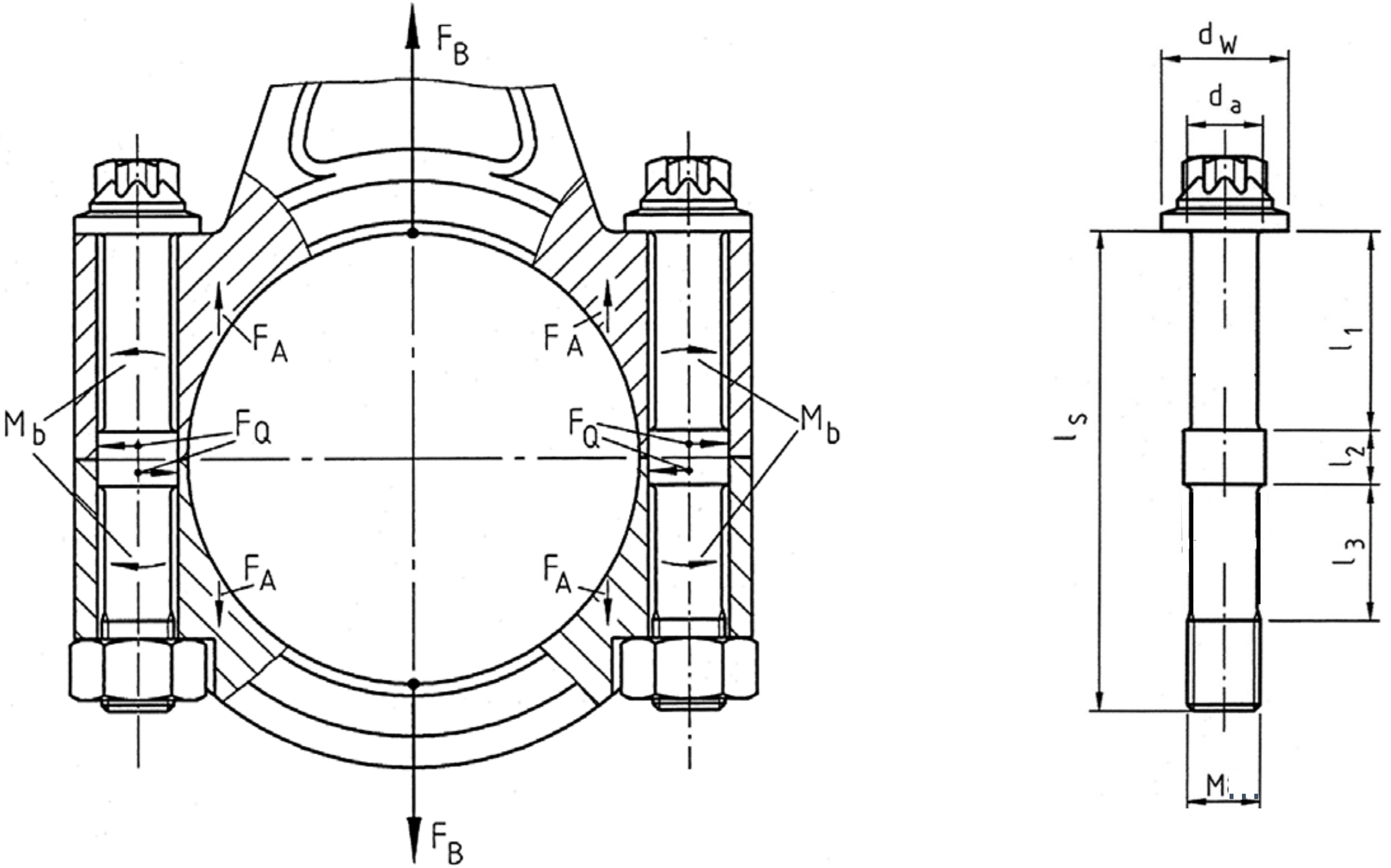
Sections 8, 9, 10, 11, 12 - Bolt clamping

Section 8 introduces the reader to the basic mechanics of bolt clamping and the related interplay between bolt and part axial compliances. It introduces the “interference diagram”.

Sections 9 and 10 give ideas and formulas respectively to calculate axial resilience, or deformability or compliance, of the bolt and parts it clamps.

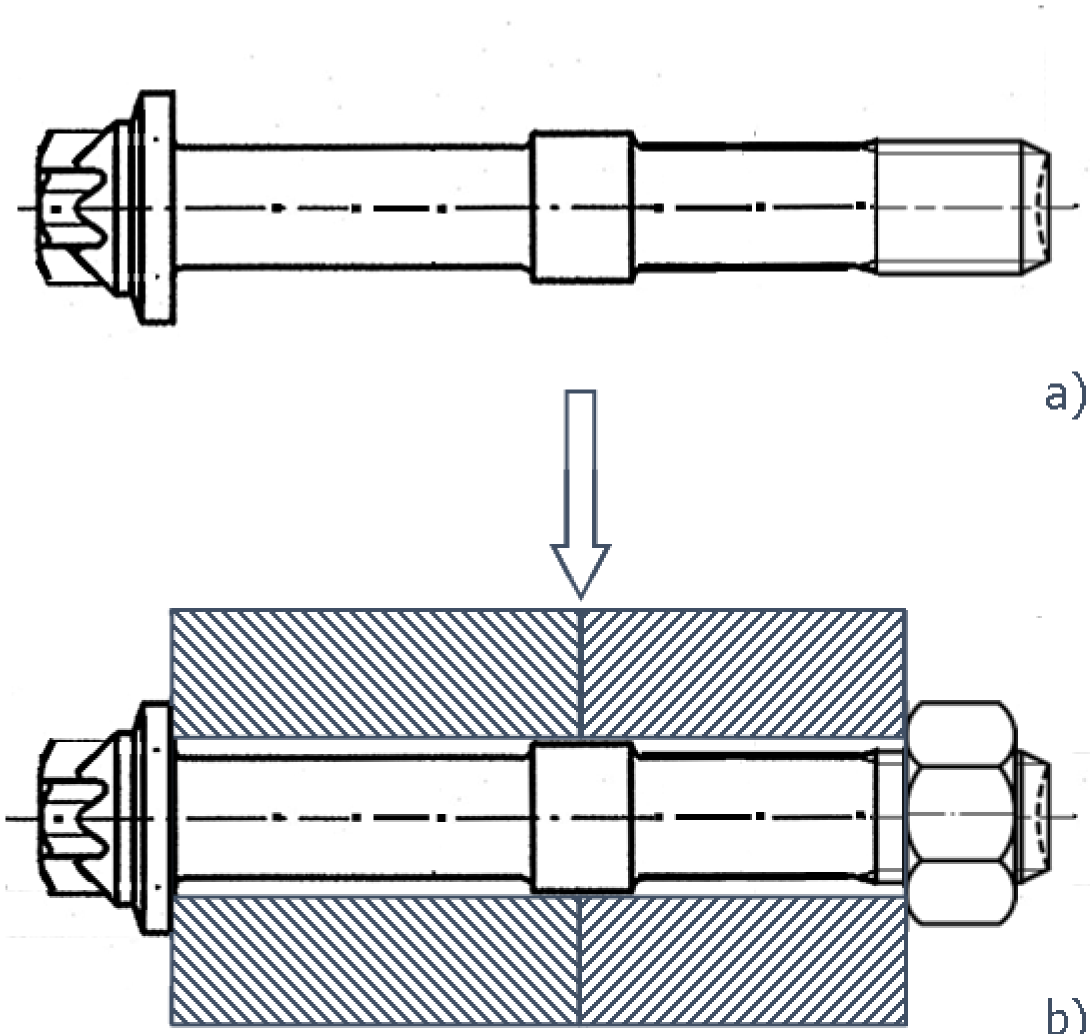
Section 11 introduced the string model, as a tool which can reduce the complexity of the treatment of bolt / part interaction. The phenomena occurring during assembly are seen on the interference diagram. Which proves to be a powerful although elementary graphical tool.

Section 12 shows how the external force applied to the serrated parts during operation is shared between bolt and parts; how this is represented on the interference diagram for the purposes of mechanical design.



VDI Richtlinien - VDI 2230 Blatt 1 / Part 1 – 2003

This figure shows the very special case of a connecting rod of an engine where the rod big-end and the cap are coupled by two reduced-shank bolts, which in the middle have a calibrated section which fits exactly into the through hole in order to prevent any possible transversal sliding of one tightened part against the other. It is probably one of the most elaborated type of bolt, which will now be used to introduce bolt calculations.



a)

b)

Concentrical joint

In the most elementary model, the bolt-part assembly considers the bolt in a) (here in the special form of a connecting rod bolt, which is probably one of the most complex shapes) clamping parts, like in b), which have ideal features seldom realised in practice:

- the parts are symmetrical to the bolt axis
- the loads are such that their resultants act along the bolt axis

This model is in most cases just a first approximation to the problem.

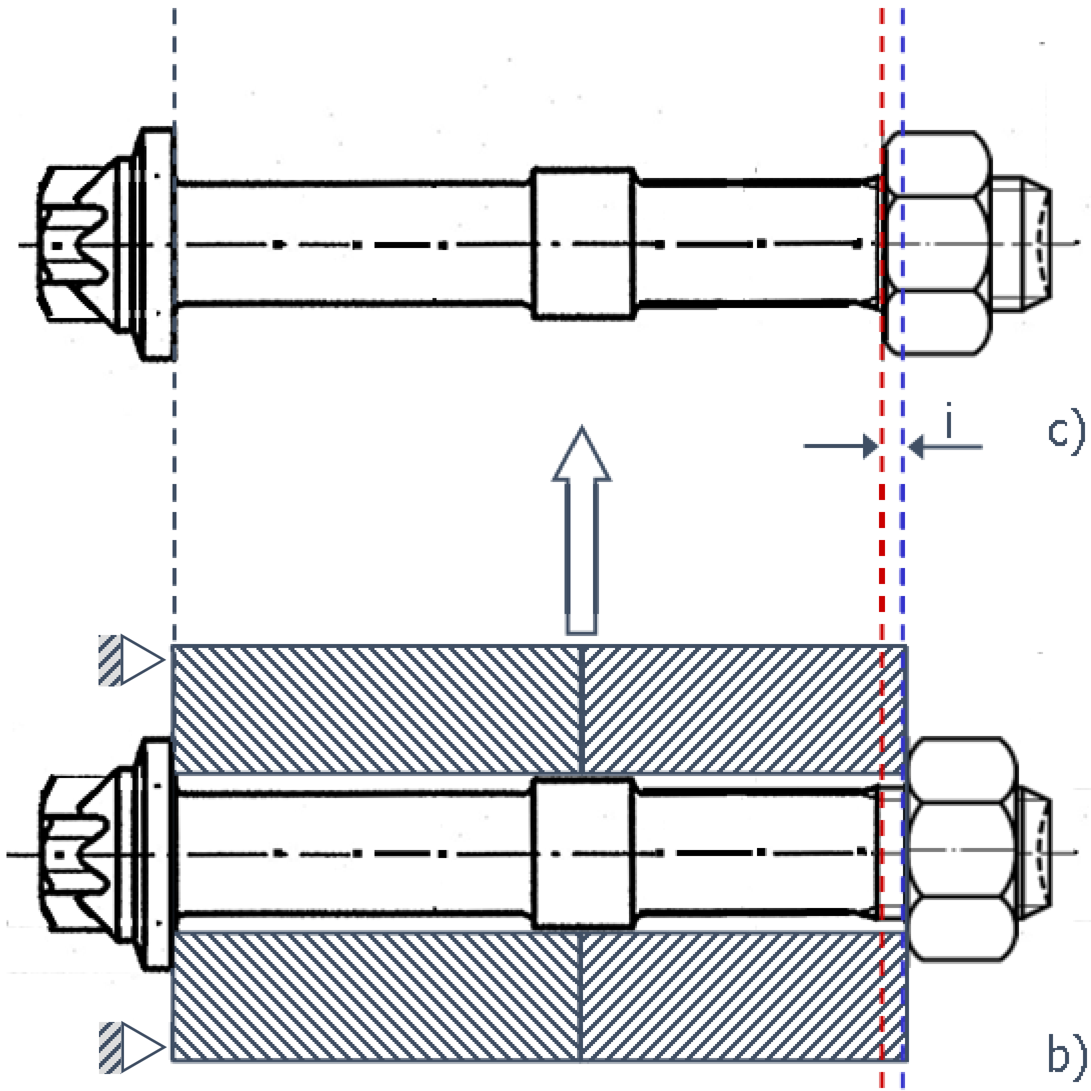


Figure b) showed the bolt head and the nut which just touched at zero load the two end surfaces of the clamped parts (head and nut in initial contact with parts, blue line).

At this stage either the bolt or the nut is rotated, producing in both cases a decrease of their inner distance.

The red line shown in c) is the position under zero load, i.e., if the parts to be clamped were not present, the inner head-to-nut distance would decrease of an amount “i”, interference”.

Please note that torsion in the bolt, is not considered because it does not produce any axial length variation!

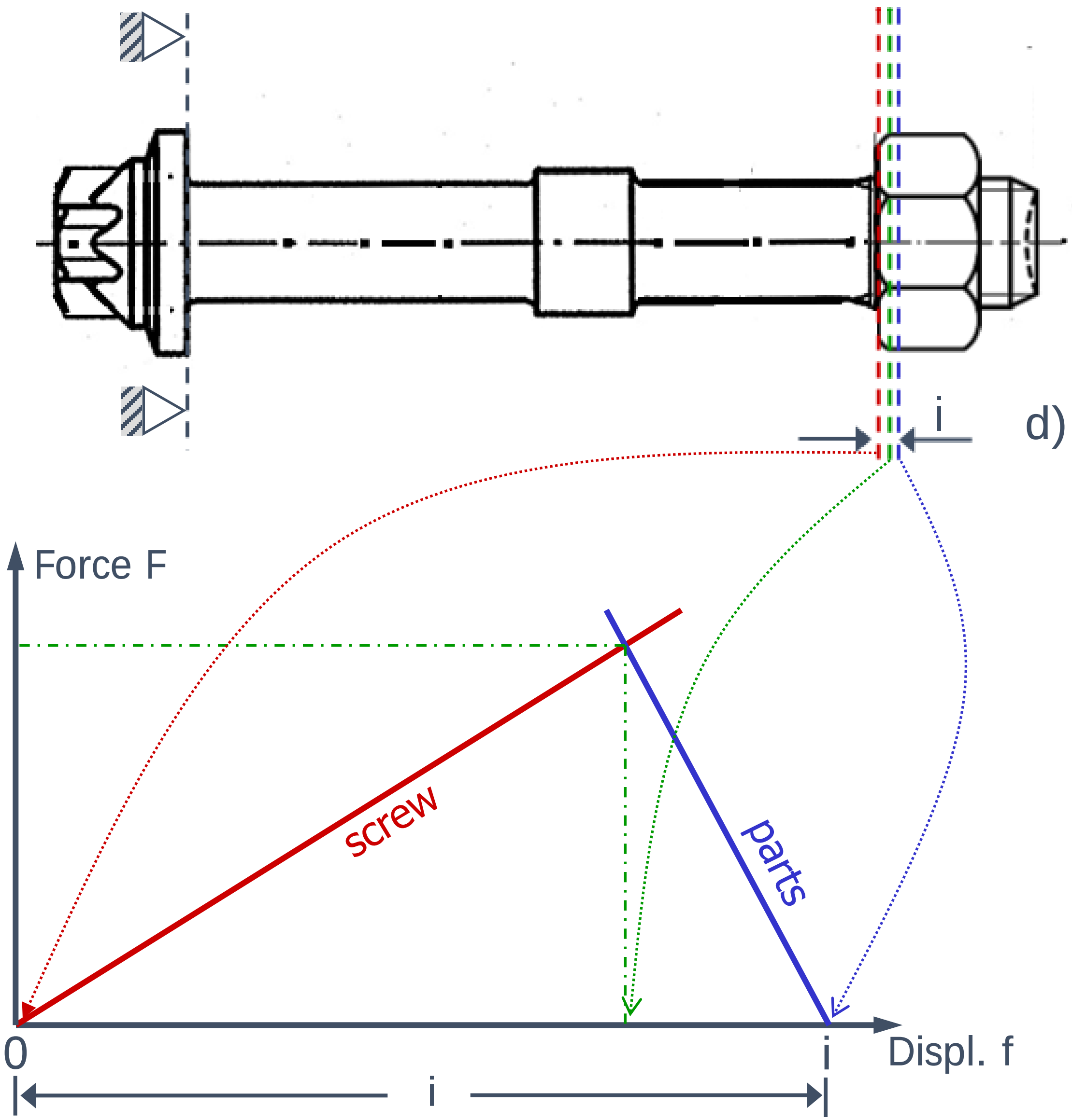
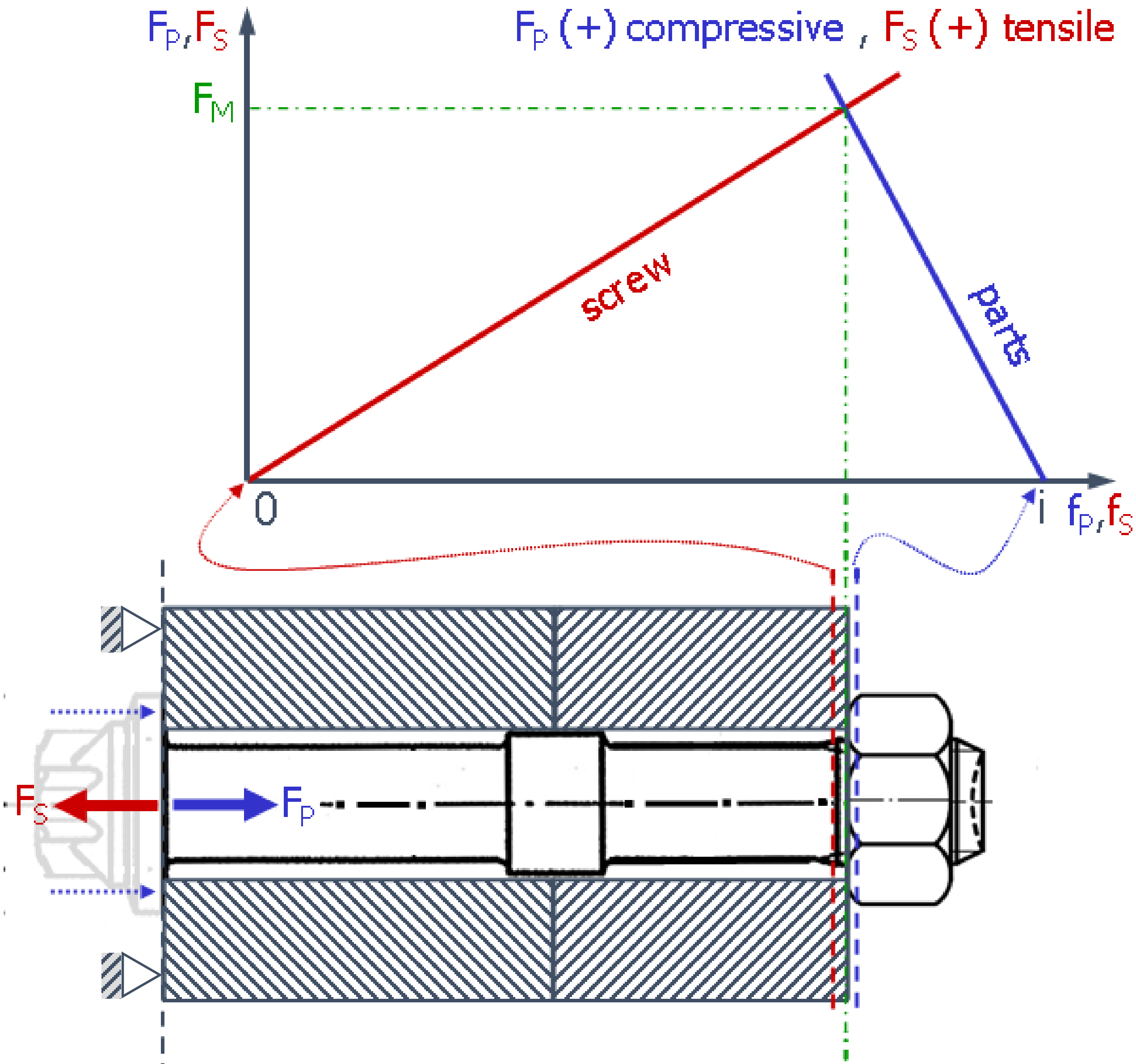


Figure d) indicates the final position (green line) reached by the nut (here the head is fixed – consider that the relative motion is what matters) due to the fact that clamped parts are compressed.

The “interference diagram” or “joint diagram” on the left expands “i” to a visible scale.

0 : nut face position after nut rotation but without clamping axial load

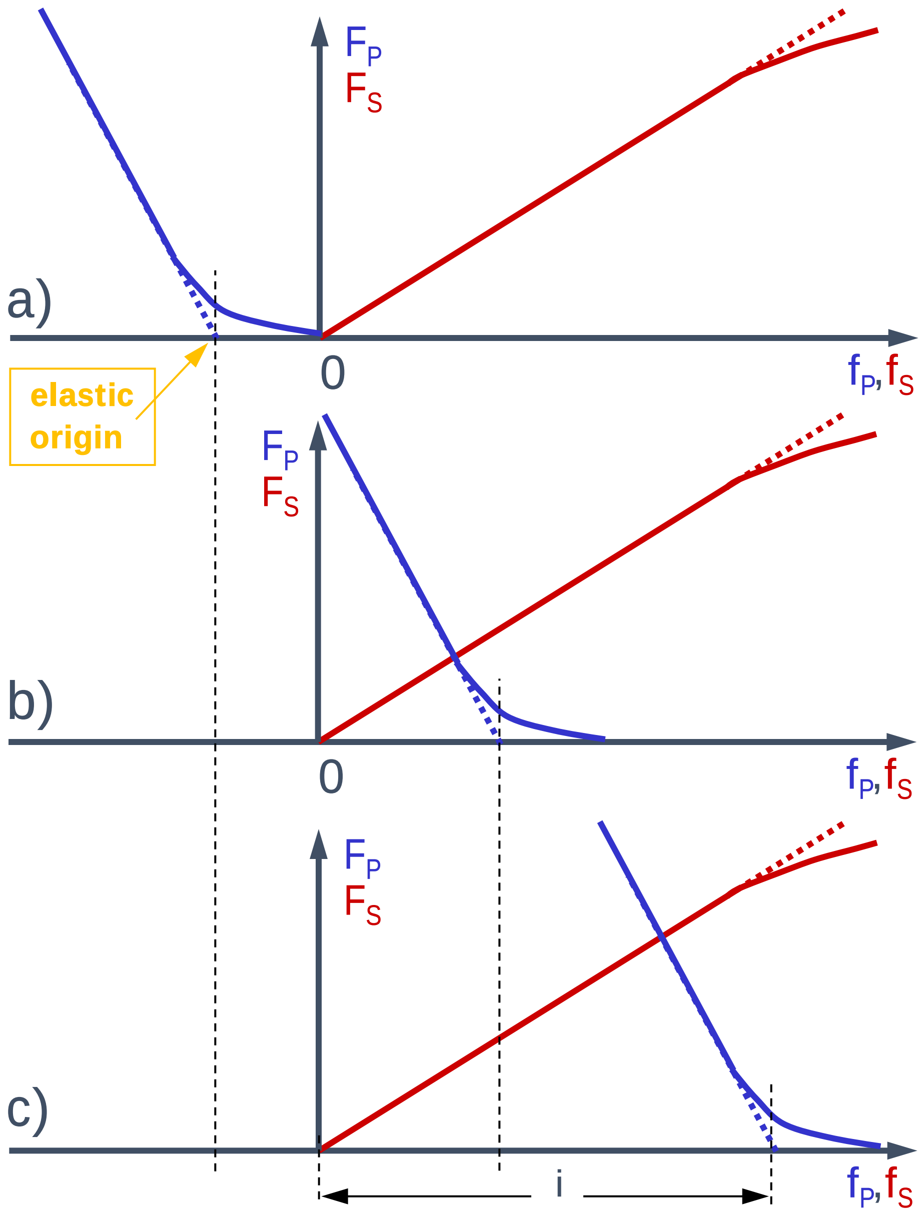
i : part face position in the relaxed state



The resilience $\delta_S = f_S/F_S$ of the bolt takes into account not only its elastic deformation within the clamp length but also any elastic deformations which occur outside this region and also have an effect on the deformation behaviour of the bolt in the joint. In this section only axial resilience is taken into account. The calculation is quite straightforward.
(see Section 9)

The calculation of the elastic resilience $\delta_P = f_P/F_P$ of the parts preloaded by the bolt, also designated as plate resilience, proves to be difficult on account of the three-dimensional stress-and-strain which forms when preload is applied.
(see Section 10)

8 - Introduction to clamping (6/6)



These figures show the progression of clamping from a) the first contact between head/nut underfaces and part, and therefore put into evidence how clamping is generated during the turnng of head or nut.

Moreover they link the linear approach to clamping of this Section to non-linearities shown in Sect. 5.

The **part curve** has the linear tract described in Sect. 10 “part resilience”, the **elastic origin** and the alignement zone (non-linear) which in Sect. 5 sl. 7 was attributed to the bolt. The **bolt curve** has the linear tract “bolt resilience” described in Sect. 9, and the terminal non-linear tract due to bolt yielding.

Fig. b) shows the relative position of part and bolt curves when, for instance, one full turn has been given to the nut and therefore the bolt clamped length has been reduced by one pitch P. It is preferred here to represent the bolt as fixed.

Fig. c) shows bolt and part curves after two turns. The distance between part and bolt elastic origins is the (elastic) interference “i” .

In general, with:

n_i : number of nut/head turns

α_i : nut/head rotation angle

P : thread pitch

$$\Rightarrow \quad \boxed{i = n_i P = \frac{\alpha_i}{2\pi} P}$$

Sections 8, 9, 10, 11, 12 - Bolt clamping

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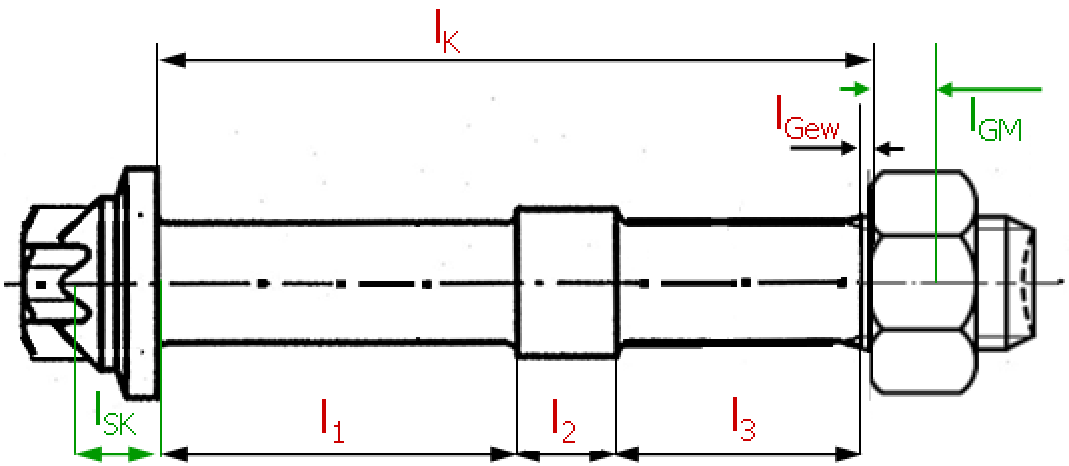

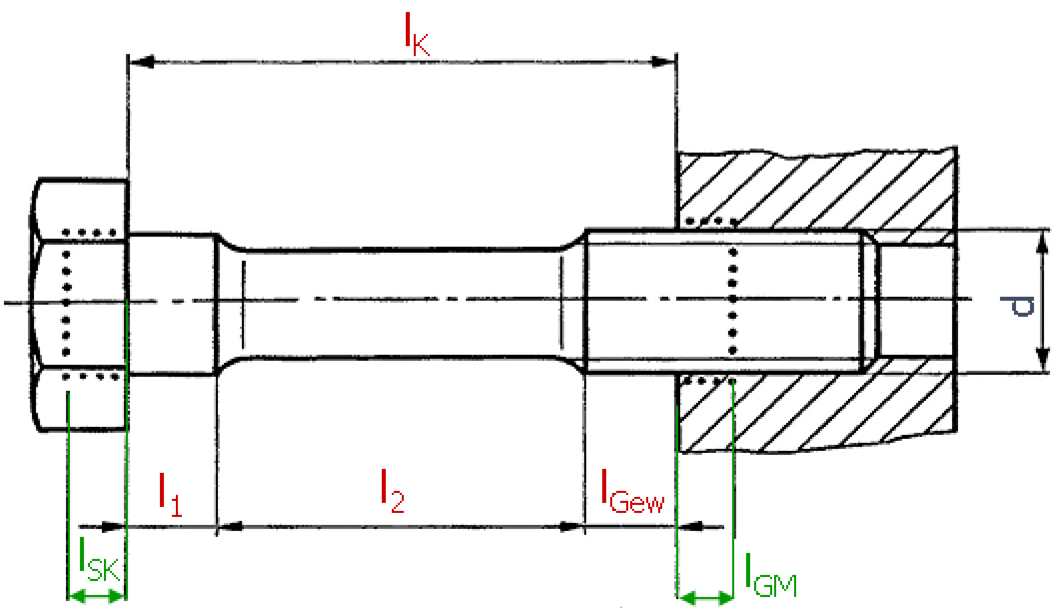

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The bolt consists of a number of individual elements which can be considered cylindrical bodies of length l_i and cross section A_i , having resilience:

$$\delta_i = \frac{l_i}{E_S A_i}$$

The cylindrical elements are arranged in series, the total elastic resilience δ_S is the sum of all resiliences within the clamp lengh

$$\delta_1 + \delta_2 + \ \ldots + \delta_{Gew}$$

plus those of deformation regions outside clamp length l_K:

$$\delta_{SK} + \delta_{GM}$$

Then:

$$\delta_S = \delta_{SK} + \delta_1 + \delta_2 + \ \ldots + \delta_{Gew} + \\ + \ \delta_{GM}$$

δ_{SK}: elastic resilience of the bolt (**S**chraube) head (**K**opf)

Standardized hexagon head bolts and hexagon socket screws:

$$\delta_{SK} = \frac{l_{SK}}{E_S A_N} \quad \begin{cases} A_N : \text{nominal cross section} = \dfrac{\pi}{4} d^2 \\ l_{SK} : 0{,}5\ d \text{ for hexagon head bolts} \\ l_{SK} : 0{,}4\ d \text{ for socket cap screws} \end{cases}$$

δ_{GM}: elastic resilience of the engaged thread (**G**ewinde) and of the nut or tapped thread region (**M**utter)

$\delta_{GM} = \delta_G + \delta_M$ with: δ_G : elastic resilience of the engaged thread

$$\delta_G = \frac{l_G}{E_S A_{d_3}}; \quad l_G = 0{,}5d \quad ; \quad A_{d_3} = \frac{\pi}{4} d_3^2$$

δ_M : elastic resilience of nut or tapped thread region

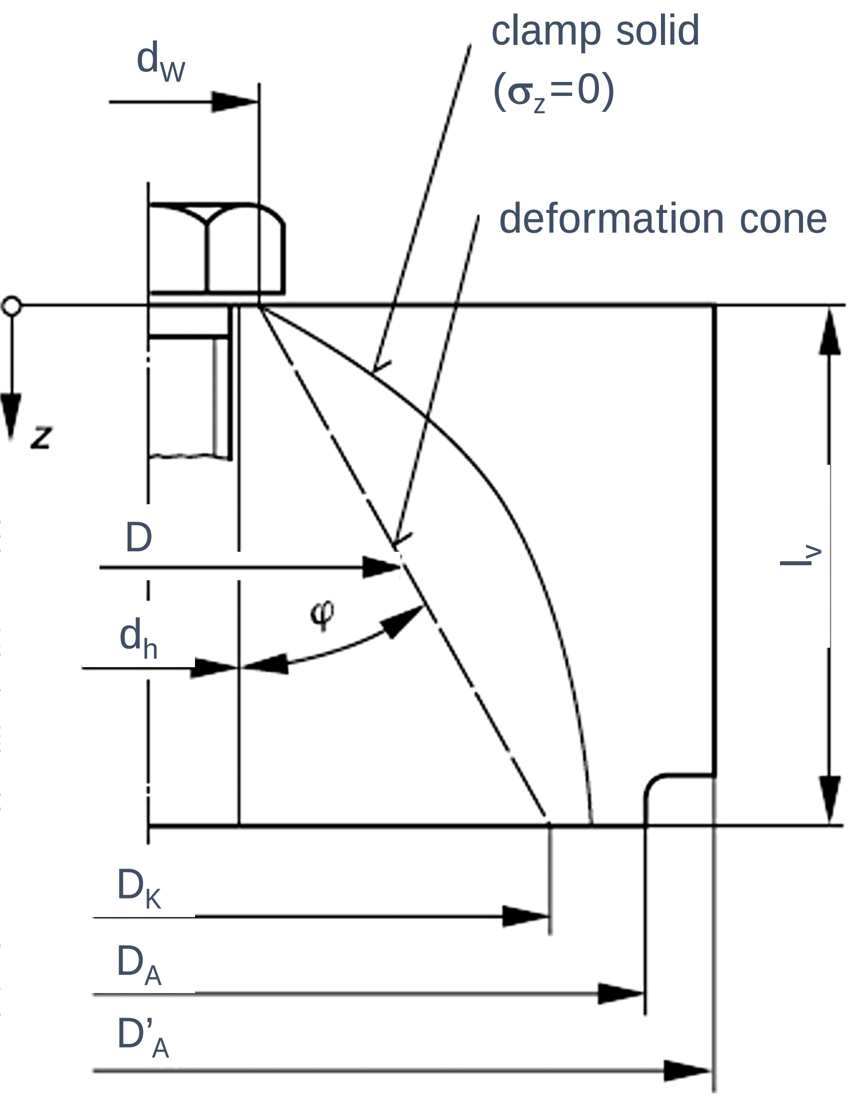
$$\delta_M = \frac{l_M}{E_x A_N}$$

$l_M = 0{,}4\ d$ for bolted joints

$l_M = 0{,}33\ d$ for tapped thread joints

$E_x =$ nut or tapped thread Young's modulus

$$\delta_{Gew} = \frac{l_{Gew}}{E_x A_{d3}}$$



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Stresses in clamped part affect a volume which widens moving away from the bolt head or nut. Calculation of part resilience is done on a “substitution” cone having the same resilience.

The contact (surface) resilience is not taken into account; for head / nut / washer contact it is recovered while turning the head/nut. If the clamped parts are a large number of thin sheets, which are never fully flat, surface effects must be determined experimentally case by case.

D(z) : diameter of cone

$$D = d_w + 2\ z \tan\varphi$$

A(z) : cone cross-section area

$$A(z) = \frac{\pi}{4}\left[\left(d_w + 2z\ \tan\varphi\right)^2 - d_h^2\right]$$

The cone resilience is:

$$\delta_P = \frac{1}{E_P}\int_{z=0}^{z=l_v}\frac{dz}{A(z)} = \frac{4}{\pi E_P}\int_{z=0}^{z=l_v}\frac{dz}{[(d_w + 2z\tan\varphi)^2 - d_h^2]} =$$

$$\delta_P = \frac{4}{\pi E_P}\int_{z=d_w}^{z=2l_v\tan\varphi}\frac{dx}{[x^2 - d_h^2]} \quad \text{with: } d_w + 2z\tan\varphi = x$$

$$\text{with the known: } \int\frac{dy}{1-y^2} = \frac{1}{2}\ln\left|\frac{1+y}{1-y}\right| + c$$

then :

$$\delta_P = \frac{1}{\pi E_P d_h \tan\varphi}\left[\ln\left(\frac{(d_w + d_h)(d_w + 2l_v\tan\varphi - d_h)}{(d_w - d_h)(d_w + 2l_v\tan\varphi + d_h)}\right)\right]$$

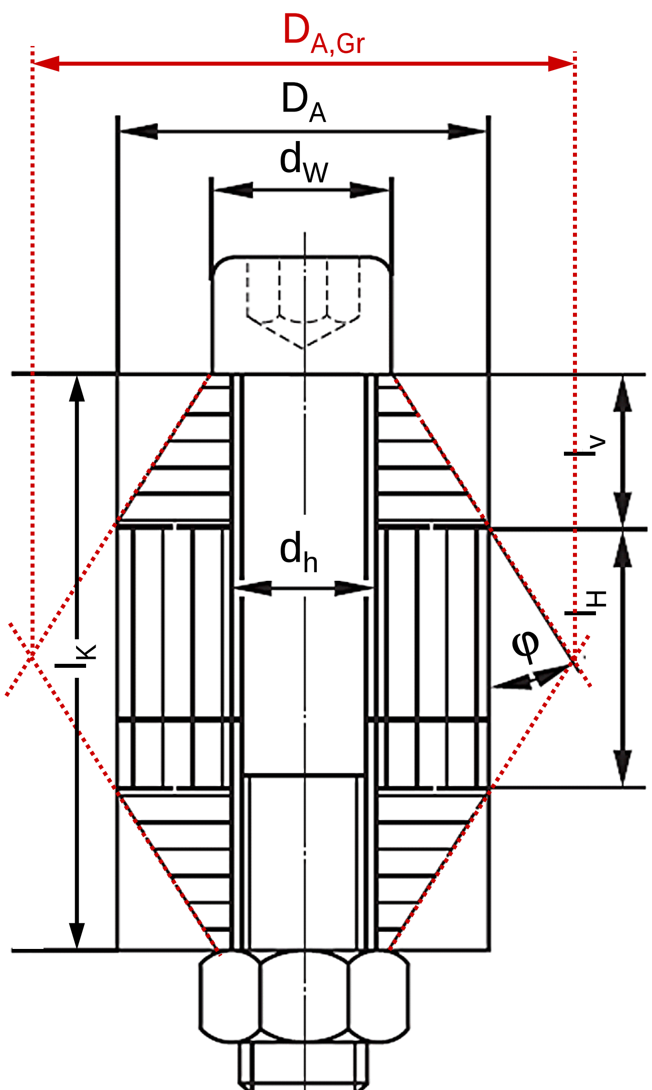
A limit (**Gr**enze) diameter of cone intersection $D_{A,Gr}$ needs to be defined to settle the question as to whether the deformation cone is present

$$D_{A,Gr} = d_w + w \cdot l_k \cdot \tan(\varphi)$$

where

$w = 1$ for bolted joints,

$w = 2$ for tapped thread joints.



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If the cylindrical part has a diameter $D_A \leq D_{A,Gr}$, where:

$D_{A,Gr}$: limit (Grenze) diameter, of cone intersection

then the part is composed of two cones and a central cylindrical sleeve (Hülse) of length l_H. Normally the parts are made of the same material with one E_P; however, attention must be paid to special cases.

If $D_A \geq D_{A,Gr}$ then the following holds:

$$\delta_P = \frac{2\ln\left(\frac{(d_w + d_h)(d_w + l_K\tan\phi - d_h)}{(d_w - d_h)(d_w + l_K\tan\phi + d_h)}\right)}{\pi E_P d_h \tan\phi}$$

If $d_W \le D_A \le D_{A,Gr}$ then:

$$\delta_P = \frac{\dfrac{2}{d_h \tan\varphi} \ln\left(\dfrac{(d_w + d_h)(D_A - d_h)}{(d_w - d_h)(D_A + d_h)}\right) + \dfrac{4}{D_A^2 - d_h^2}\left[l_K - \dfrac{(D_A - d_w)}{\tan\varphi}\right]}{\pi E_P}$$

Investigations have shown that the (imaginary) cone angle φ of the substitution deformation solid is not constant. As a result of the parabolic deformation solid described before and of the supporting effect of the surrounding material, it is substantially influenced by the main dimensions of the plate:

$$\tan\varphi = 0{,}362 + 0{,}032 \ln\frac{l_K}{2d_w} + 0{,}153 \ln\frac{D'_A}{d_w}$$

For the non-cylindrical components (rectangular flange, detail of a multi-bolted joint) generally occurring in practice, there are at present no reliable findings for calculating the plate resilience. Such geometries are considered to be cylindrical as an approximation. The (substitution) outside diameters D_A and D'_A are then calculated as *twice the mean* edge distance at the interface, i.e., on the basis of section dimensions at the level of the part separation surface.

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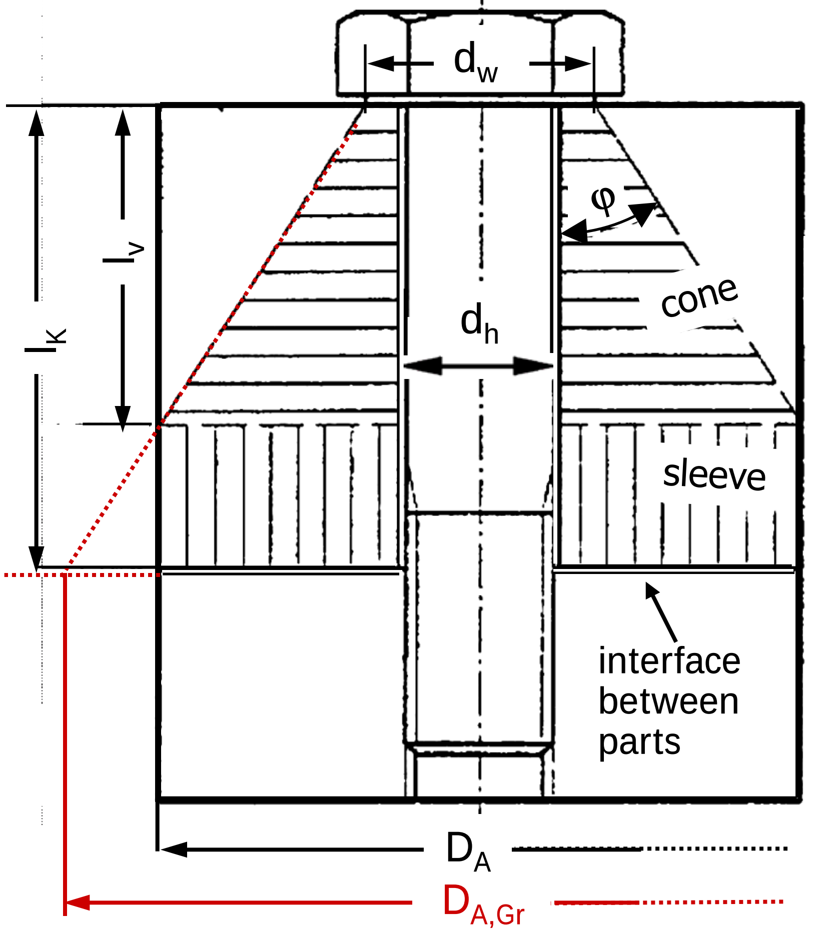

In the case of tapped thread joint, to simplify the calculation of the plate resilience, the top cone and the bottom truncated cone are replaced by *one* substitution deformation cone of the same resilience, which *can* be followed by a sleeve.

The corresponding formulas for resilience are:

If $D_A \geq D_{A,Gr}$:

$$\delta_P = \frac{\ln\left(\frac{(d_w + d_h)(d_w + 2 l_K \tan\varphi - d_h)}{(d_w - d_h)(d_w + 2 l_K \tan\varphi + d_h)}\right)}{\pi E_P\, d_h \tan\varphi}$$

If $d_W < D_A \leq D_{A,Gr}$: $\delta_P = \frac{1}{\pi E_P}\left\{\frac{1}{d_h \tan\varphi}\ln\left(\frac{(d_w + d_h)(D_A - d_h)}{(d_w - d_h)(D_A + d_h)}\right) + \frac{4}{D_A^2 - d_h^2}\left[l_K - \frac{(D_A - d_w)}{2\tan\varphi}\right]\right\}$

In the case of tapped thread joint the substitution cone angle which produces the same resilience as the real body is interpolated by the formula:

$$\tan\varphi = 0{,}348 + 0{,}013 \ln \frac{l_K}{2d_W} + 0{,}193 \ln \frac{D'_A}{d_W}$$

Sections 8, 9, 10, 11, 12 - Bolt clamping

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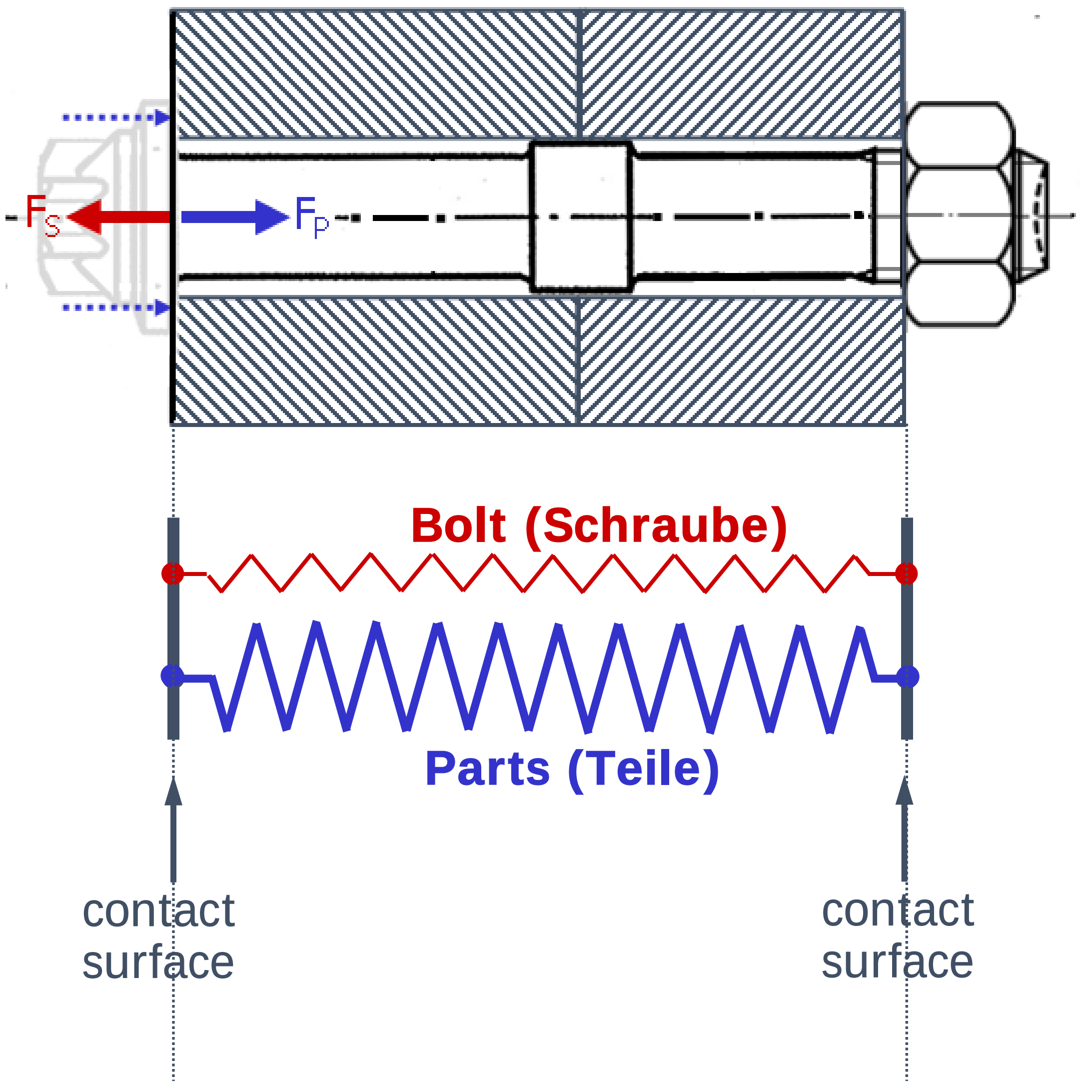
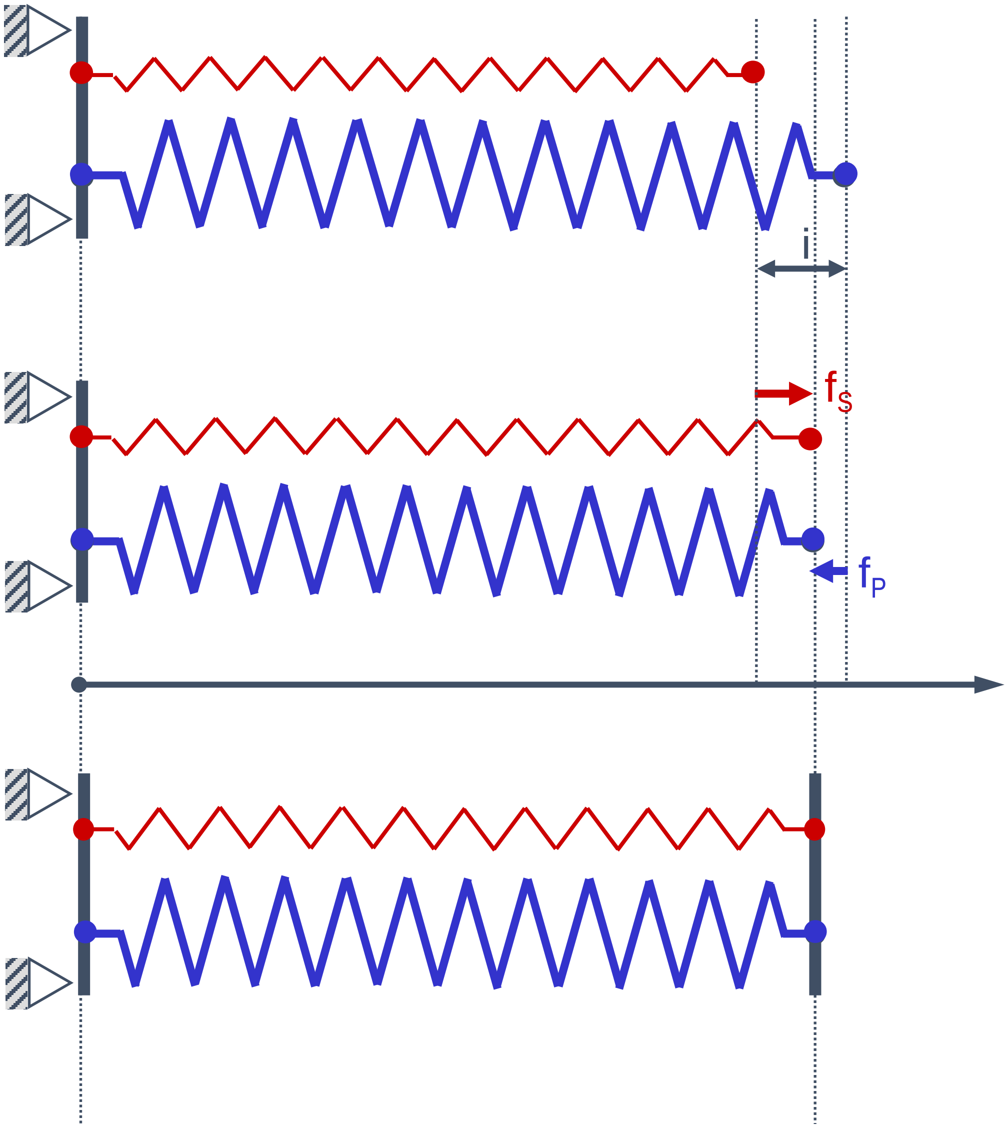


The forces and axial deformations in the single-bolted joint can be described by means of a simple mechanical spring model.

In this model, the bolt and the clamped parts are considered as tension and compression springs with the elastic resiliences δ_S and δ_P



Initial dimensions of bolt and parts

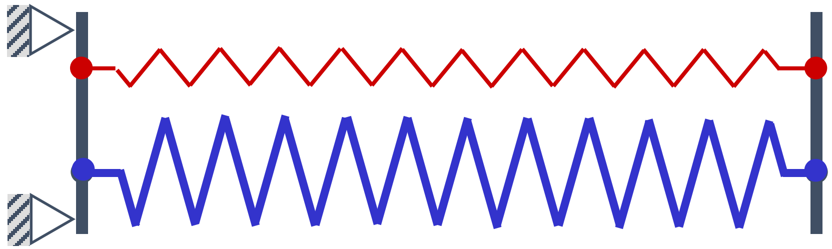
Assembly displacement compatibility:

$|f_S| + |f_P| = i$

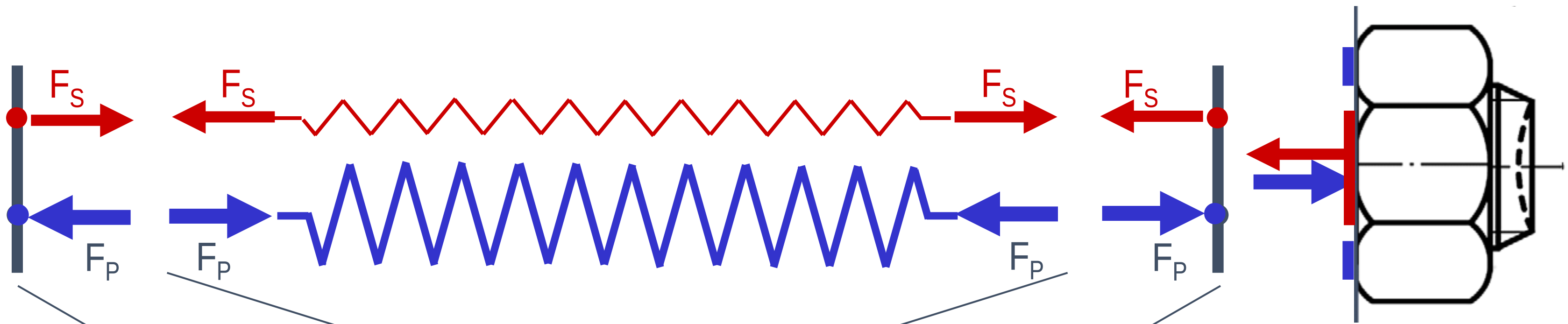
$\Leftarrow$ This can be done in many ways, however one only is good: the one satifying equilibrium

Final common length

11 - Spring model and tightening (3 / 7)



Final common length



bolt and parts forces

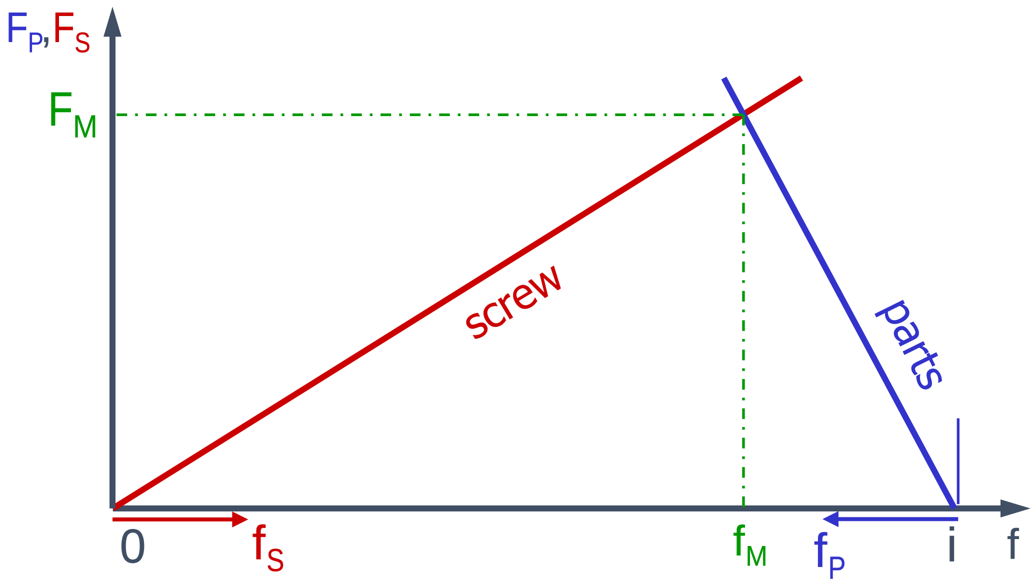
contact surface forces

Assembly force equilibrium:

$|F_S| = |F_P|$

or

$F_S = F_P$



$f_M = F_M\,\delta_S$: bolt elongation

$f_P = i - f_M = F_M \delta_P$: parts shortening

The sum of bolt elongation and parts shortening is equal to interference i :

$$i = F_M(\delta_S + \delta_P)$$

Remember that it was also:

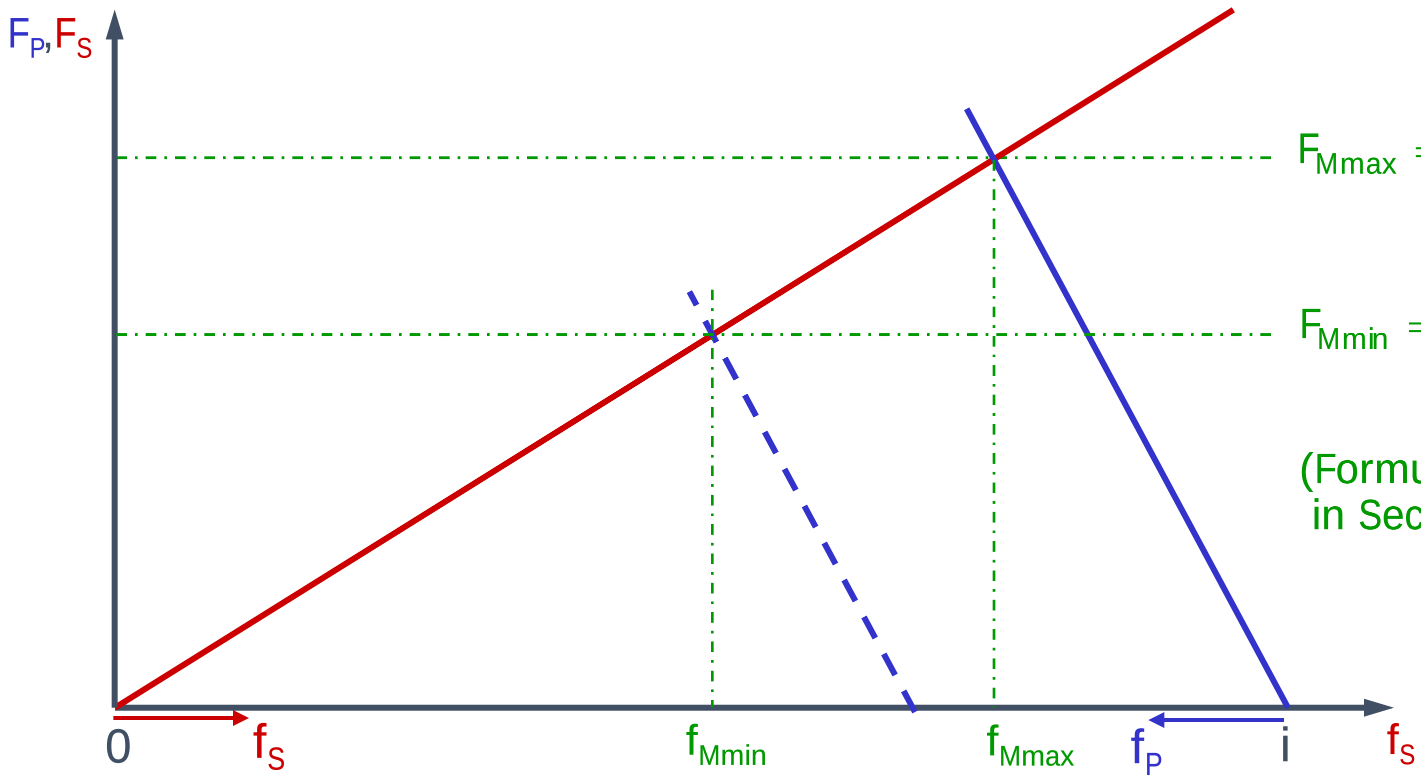
$$i = n_i P = \frac{\alpha_i}{2\pi} P$$

Joint assembly (german: **Montage)**

Bolt and part displacements under load, f_S and f_P, are measured from an initial dimension "at rest" (i.e., with zero axial force); for the bolt this is the head-to-nut inner face distance in the tightened configuration but with the parts removed; for the parts it is the axial dimension of the part "at rest", i.e., when no load is applied (in both cases, wear and surface adaptation due to sliding under load during assembly are taken into account).

The resultant force on the bolt section, at the ends and through the length, is F_S , tensile. The resultant force on the parts, at the ends and through the length, is F_P, compressive. Due to equilibrium $F_S = F_P$. At assembly they are called F_M .

Uncertainty of tigthening force due to the assembly process



$$F_{Mmax} = \frac{0{,}9 R_{p0,2} A_0}{\sqrt{1+3k^2}}$$

$$F_{Mmin} = \frac{0{,}9 R_{p0,2} A_0}{\alpha_A \sqrt{1+3k^2}}$$

(Formulas explained in Sect. 13)

Taking into account the "tightening factor" (elsewhere also named "tightening uncertainty factor"):

$$\alpha_A = \frac{F_{Mmax}}{F_{Mmin}}$$

Reminder: for torque control α_A could be between 1,6 and 2,5 i.e. reduction from 37.5% to 60% !!!

F_{Mmin} is also found and represented on the joint diagram.

Loss of clamping force due to vibration and fretting - 1

This happens later!

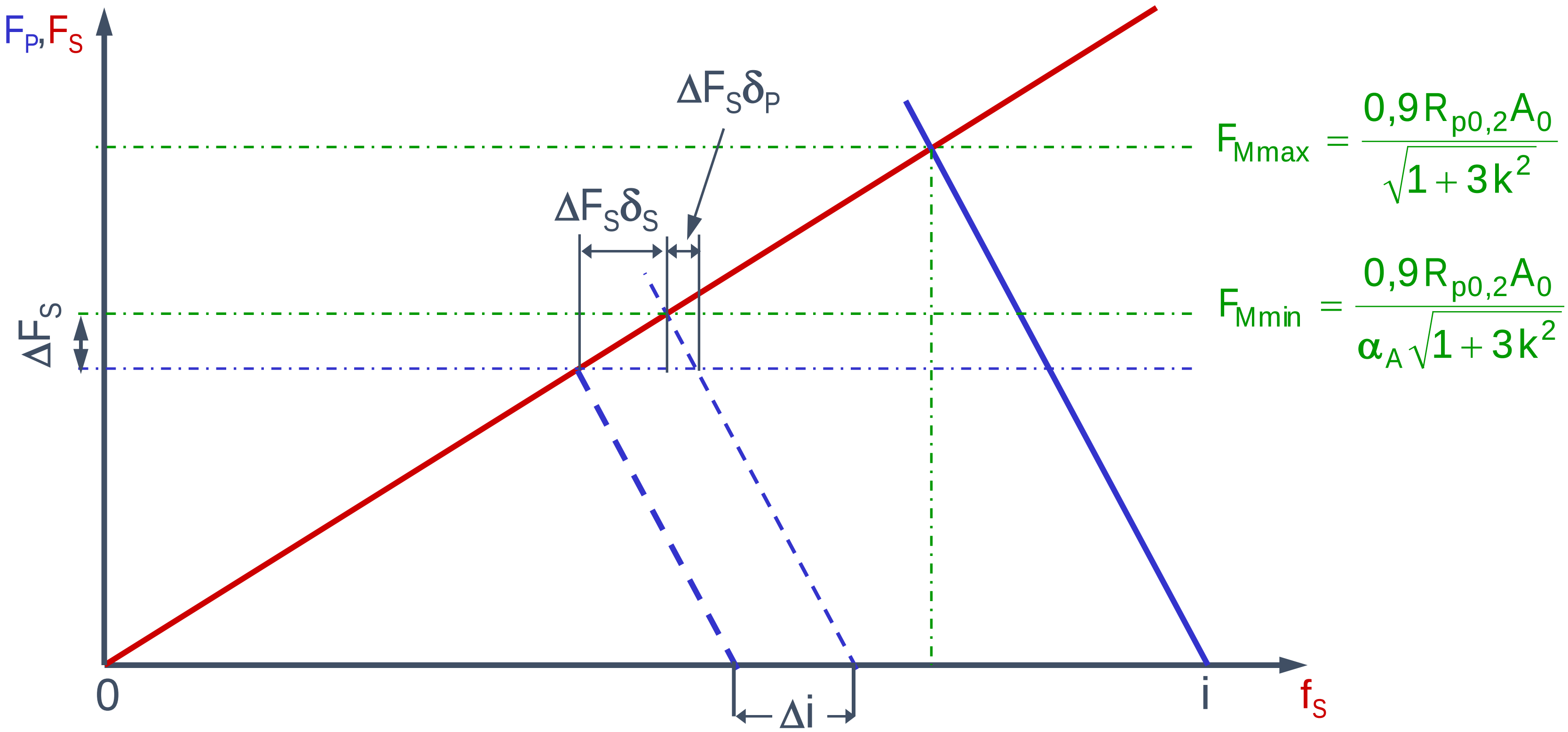
During operation the parts may wear at their interfaces due to fretting (microsliding) induced by vibrations. **This results in a clamping force loss.**

It depends on the number of clamped surfaces beyond the thread which is already included. Loss of interference Δi is given in the Table shown below (from VDI 2230)

Table 5.4/1. Guide values for amounts of embedding of bolts, nuts and compact clamped parts made of steel

Average roughness height Rz according to DIN 4768	**Loading**	**Gulde values for amounts of embedding in µm** in the thread	 per head or nut bearing area	 per inner interface
< 10 µm	tension/compression	3	2.5	1.5
	shear	3	3	2
10 µm up to 40 µm	tension/compression	3	3	2
	shear	3	4.5	2.5
40 µm up to 160 µm	tension/compression	3	4	3
	shear	3	6.5	3.5

Loss of clamping force due to vibration and fretting - 2



Fretting interference loss Δi produces a decrease ΔF_S of clamping force:

$$\Delta i = \Delta F_S \delta_S + \Delta F_S \delta_P \quad \Rightarrow \quad \Delta F_S = \frac{\Delta i}{\delta_S + \delta_P}$$

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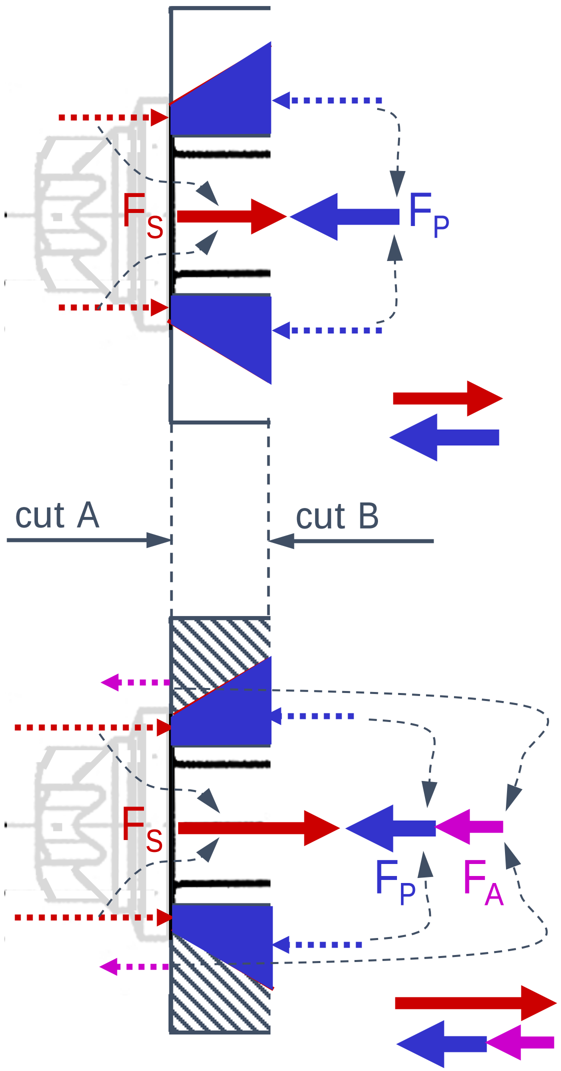


Joint assembly

This figure represents one cutaway end of a bolted joint. Cut planes A and B separate a part section, which is equilibrium under the applied axial forces, whose resultants are shown in solid lines.

The resultant force on the bolt section is F_S .

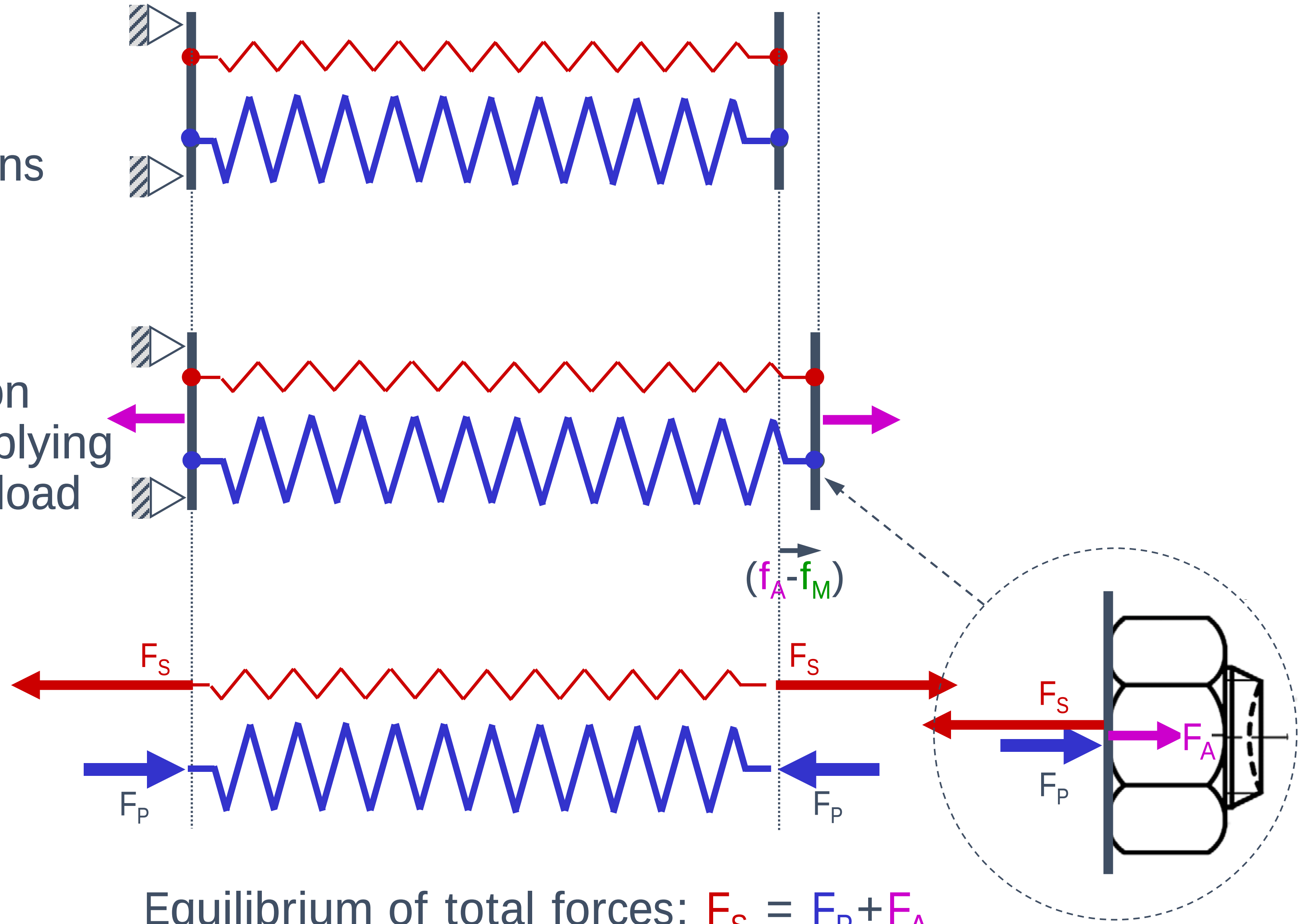
A distributed force under the head transmits head compression to the clamped part at cut A. Total value of compression is $F_S=F_P$. On cut B another distribution provides a total resultant force F_P acting along the bolt/part axis. At assembly F_S and $F_P \Rightarrow F_M$

Joint loading

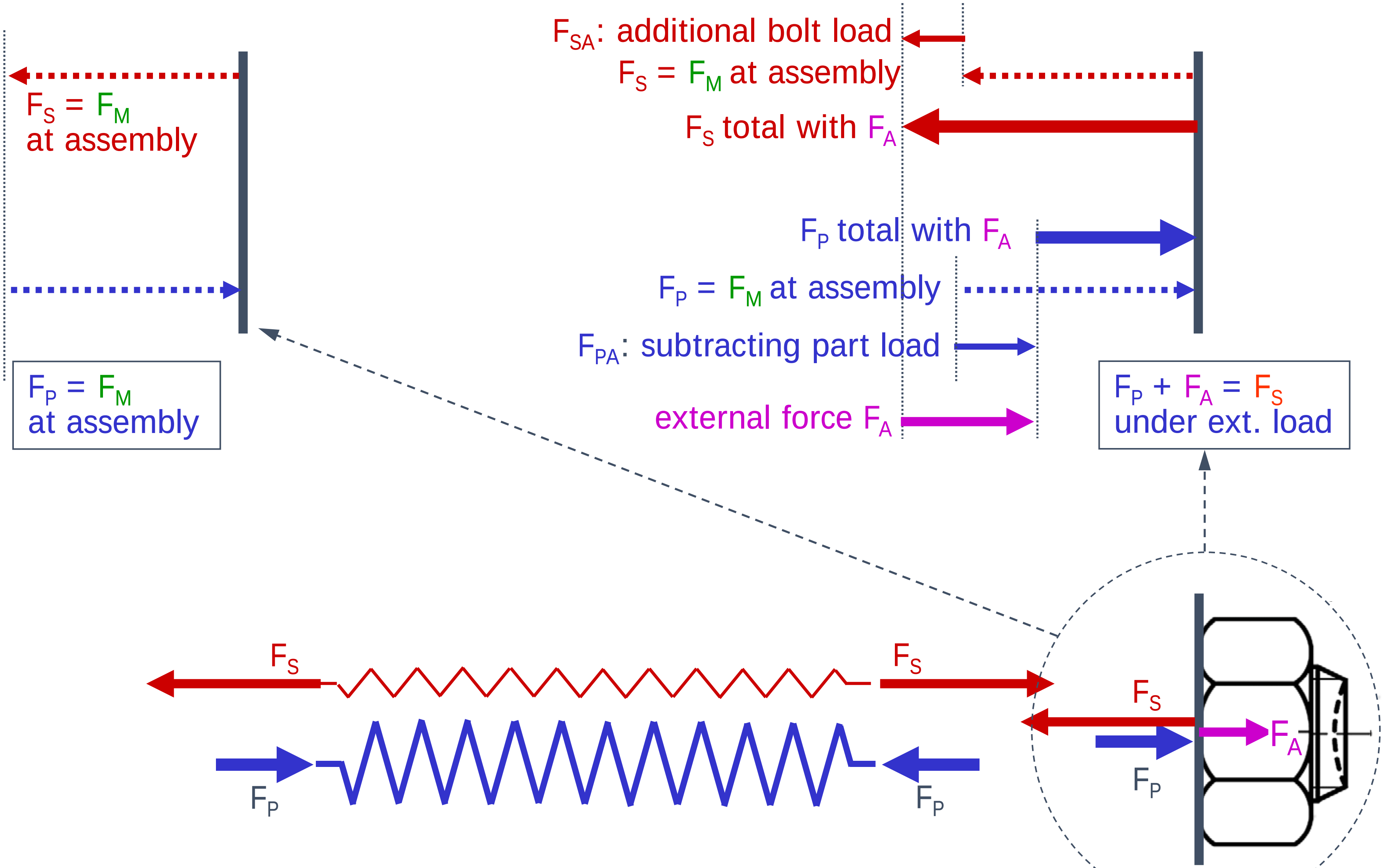
Now an external force F_A is applied to the part on (or very near to) its outer surface (the distribution is indicated). The part receives from the head/part contact surface - cut A - the force F_S , from the adjacent part material through, cut B, the resultant force F_P . Equilibrium of the part fragment requires that: $F_P+F_A=F_S$

Initial dimensions

Elongation when applying external load



Equilibrium of total forces: $F_S = F_P + F_A$



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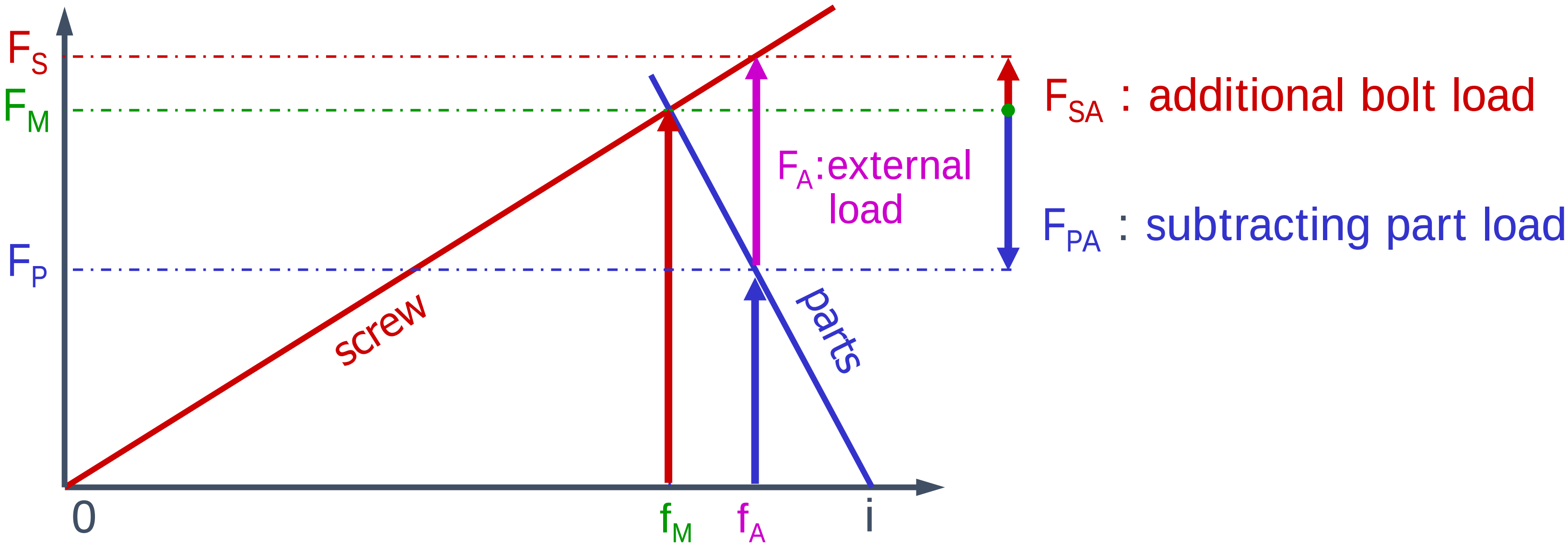


Joint loading

An external axial force F_A is applied to both ends of the part, on (or very near to) their surfaces. Equilibrium requires $F_P+F_A=F_S$. The bolt elongates up to f_A .

$$\begin{cases} F_P+F_A=F_S \\ (F_M-F_{PA})+F_A=(F_M+F_{SA}) \end{cases}$$

$$F_A=F_{PA}+F_{SA}$$

$$\begin{cases} f_A-f_M = \delta_S\, F_{SA} \\ f_A-f_M = \delta_P\, F_{PA} \end{cases} \Downarrow \qquad F_{PA}=\frac{\delta_S}{\delta_P}F_{SA}$$

$$\Longrightarrow \quad F_A = \frac{\delta_S}{\delta_P}F_{SA}+F_{SA} = F_{SA}\left(\frac{\delta_S+\delta_P}{\delta_P}\right)$$

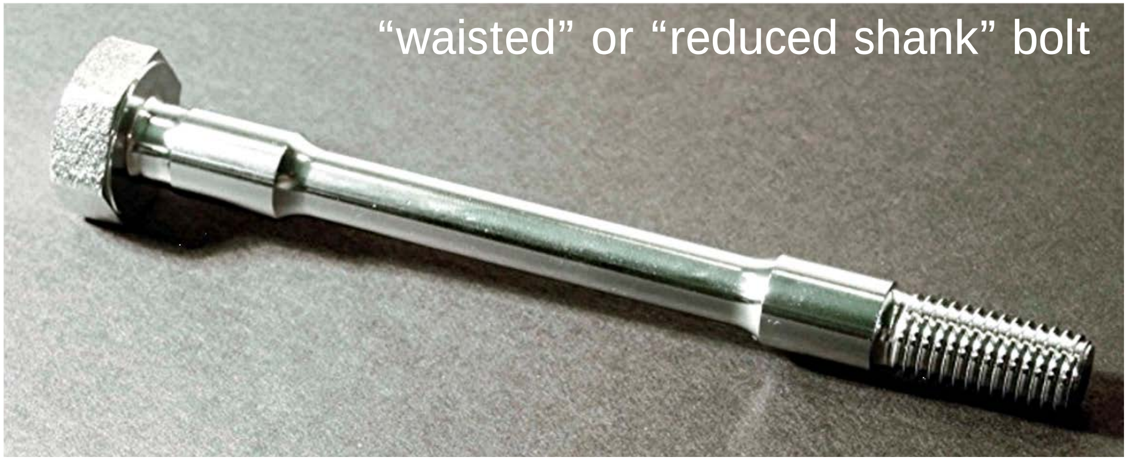
then : $\boxed{F_{SA} = F_A\dfrac{\delta_P}{\delta_S+\delta_P}}$

Sections 13, 14 - Static and fatigue design

The purpose of Sections 13 and 14 is to proceed from forces, calculated in the previous sections, to stresses. Static design under maximum load is explored, as well as fatigue design under pulsating stresses.

Section 13 illustrates how stresses are calculated to design bolts against yield under operating loads. At the same time, also limits to the minimum part clamping forces are explored as well.

Section 14 explores criteria to design the bolt at fatigue. The section is concluded with the influence of technological processing on the material's capabilities to withstand fatigue stresses, and how to improve such capabilities.



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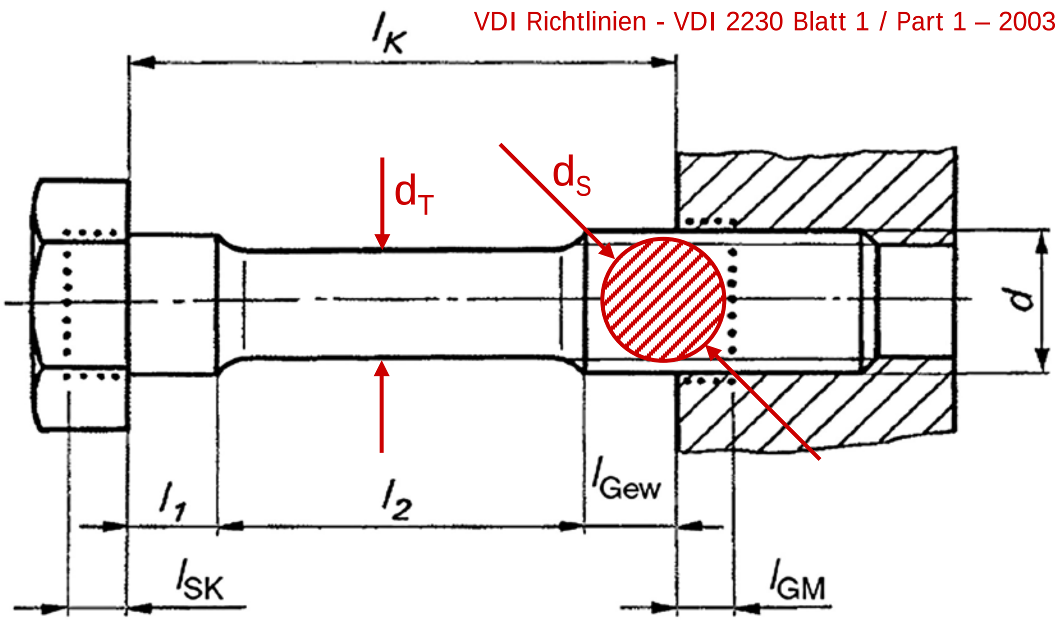

Bolt diameters

In the threaded section:

d_3 : minimum bolt cross section diameter (minor diameter)

For strength calculation the lowest of the following diameters shall be employed:

d_T : shank diameter

d_S : “stress” diameter, i.e., the diameter of the “stress area” or “core area”

With reference to symbols of Sect.6 sl.3, the stress diameter according to DIN 13-28:

$$d_S = \frac{d_2 + d_3}{2}$$

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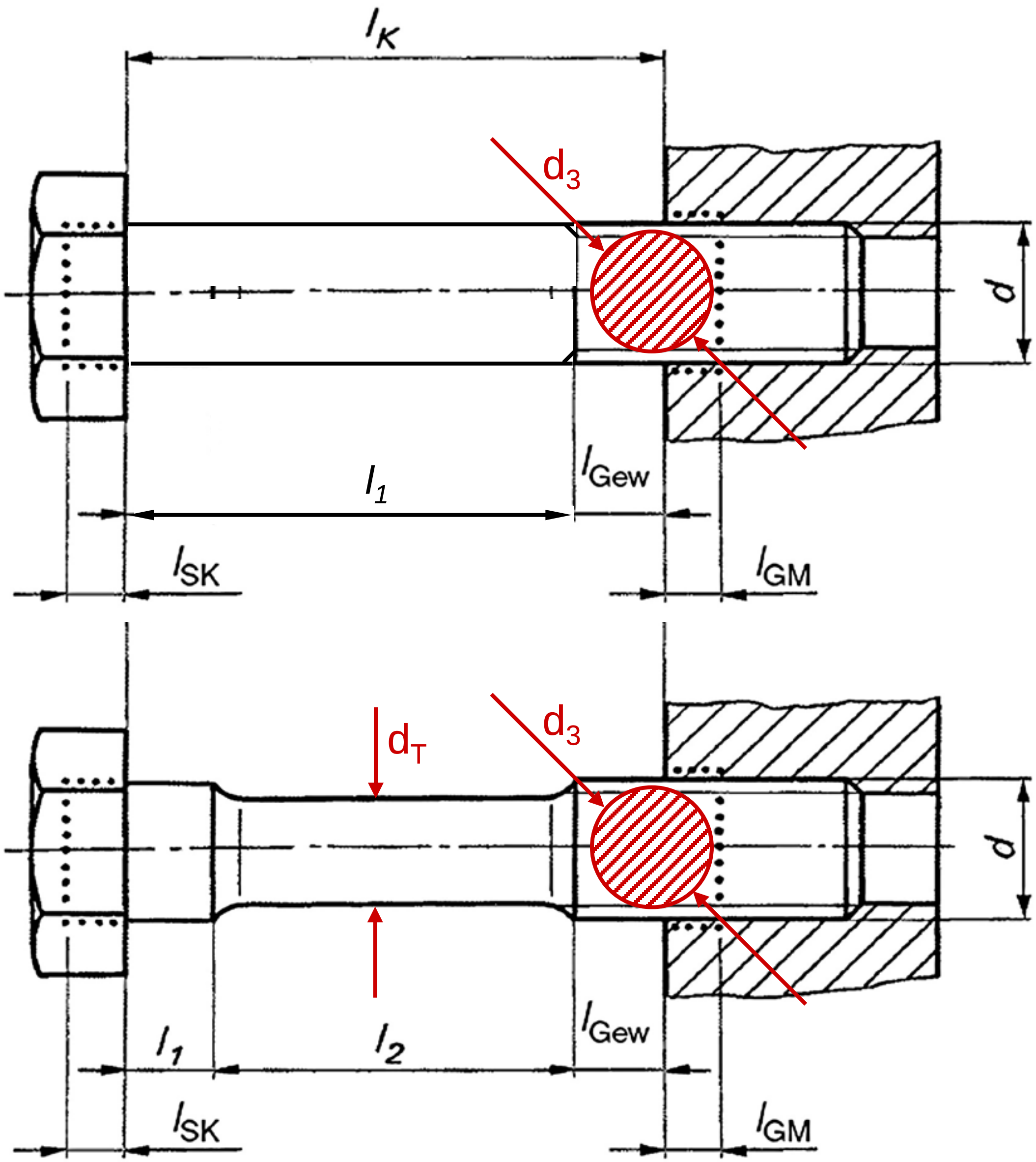


Bolt areas

In strength calculations the minimum applicable diameter and area depend then on bolt shape.

In the formulas which will follow these symbols will be employed:

d_0: diameter at the relevant smallest cross section of the bolt, either d_3 or d_T

A_0: appropriate minimum cross-sectional area of the bolt, based on d_0

$$A_0 = \pi d_0^2 / 4$$

W_P: minimum polar resistance modulus

$$W_P = \pi d_0^3 / 16 \cong 0{,}2\, d_0^3$$

Bolt stresses

The aim is to utilize the bolt strength to the greatest possible extent. Normally the equivalent stress in the bolt (nominal stress) at assembly is 90% of $R_{p0,2}$ (minimum guaranteed).

The thread torque M_G produces tangential stresses τ , the assembly bolt load F_M produces a (mean or nominal) tensile stress σ . The maximum equivalent stress during the tightening process is then:

$$\sigma_{eq} = \sqrt{\sigma^2 + 3\tau^2} = \sqrt{\left(\frac{F_{M\max}}{A_0}\right)^2 + 3\left(\frac{M_G}{W_P}\right)^2} = \frac{F_{M\max}}{A_0}\sqrt{1+3k^2}$$

see page 88

$$k = \frac{\tau}{\sigma} = \frac{M_G}{F_{M\max}}\frac{A_0}{W_P} = \frac{4}{d_0}\left[0.16\ P + 0.58\ d_2\ \mu_G\right]$$

$$\sigma_{eq} = 0{,}9R_{p0,2} \quad \Rightarrow \quad \frac{F_{M\max}}{A_0}\sqrt{1+3k^2} = 0{,}9R_{p0,2} \quad \Rightarrow \quad F_{M\max} = \frac{0{,}9R_{p0,2}A_0}{\sqrt{1+3k^2}}$$

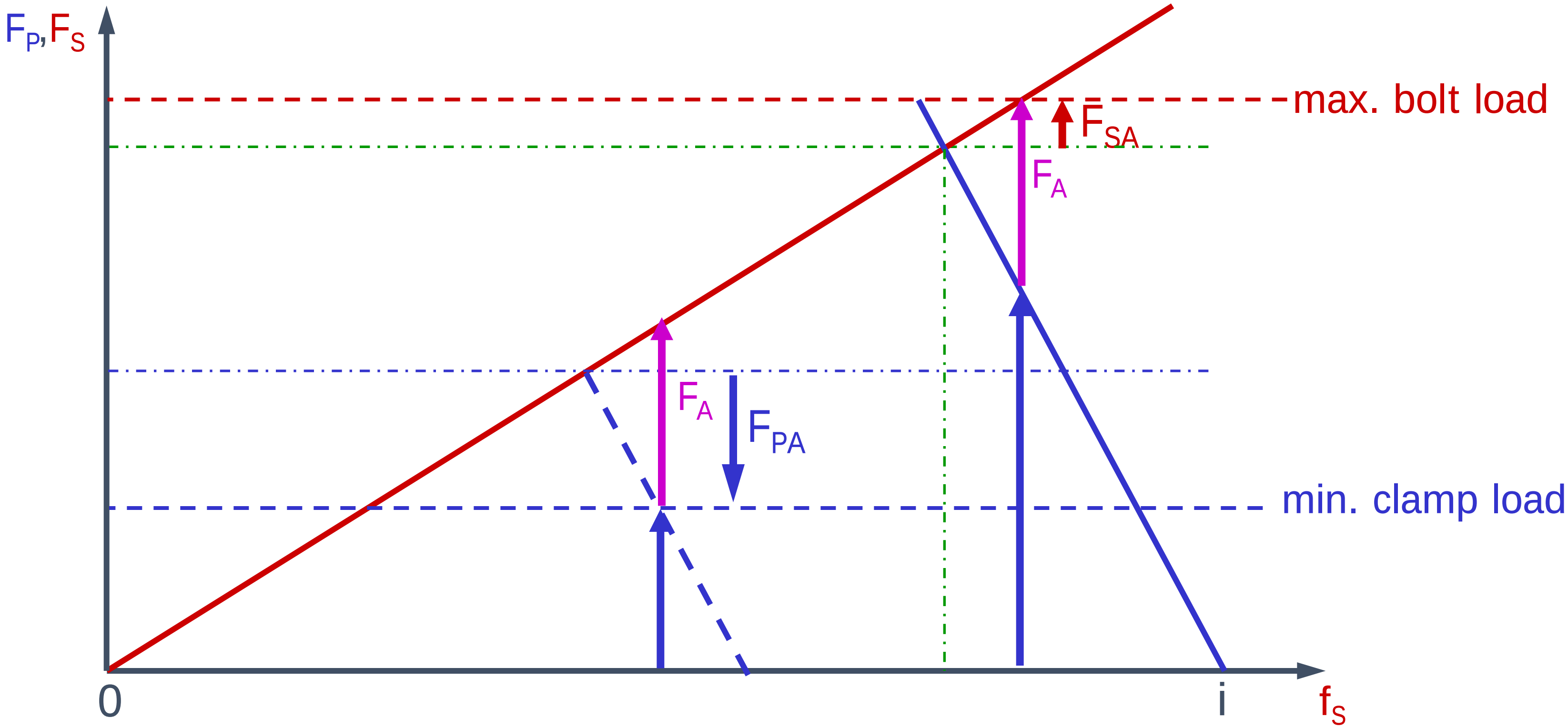
Bolt stresses - refinements

When the yield point of the material, i.e. the fully plastic state, is reached in the cross section A_0 , then there is a constant torsional stress τ over the cross section. This condition is satisfied by a correction to the torsion section modulus. The following applies: $W_P' = \pi\, d_0^3 / 12$.

$$\sigma_{eq} = \sqrt{\sigma^2 + 3\tau^2} = \sqrt{\left(\frac{F_{M\max}}{A_0}\right)^2 + 3\left(\frac{M_G}{W_P'}\right)^2} = \frac{F_{M\max}}{A_0}\sqrt{1 + 3k^2}$$

$W_P' = \pi\, d_0^3 / 12$: “corrected” minimum torsion section modulus

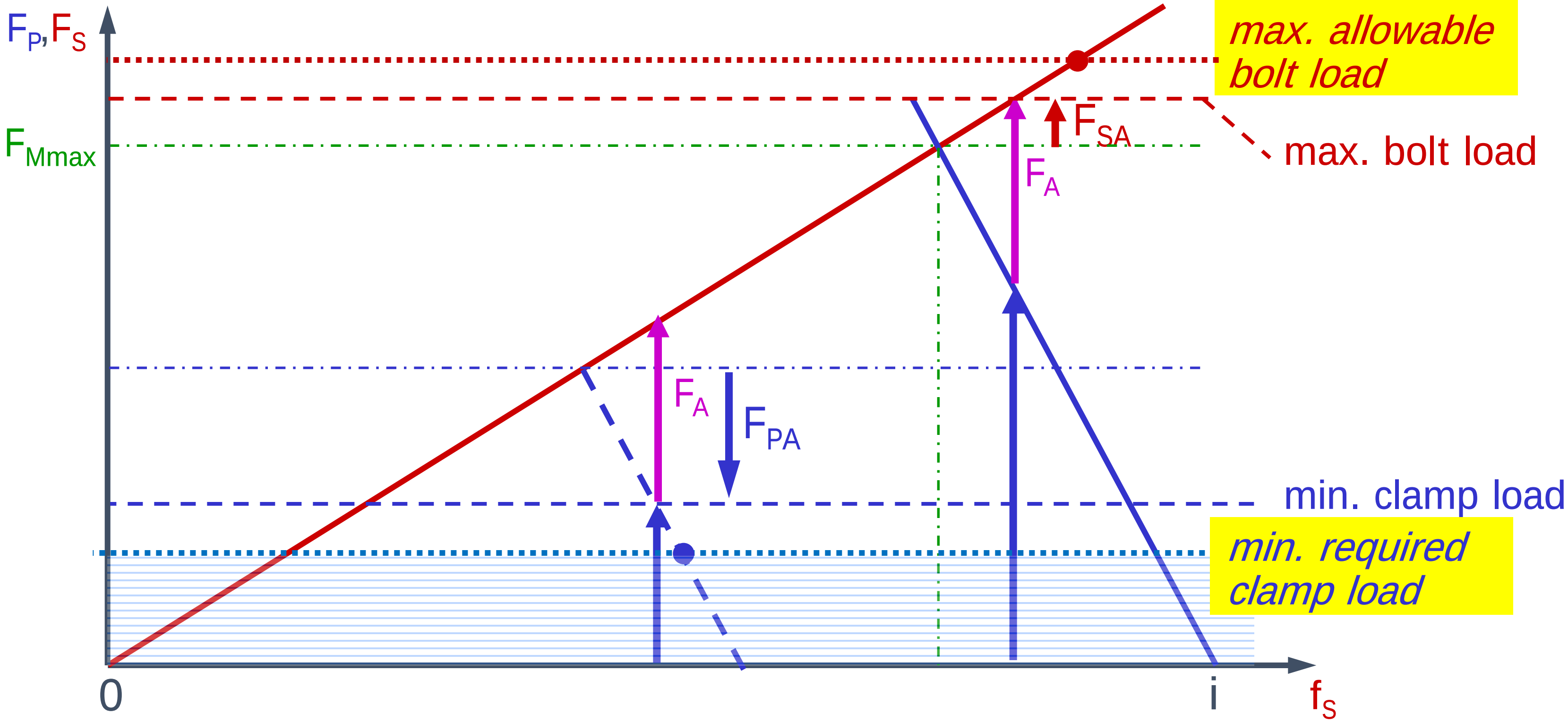
$$k = \frac{3}{d_0}\left[0{,}16\, P + 0{,}58\, d_2\, \mu_G\right]$$



When in operation, the joint is subjected to an external axial load F_A, shared by bolt and parts according to the elastic load sharing; both at the highest and at the lowest possible clamping loads.

The bolt must be designed against the maximum stress and against fatigue.

The part must be designed against a the minimum clamp load.



The upper dotted red line indicates the value of the *maximum allowable bolt load*, on the basis of the maximum bolt stress reached during operation.

The lowest dotted blue line indicates the value of the *minimum required clamp load*, below which the operation is not safe to joint opening, sliding or insufficient compression at the interface of the clamped parts.

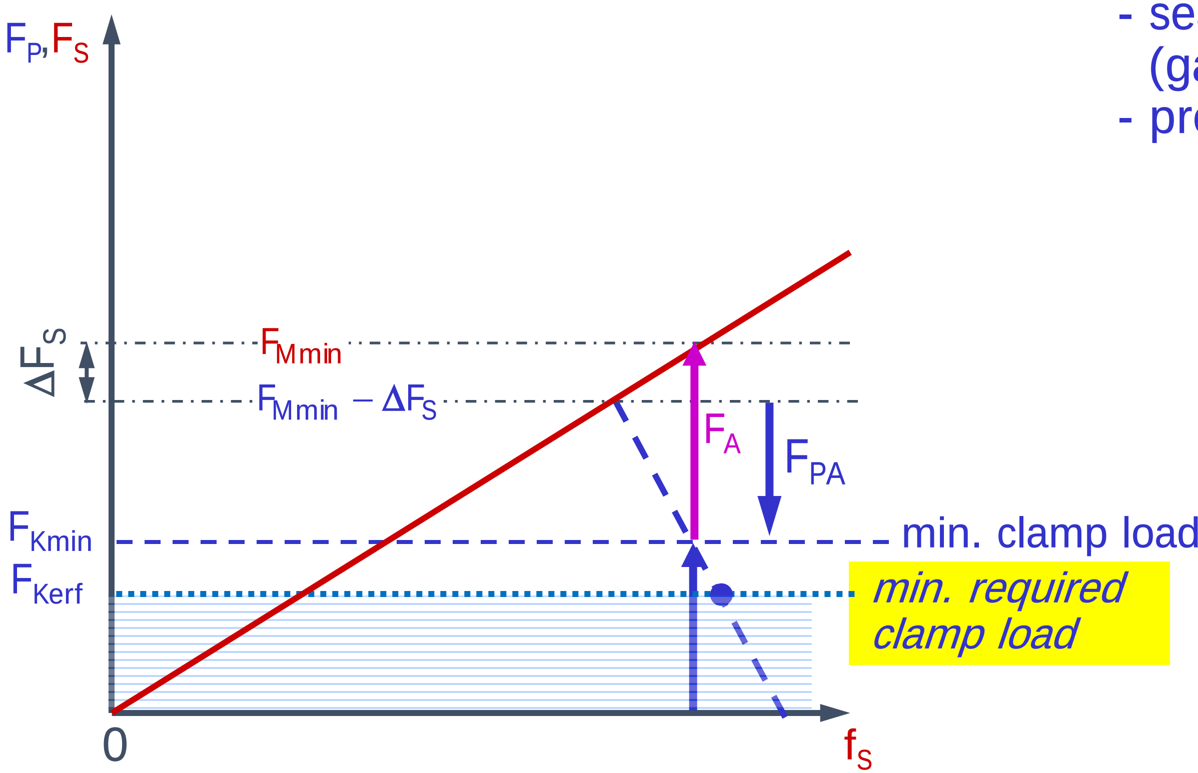
Design against minimum clamp load:

$$F_{K\,min} = F_{M\,min} - \Delta F_S - F_{PA} \geq F_{Kerf}$$

F_{Kerf} : min. *required* clamp load (*erforderlich*)

i.e. the lowest of:

- friction grip to transmit a transverse load or torque
- sealing against a medium (gas or liquid)
- prevention of joint opening

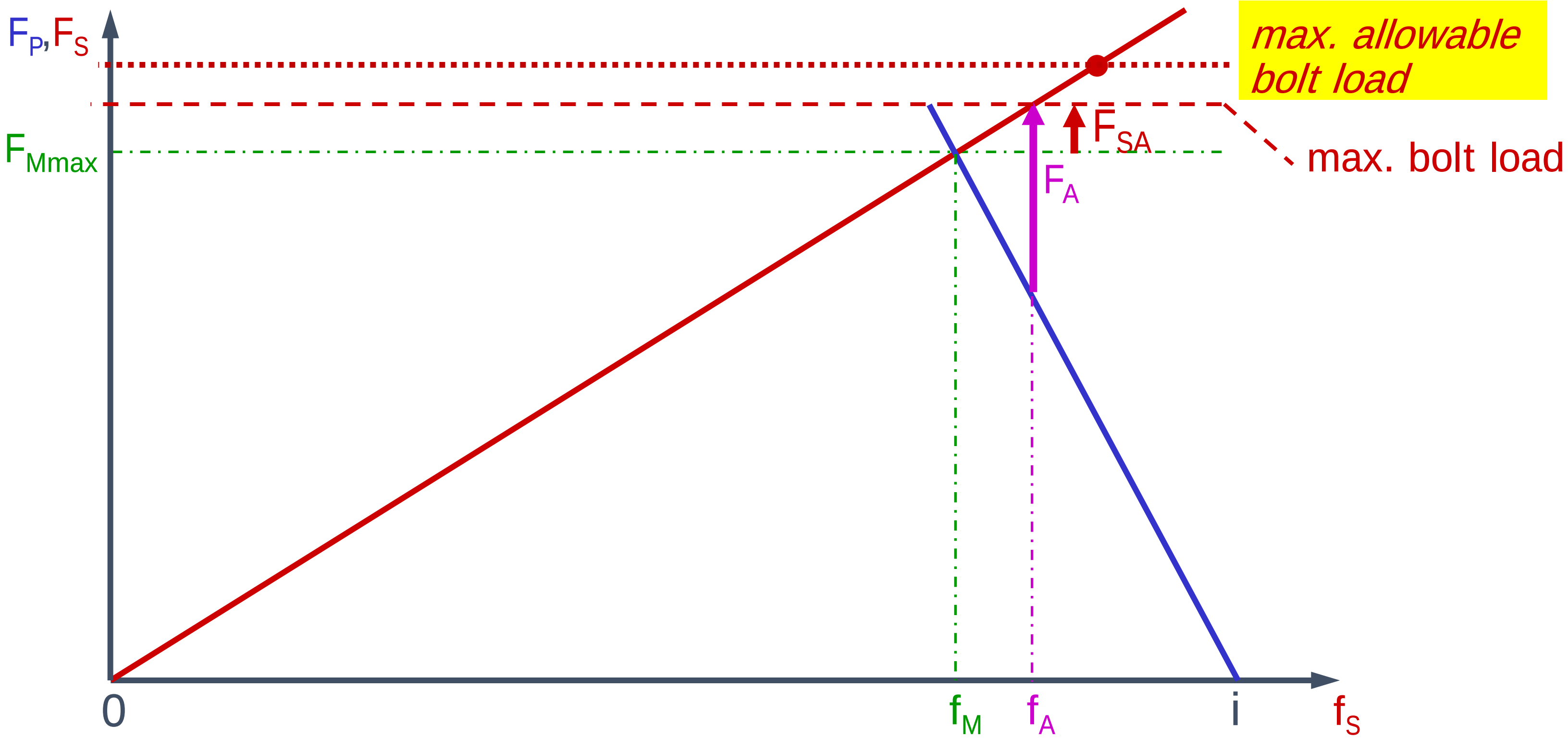


$$F_{Mmin} = \frac{0{,}9\,R_{p0,2}A_0}{\alpha_A\sqrt{1+3k^2}}$$

$$\Delta F_S = \frac{\Delta i}{\delta_S + \delta_P}$$

$$F_{PA} = F_A \frac{\delta_S}{\delta_S + \delta_P}$$

Design against maximum bolt stress



The max allowable bolt load is the one which produces in the bolt a total equivalent stress σ_{eq} equal to $R_{p0,2}$.

A first safe approximation assumes that the bolt has no torque elastic relaxation after the end of the tightening process ...

... ... design against maximum bolt stress

From: $F_{Mmax} = \dfrac{0{,}9 R_{p0,2} A_0}{\sqrt{1+3k^2}}$ $\quad \sigma_{Mmax} = \dfrac{F_{Mmax}}{A_0}$ $\quad \tau_{Mmax} = \dfrac{M_G}{W_p}$

with: $k = \dfrac{4}{d_0}\left[0{,}16\,P + 0{,}58\,d_2\,\mu_G\right]$

and: $\sigma_{tot} = \sigma_{Mmax} + \sigma_{SA}$ $\quad$ with: $\sigma_{SA} = \dfrac{F_{SA}}{A_0}$; $F_{SA} = F_A \dfrac{\delta_P}{\delta_S + \delta_P}$

it follows: $\sigma_{eq} = \sqrt{\sigma_{tot}^2 + 3\tau_{Mmax}^2} = \sqrt{\left(\sigma_{Mmax} + \sigma_{SA}\right)^2 + 3\,\tau_{Mmax}^2} =$

$$= \sqrt{\left(\frac{F_{Mmax}}{A_0} + \frac{F_{SA}}{A_0}\right)^2 + 3\left(\frac{M_G}{W_P}\right)^2} =$$

$$= \frac{F_{Mmax}}{A_0}\sqrt{\left(1 + \frac{F_{SA}}{F_{Mmax}}\right)^2 + 3k^2} \le R_{p0,2}$$

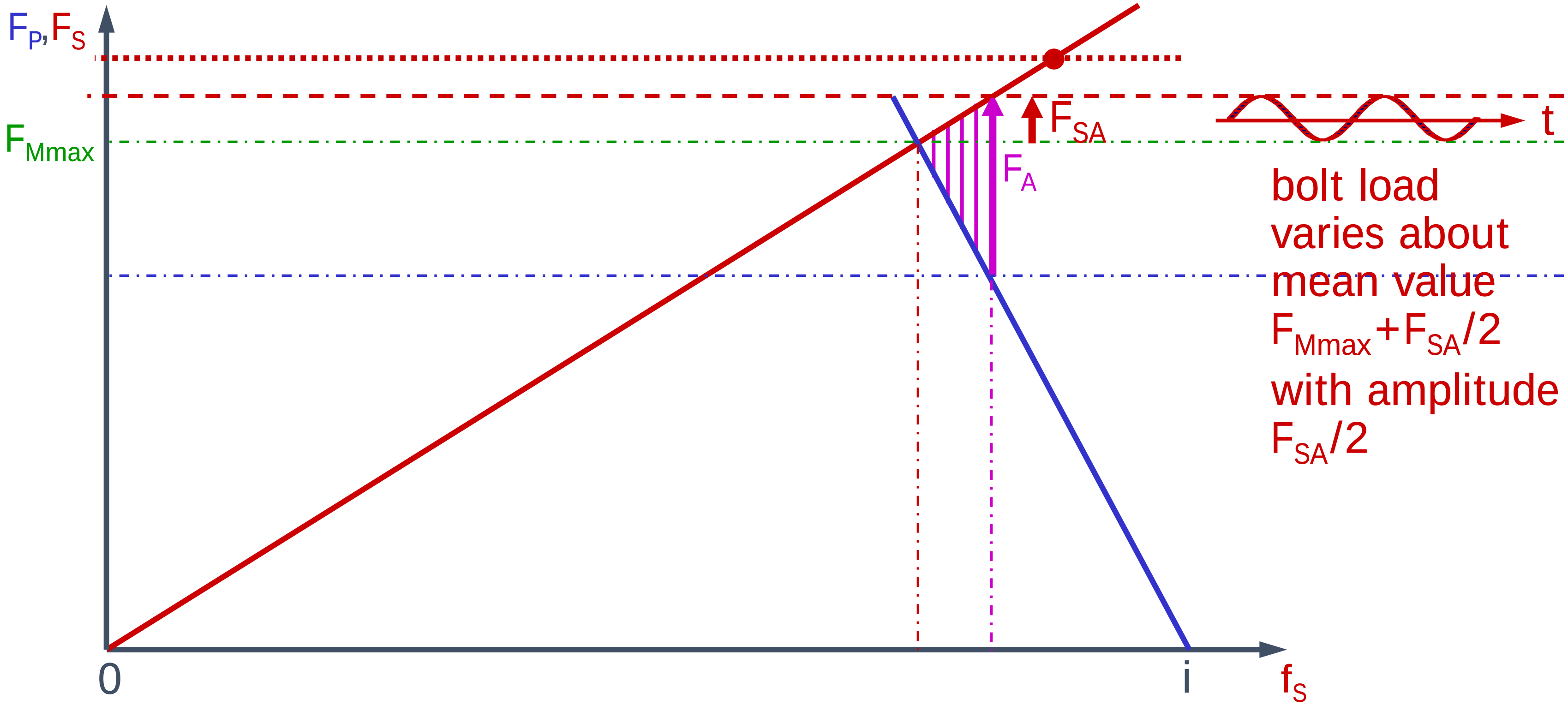
Sections 13, 14 - Static and fatigue design

The purpose of Sections 13, 14 and 15 is to proceed from forces, calculated in the previous sections, to stresses. Static design under maximum load is explored, as well as fatigue design under pulsating stresses.

Section 13 illustrates how stresses are calculated to design bolts against yield under operating loads. At the same time, also limits to the minimum part clamping forces are explored as well.

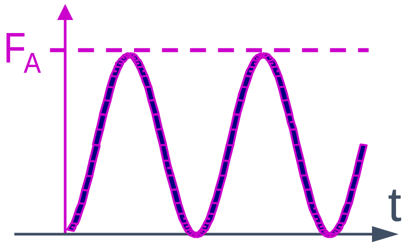
Section 14 explores criteria to design the bolt at fatigue. The section is concluded with the influence of technological processing on the material's capabilities to withstand fatigue stresses, and how to improve such capabilities.

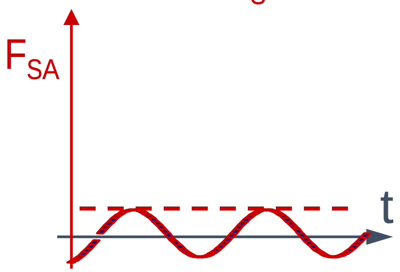
Design against bolt fatigue



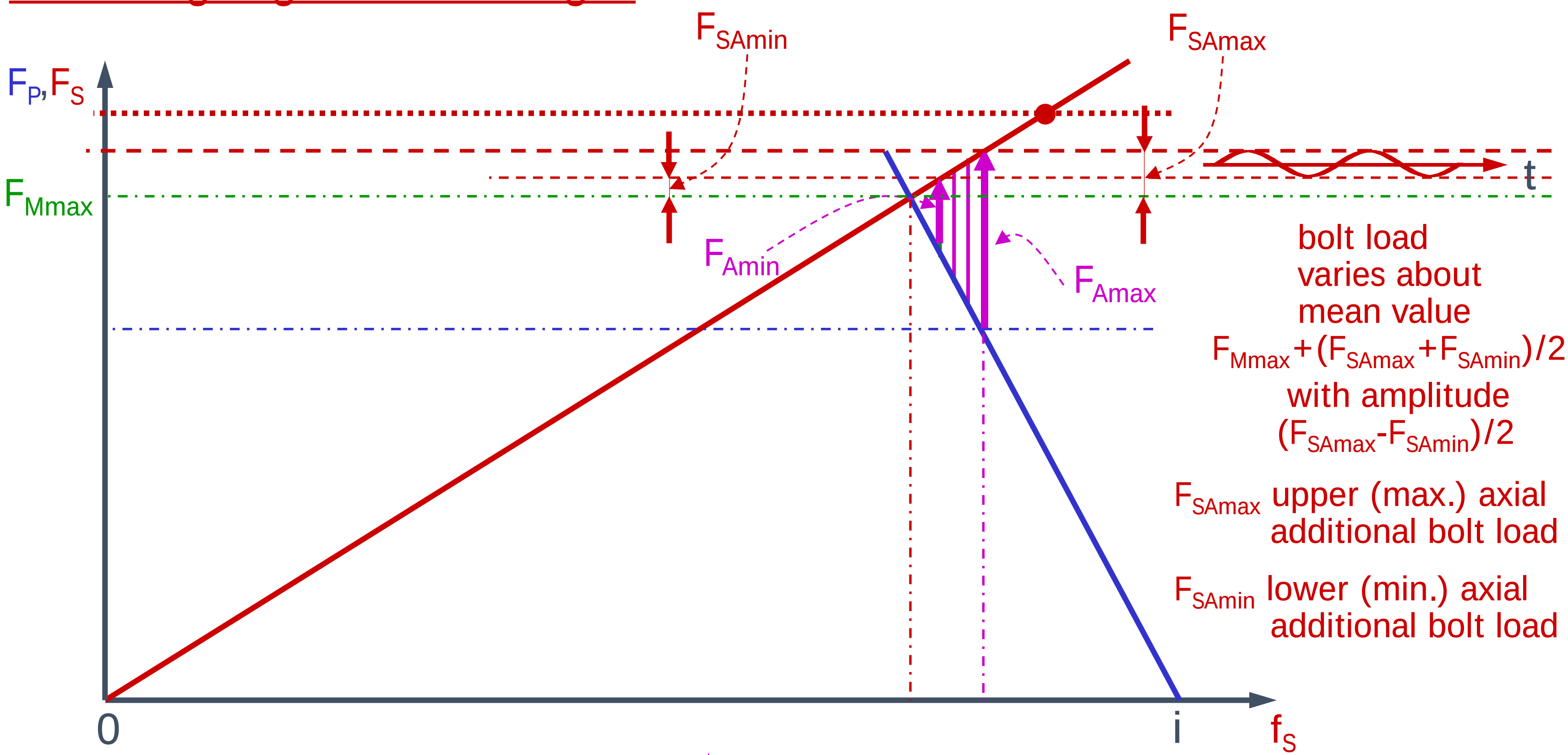
bolt load varies about mean value $F_{Mmax}+F_{SA}/2$ with amplitude $F_{SA}/2$

In this case the external force varies (with time) between zero and a maximum F_A and the bolt load F_{SA} as indicated:





... ... design against bolt fatigue

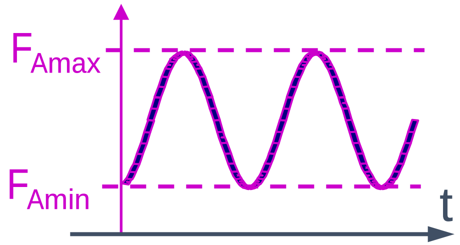


bolt load varies about mean value $F_{Mmax}+(F_{SAmax}+F_{SAmin})/2$ with amplitude $(F_{SAmax}-F_{SAmin})/2$

F_{SAmax} upper (max.) axial additional bolt load

F_{SAmin} lower (min.) axial additional bolt load

In this case the external force varies (with time) between a minimum F_{Amin} and a maximum F_{Amax} :



Symbols for stresses

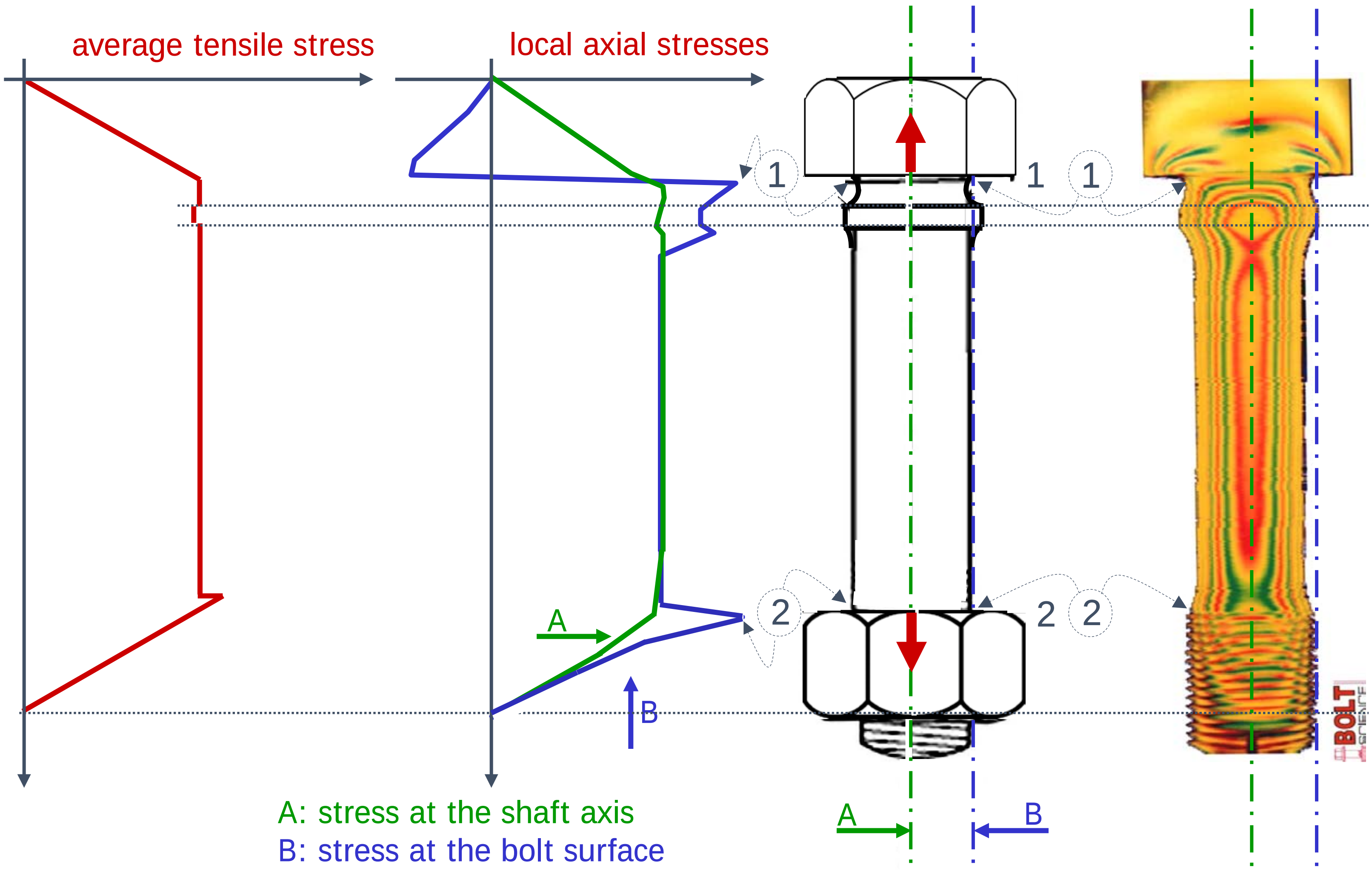
σ_a, σ_m : alternating and mean stress acting on the bolt

$$\sigma_a = \frac{F_{SA\,max} - F_{SA\,min}}{2A_S} \qquad \sigma_m = \frac{1}{A_S}\left(F_{M\,max} + \frac{F_{SA\,max} + F_{SA\,min}}{2}\right)$$

A_S **:** **"**stress area" of the bolt thread according to DIN 13-28 (sect. 3, sl. 4, 6)

These are nominal stresses. A high notch effect is present at the first load-bearing thread turn of the bolt. The local stress peaks here, depending on the design conditions, may be up to ten times higher than the nominal stresses. The load-bearing capacity of bolted joints is therefore markedly lower during alternating stress compared with static stress.

A bolt is considerably more than a “bar in tension” !



A: stress at the shaft axis

B: stress at the bolt surface

Warning: local peak stresses at the thread root radius are not represented here

An analysis of stress magnitudes reveals three "weak points", where stress peaks are present. Consider that if stress goes beyond yield at these points due to max load (in assembly and in operation) then a compressive residual stress of plastic origin is produced, which is not dangerous. The danger is in the elastic stress oscillations produced by the alternating loads taken by the screw at fatigue: in fact, these cycles are fully in the elastic field.

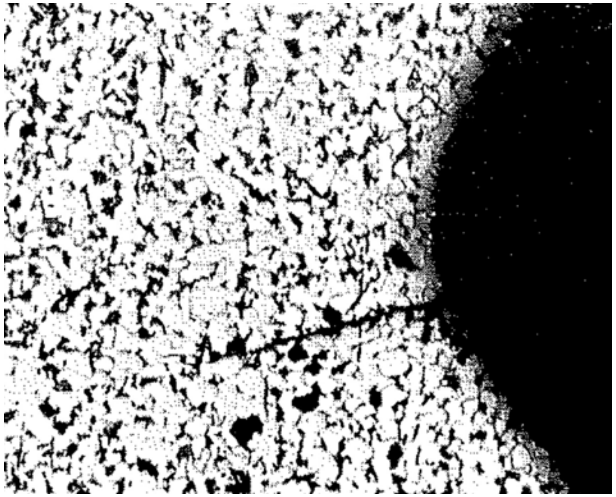
These points are:

1) the fillet where the head joins the shank

2) the bottom radius of first thread to engage the nut

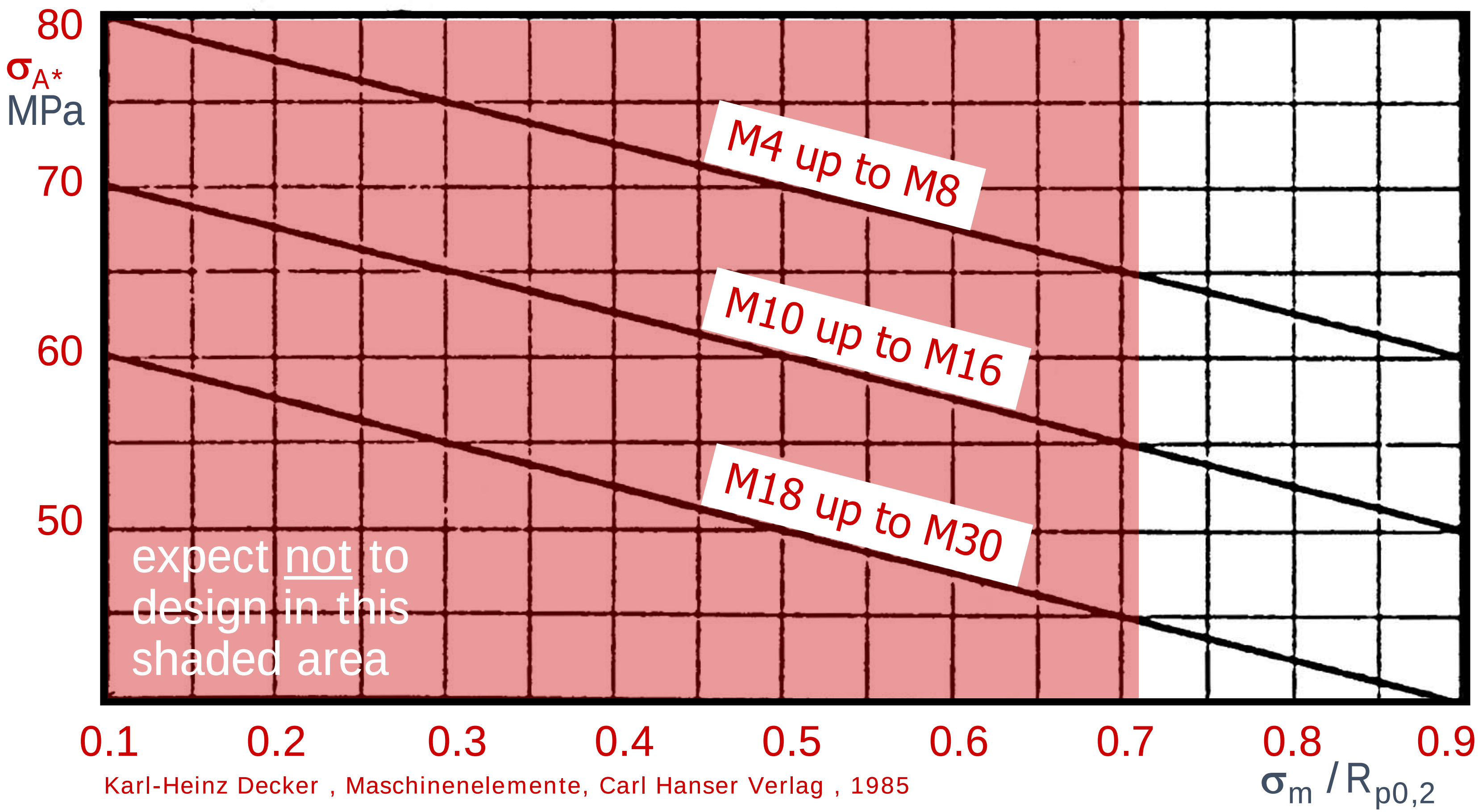
See next slide: these are the points at which the fastener usually fails.

Sometimes a critical point is the thread run-out point, where the threads meet the shank.



Fatigue crack developed at the thread bottom radius

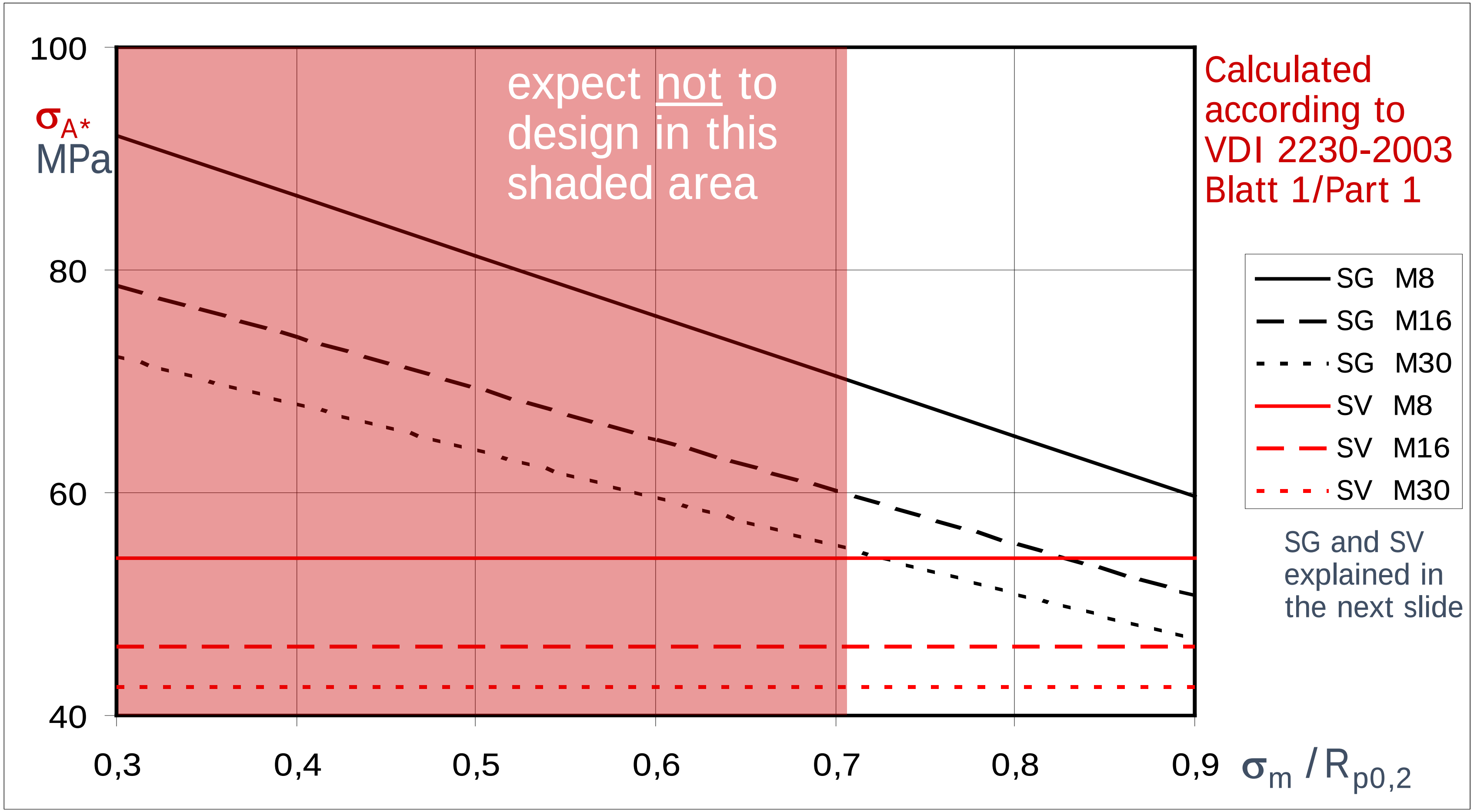
This fatigue limit diagram includes the notch effect due to the thread



Karl-Heinz Decker , Maschinenelemente, Carl Hanser Verlag , 1985

Stress amplitude of the fatigue limit is set at σ_A=0,9 σ_{A*}

This were values given VDI 2230 in editions before 2003. Fatigue limits mentioned here apply at numbers of alternating cycles of more than $N_D \geq 2 \cdot 10^6$.



$\sigma_{A,SG}$: stress amplitude of the endurance limit of SG bolts rolled after heat treatment

$\sigma_{A,SV}$: stress amplitude of the endurance limit of SV bolts rolled before heat treatment

*next slide explains what SG and SV mean

The importance of technology

According to VDI 2230-2003:

- for bolt threads (SV: schlussvergütete) rolled <u>before</u> heat treatment, the stress amplitude of the fatigue limit:
  $\sigma_{ASV} = 0{,}85\ (150/d + 45)$, σ in MPa, d in mm
  (within the validity range of: $0{,}3 \leq \sigma_{Sm}/R_{0,2} < 1$)

- for bolt threads (SG: schlussgewalzte) rolled <u>after</u> heat treatment, the stress amplitude of the fatigue limit:
  $\sigma_{ASG} = (2 - \sigma_{Sm} / R_{p0,2})\ \sigma_{ASV}$
  (within the validity range of: $0{,}3 \leq \sigma_{Sm}/R_{0,2} < 1$)

This higher fatigue strength is a function of the beneficial residual compressive stresses.

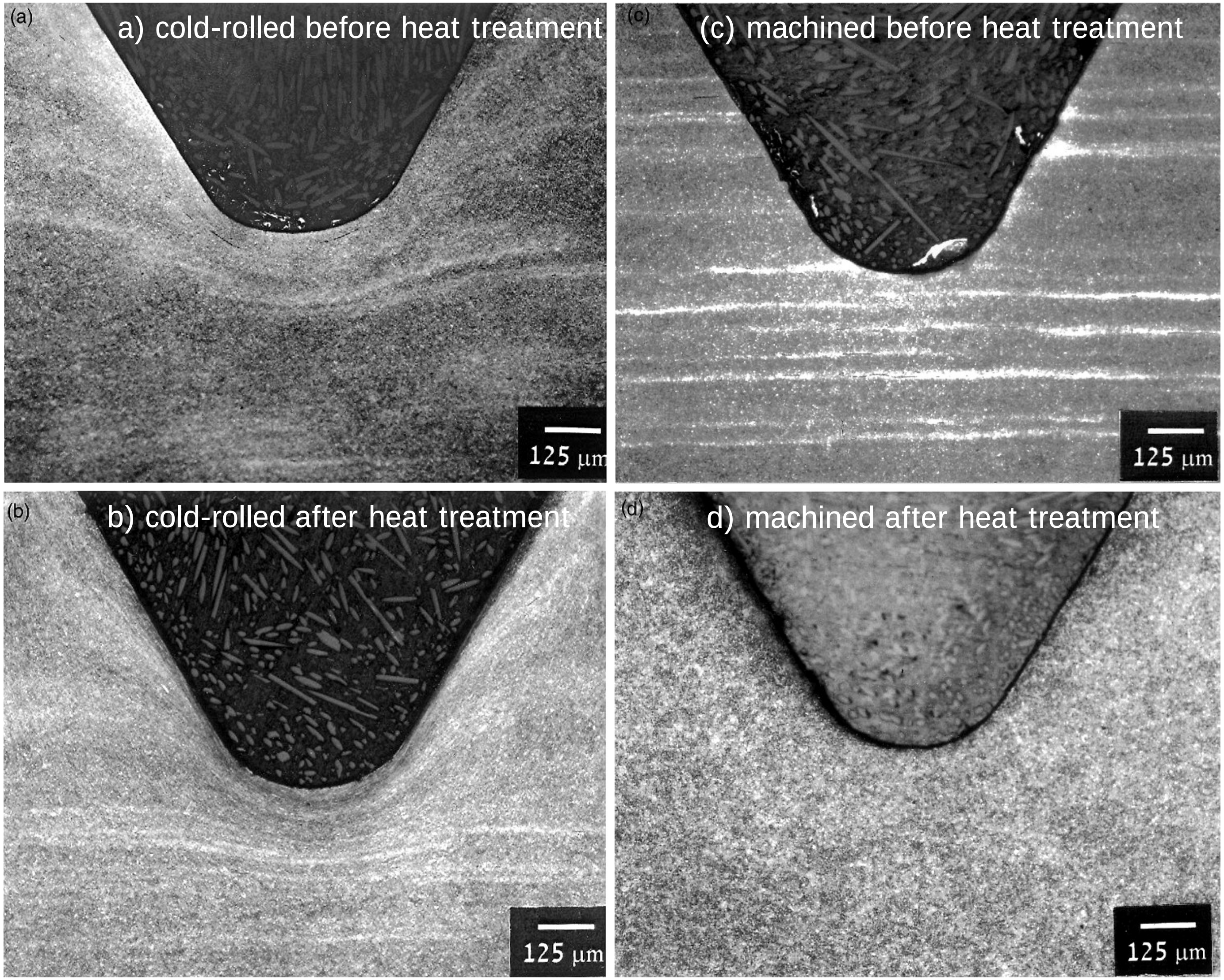
Less expensive because the tool rolls into a softer material; however, the final quenching destroys compressive residual stresses

More expensive because it requires higher quality tools; however, final rolling imparts the desired compressive stresses.

Fine threads vs. standard threads

The fatigue limit of fine threads decreases with increasing strength and the degree of fineness of the threads. For example, in bolts having the strength grade 12.9, it may be down to 30 % lower than in standard threads. (ref. VDI 2230-2003)

14 - Bolt design, fatigue (9/10)



Optical microscope metallographic examinations (micrographs) showing the typical microstructure in the thread region of bolts

S. Iferganea et al. , The effect of manufacturing processes on the fatigue lifetime of aeronautical bolts, Engineering Failure Analysis 8 (2001) 227-235

14 - Bolt design, fatigue (10/10)

The sequence of processes may also affect the fatigue properties of threaded bolts. It has already been reported [1] that fatigue properties are markedly enhanced by rolling the threads after, instead of before, heat treatment. If heat treatment is conducted after rolling, decreased fatigue lifetime results from grain growth at the surface of the threads, anhilation of residual stresses, and accelerated propagation of microcracks that were introduced during thread processing. However, for machined threads, the influence of machining/heat treatment sequence is not evident, but mainly depends on the material and the machining parameters. In high-strength steel, for example, microcracks that had been formed at the thread root during machining were found to propagate during heat treatment, thus reducing the fatigue lifetime [2]. On the other hand, after heat treatment the high-strength steel is less plastic, exhibiting a higher tendency to cracking during machining [3].

Naturally, manufacturers prefer to produce threads (either by rolling or machining) before heat treatment in order to reduce the roller/tool wear and to facilitate the process. Hence, for either quality assurance or failure analysis purposes, it is important to be able to determine unambigiously the type and sequence of processes by which threads were produced. Unfortunately, it is sometimes difficult to determine this from inspection of one batch of bolts.

Source: S. Iferganea et al. , The effect of manufacturing processes on the fatigue lifetime of aeronautical bolts, Engineering Failure Analysis 8 (2001) 227-235

Conclusions

1. In order to maximize fatigue lifetime, it is crucial to fabricate the threads by cold-rolling following heat treatment (in comparison to the other three combinations studied in this work).
2. Fatigue lifetime tests, microhardness testing and metallographic examination may be used as laboratory tools to distinguish between bolts fabricated by cold-rolling following heat treatment and bolts fabricated by any of the other three procedures.
3. Static tensile tests, hardness tests and fractographic examination (using SEM) cannot be used as laboratory tools for such a distinction.