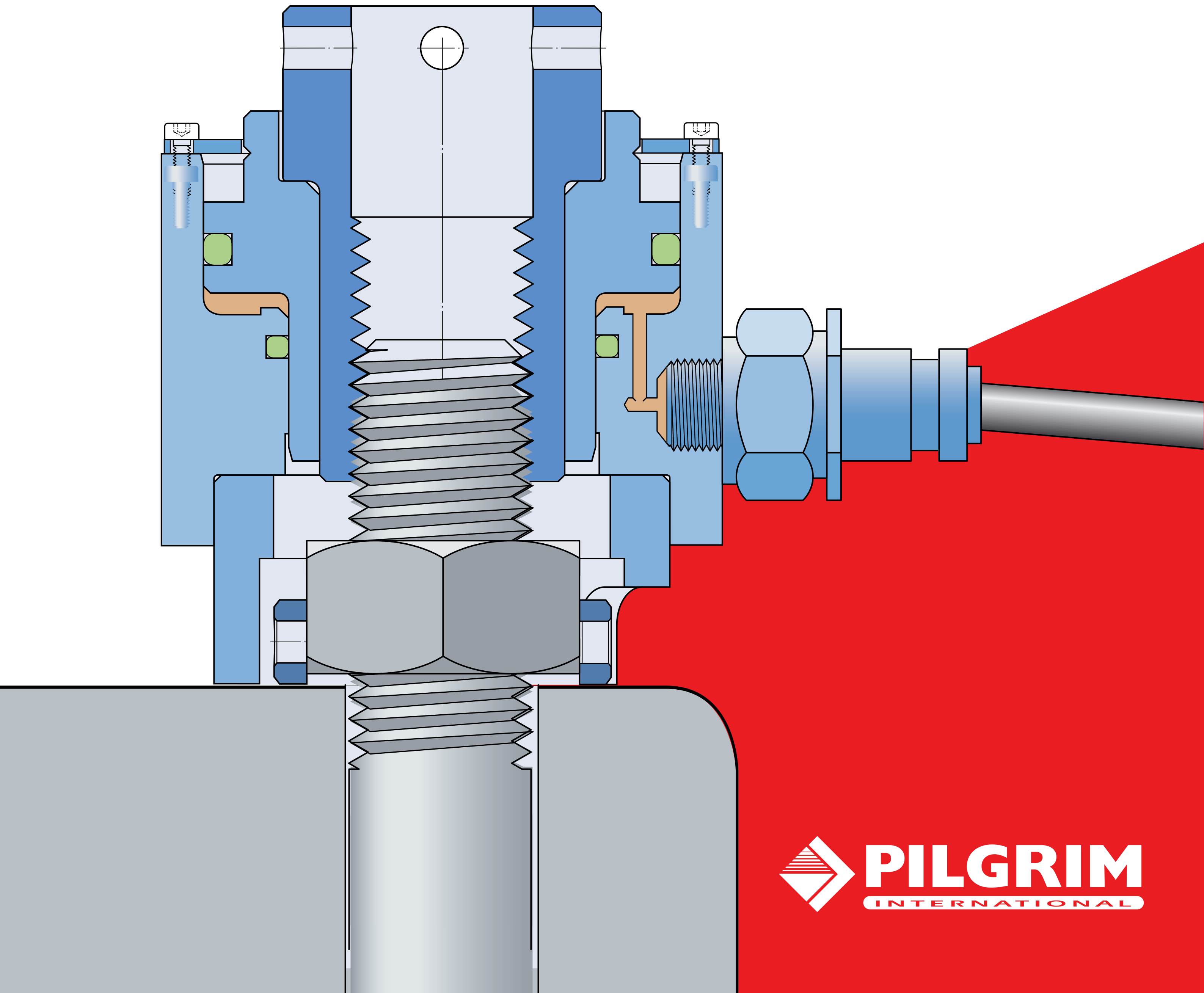
BOLT TIGHTENING HANDBOOK



Contents

The Pilgrim brand now stands for more than ever before, and means more to you as a valued customer.

While Pilgrim maintains its leadership as the hallmark of quality for Controlled Bolting Solutions throughout the world, new dimensions in technical advances, product support and services have evolved Pilgrim into a truly solutions-oriented supplier, creating greater value for customers.

These solutions encompass ways to bring greater productivity to customers, not only with breakthrough application specific products, but also through leading-edge design simulation tools and consultancy services, plant asset efficiency maintenance programmes, and the industry's most advanced supply management techniques.

HYDROCAM

Introduction

Without a doubt, bolted assemblies are the most commonlyused joints in mechanics. These types of assemblies employ two basic elements:

- on the one hand, some kind of threaded component:
  - screws and nuts,
  - studs with nuts on one end,
  - studs with nuts on both ends.

  These components are sometimes used with different kinds of washers (→ **fig. 1**).
- on the other hand, some means for tightening.

These types of tightening means are the subject of this Handbook.

In this document the word “bolt” will be used in a generic sense to cover all three of the types of screwing components mentioned above.

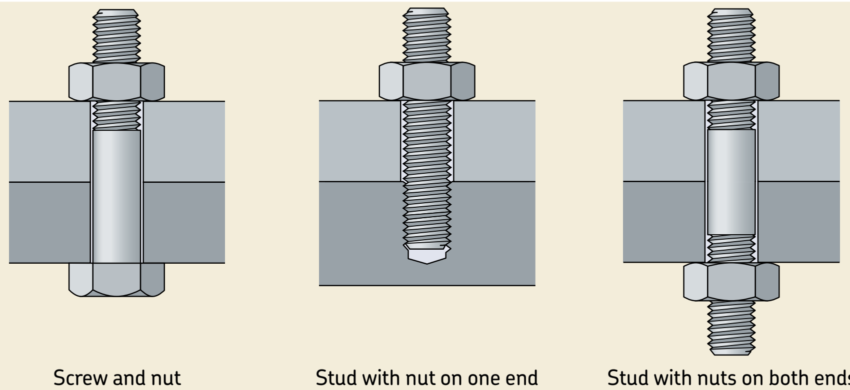
Although bolted assemblies at first appear very simple, they cause several problems for design engineers, assemblers, and maintenance departments.

Rough-dimensioning methods are too often used at the design stage, leading to substantial oversizing of all the components of the assembly, which does not ensure assembly safety, quite the contrary.

In reality, the design of a bolted assembly requires a methodical and rigorous approach, since mistakes can lead to failures with often costly and sometimes disastrous consequences.

Many surveys show that failures of bolted assemblies are mainly due to the fact that they were not properly designed (analysis, drawing, calculation, choice of components) or implemented (tightening method, tooling, checking).

Fig. 1



The surveys also show that among the possible causes of assembly failure (overloading, improper design, manufacturing defects etc.) the most frequent is poor assembly. Undertightening, overtightening and irregulartightening alone cause 30% of all assembly failures.Furthermore, in addition, 45% of all fatigue incidents are due to poor assembly (→ **diagram 1**).

Correct tightening of a bolt means making the best use of the bolt’s elastic properties. To work well, a bolt must behave just like a spring.

In operation, the tightening process exerts an axial pre-load tension on the bolt. This tension load is of course equal and opposite to the compression force applied on the assembled components. It can be referred to as the “tightening load” or “tension load”.

Primary causes of fatigue failure of bolted joints

Diagram 1

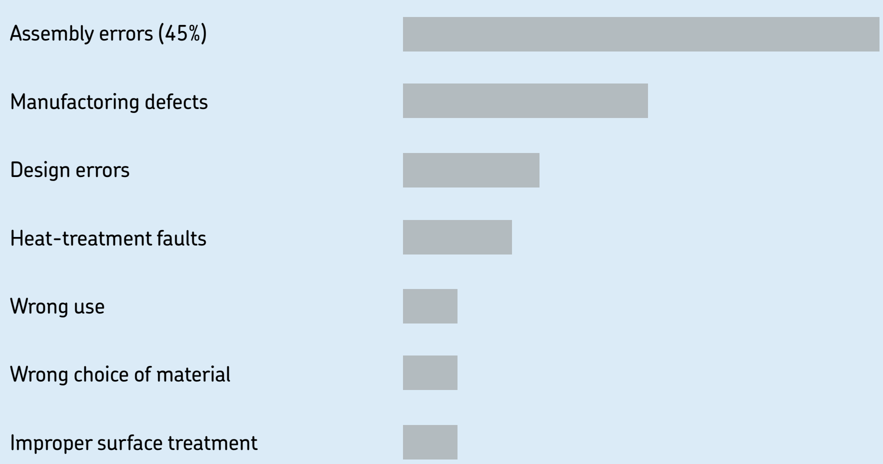
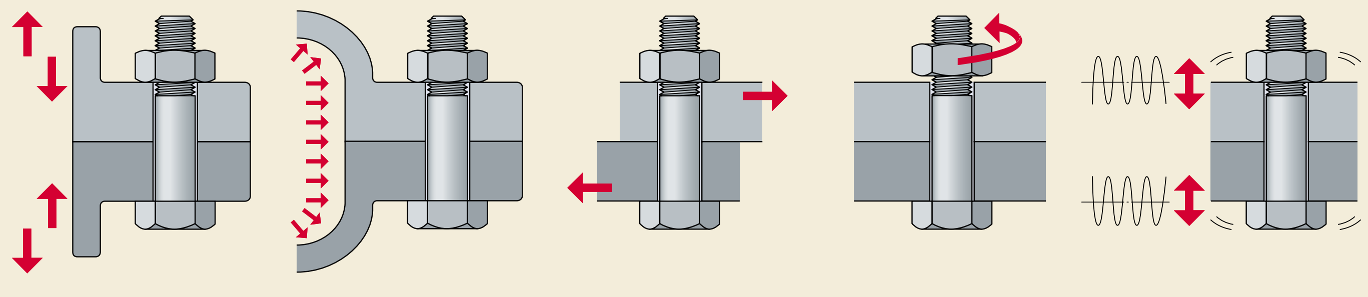


Fig. 2



Depending on the application, the purpose of the tightening load is multiple:

- ensure the rigidity of the whole assembly and make it capable of supporting external loads due to traction, compression, bending moments and shear;
- prevent leakage at seals;
- avoid shear stresses on the bolts;
- resist spontaneous loosening effects;
- reduce the influence of dynamic loads on the fatigue life of the bolts (→ **fig. 2**).

Furthermore, all components (bolts and assembly parts) must perform these tasks while remaining below the yield point of their respective materials.

Bolt-tightening is optimal when the bolt is properly tightened: not too much, not too little! A bolt can fail just as often – and even more so – when it is not tightened enough, as when it is over-tightened.

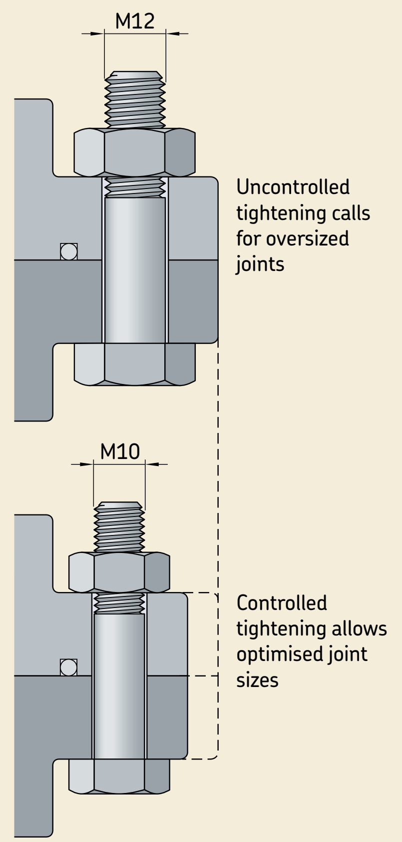
Controlling bolted assemblies

It is fundamental to control the level of the tightening load, as well as the accuracy of the tightening value, to ensure that required performance of the bolted assembly will be achieved.

Complete control over the tightening conditions – from the outset of the design stage – ensures the best use of the bolt's mechanical properties of bolts, (→ **fig. 3**). See the paragraph "Mechanical properties of bolts".

In this "Bolt-tightening handbook" and in the catalogue " HYDROCAM Bolt Tensioners – Industrial Tightening Systems", engineering and design departments will find the theoretical and practical information they need to optimize bolted assembly design and systems operators will find the information they need to control tightening.

Fig. 3



Mechanical properties of bolts

Bolts are most often made of steel. Like most metals, steel is elastic, at least as long as the strain (elongation) does not exceed the "elastic limit" beyond which permanent deformation occurs.

Within the "elastic limit", a metal part such as a bolt follows Hooke's law, that is to say that the strain (elongation) is proportional to the stress (load), as shown on the graph opposite (→ **diagram 2**).

Any tightening method must ensure that the stress in the bolt never exceeds point "A" (the elastic limit or "yield point"), both during the tightening operation and when the assembly is later exposed to efforts during operation (→ **fig. 4**).

When discussing structural mechanics, the following properties of materials will be considered:

E = traction elastic modulus or Young's modulus:

$$E = \frac{F}{S} \Big/ \frac{\Delta_L}{L} = \frac{F_L}{S\Delta_L} = \frac{\sigma_L}{\Delta_L}$$

where:
F = traction force
S = cross-section
L = length
Δ_L = elongation

$$\left(\frac{\Delta_L}{L} = \frac{\sigma}{E} = \frac{F}{S_E}\right)$$

for steel E = 200 000/210 000 MPa

ν = Poisson's ratio or lateral strain index:

$$\nu = \frac{\Delta_d}{d} \Big/ \frac{\Delta_L}{L}$$

for steel: 0,27/0,30
for aluminium: 0,33/0,36
for rubber: 0,49
(the fewest compressible of all solids)
for liquids: 0,5 (almost incompressible)
for cork: 0,0 × (very compressible)

Diagram 2

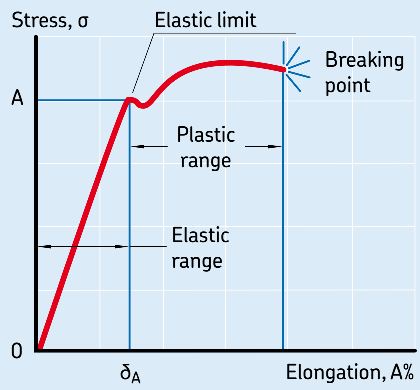
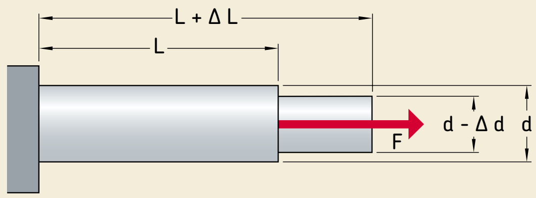


Fig. 4



K = compressibility coefficient (by analogy with liquids):

$$K = \frac{d_V}{d_P} = \frac{3\,(1 - 2\nu)}{L}$$

for liquids: K ≃ 0

G = shear modulus of elasticity:

$$G = \frac{E}{2\,(1 + \nu)}$$

for steel G = 77 000/82 000 MPa

R_m = ultimate tensile stress

R_e = elastic limit, or "Yield Point"

A % = maximum elongation at breaking point.

Traditional tightening methods

There are several methods of tightening bolts. The respective principles are quite different, as are the quality and accuracy levels achieved.

The following is a summary of the most commonly used methods.

The torque wrench

This is probably the most common tightening method. Its main advantage, especially when the bolt diameter does not exceed 30 mm, is that it is very simple and quick to use.

But in spite of theoretical developments and much experimentation, this method suffers from the following major intrinsic drawbacks:

Table 1

Tightening method	Accuracy on pre-load	γ
Calibrated torque wrenches	±20%	1,5
Power tightening tools with regular calibrationon application (measurement of elongationof the bolt or measurement of torque valueusing a calibrated torque wrench)	±20%	1,5
Impact wrenches with stiffness adjustment and periodic calibration on application (measurement of torque value using a calibrated torque wrench per batch)	±40%	2,5
Hand wrenches	±60%	4
Shock wrench (uncalibrated)	±60%	4

$\gamma = \frac{F_{0\,max}}{F_{0\,min}}$ = uncertainty factor on tightening load.

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Accuracy of the tightening load for various tightening methods using torque

Characteristics of torque tightening

High amount of uncertainty as to the final bolt tension load

The final tightening load depends on the friction coefficients in the threads of the nut and the bolt, and on the bearing-contact surfaces between the nut and the flange.

In practical terms, it is impossible to know the value of these coefficients accurately and reliably.

For a given nominal torque value, the deviation in the final tightening load of the bolt can vary between ±20% when conditions are good, and ±60% when conditions are bad (→ **table 1**).

This wide range is due to the combination of the following three phenomena:

- the tolerance in the applied torque, which can vary from ±5% to ±50%, depending on the tool (→ **table 2**);
- geometric defects and surface roughness on the threads and the bearing surfaces of the fastened components;
- degree of lubrication of bearing surfaces.

Incorporation of additional "parasite" torsion stress

In addition to the desired axial tension stress, torque tightening introduces a "parasite" torsion stress in the bolt which can reach over 30% of the tension stress.

The resulting equivalent stress in the bolt (Von Mises or Tresca criteria) is greatly increased and can exceed the yield point of the material, whereas the tension stress itself remains within admissible limits (→ **fig. 5**).

Furthermore, the residual torsion stress increases the risk of spontaneous loosening at a later stage.

Furthermore, since the torque is most often applied in a non-symmetrical manner, there is also some bending stress, but because its value is comparatively small, it is often ignored. However, in cases where the working conditions are near the limit, this bending stress should be taken into account.

Table 2

Accuracy range of torque tightening method	Equipment type Manual hand tool	Portable power tool	Non-portable power tool	Usage limits
D ±20% to ±50%		Simple shock wrenches		≥ 50 Nm
		Power tightening tools with positive clutch		≤ 50 Nm
C ±10% to ±20%		Power tightening tools with pneumatic adjustment		≤ 10 Nm
		Power tightening tools with electric adjustment		≤ 10 Nm
		Impact wrenches with stored energy (torsion bar or other means)		≥ 10 Nm
		Adjustable wrenches with angle drive		≤ 20 Nm
	Calibrated wrenches with simple release device			≤ 400 Nm
			Simple air-driven tools	No limits
B ±5% to ±10%			Hydraulic screwing tools	–
	Calibrated wrenches with release device and automatic resetting			≤ 800 Nm
	Calibrated wrenches with dial gauge			≤ 2 000 Nm
		Wrenches with angle drive and release device		≤ 80 Nm
			Air-driven tools with controlled torque	No limits
			Air pulsed tools	No limits
A < ±5%			Electric power tightening tools	No limits
	Electronic calibrated wrenches			≤ 400 Nm
			Dual-speed motors	No limits
			Servo controlled motors	No limits

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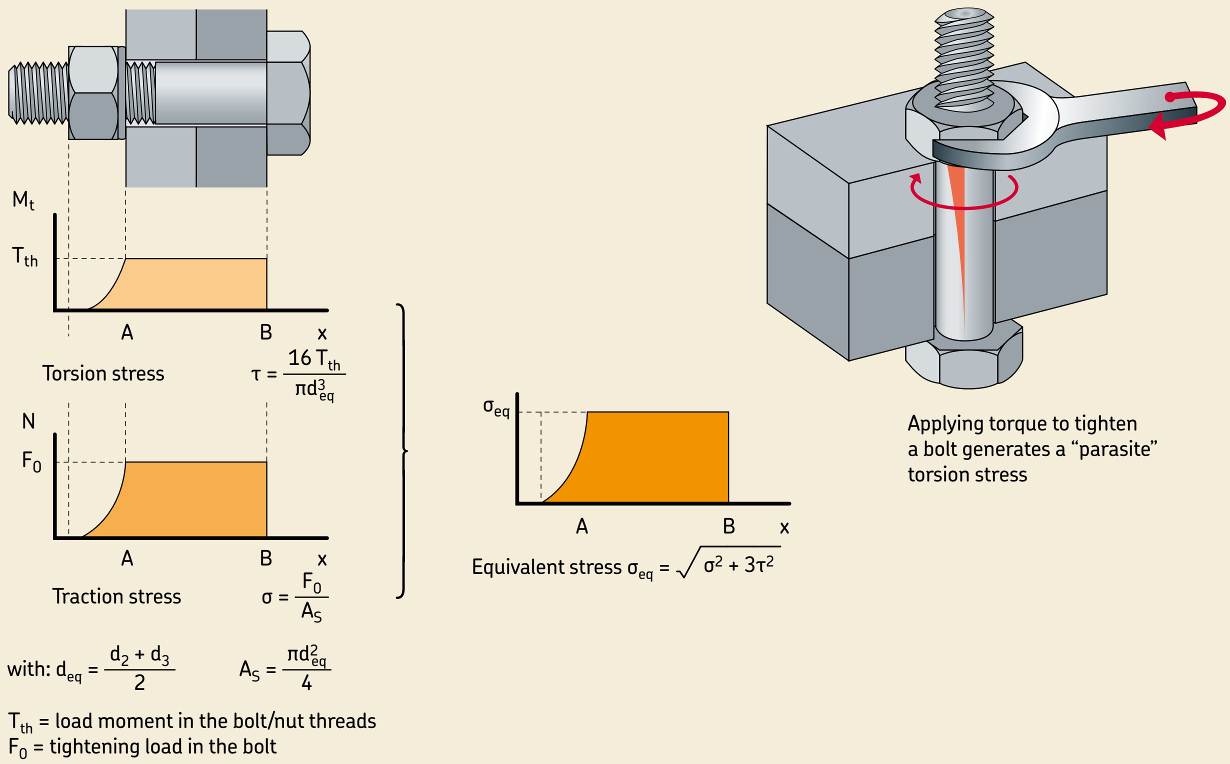
Deviation on torque in industrial applications

Fig. 5



Damage to bearing surfaces

Friction between parts under very heavy loads leads to galling and damage to the friction surfaces, namely the threads between nuts and bolts, and the bearing surfaces between nuts and flanges.

At the next tightening operation, such damage will increase the friction forces, and the error in the final tightening load will increase accordingly (→ **fig. 6**).

Difficulties in untightening

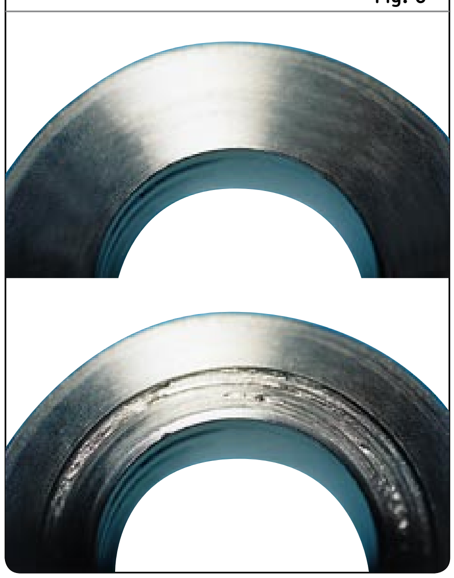
It is often much more difficult to unscrew a torqued bolt than it was to screw it on in the first place. Damage to the contact surfaces, and corrosion problems, impose higher torque loads, which can cause damage to various parts of the assembly.

Problematic tightening of large bolts

When the required tightening torque exceeds 1 000 Nm, various torque equipment must be used, such as impact wrenches, torque multipliers or hydraulic torque wrenches (→ **figs. 7** and **8**).

This equipment provides the required tightening torque. However, with the impact wrench in particular, the accuracy is unreliable. Only the hydraulic torque wrench – on the condition that top-quality equipment be used by skilled operators following correct procedures – can provide some improvement in accuracy.

Fig. 6



Damage to bearing surfaces
Tightening with a wrench causes damage to the surfaces of the assembly components. Successive assembly and disassembly increase this phenomenon.

Fig. 7

Torque multiplier

Hydraulic torque wrench

Fig. 8

Simultaneous tightening is rarely possible

With the torque method, it is generally not possible to simultaneously tighten several or all of the bolts in an assembly.

When hydraulic torque wrenches are used, several bolts can theoretically be tightened at the same time. However, only a few bolts can actually be connected at one time because of space limitations and installation difficulties.

Furthermore, this does not eliminate the inaccuracy problems decribed above.

Methods and devices for measuring final preload when tightening by torque

It is possible to reduce the deviation on the final tightening load by using an instrument to measure either the torque or the resulting bolt elongation.

But whatever the means of control, is must not be forgotten that any torque tightening method increases the equivalent stress level because of the “parasite” torsion stress.

Monitoring the torque value

This is the simplest method. However, as described above, even where the accuracy of the applied torque value is good, a great deal of uncertainty still remains as to the final tension load in the bolt.

Checking by the angle of rotation of the nut

There are two steps to this method. First, the nut is tightened to a torque value which is slightly lower than the required final torque. Then, a further, specific angle of rotation is apllied.

This slightly reduces the deviation in the final tension load. However, the uncertainty remains high, and the “parasite” torsion stress can be significantly increased.

Bolt-elongation measurement methods

The accuracy is significantly improved when direct bolt-elongation measurements are taken. Several methods can be used:

Rod and knurled-wheel method

Arod topped with a knurled wheel is screwed into a hole bored in the middle of the bolt.

When the bolt elongation equals the initial clearance left between the top of the rod and the knurled wheel, the wheel’s rotation is blocked, thereby informing the operator that the bolt is tightened.

This method has certain disadvantages:

- the extra cost of the additional parts and the drilling ;
- the bolt is weakened;
- need for preliminary calibration;
- uncertain degree of accuracy, in particular becausethe operator must turn the knurled wheel a little to checkthe remaining clearance.

Measurement by dial gauge or LVDT

The entire length of the bolt is drilled through to house a measuring rod. The variation in the distance between the top of the rod and the top of the bolt is measured with a dial gauge or an electronic sensor (LVDT).

This method is more accurate than the previous one, but it has similar disadvantages:

- the extra cost of drilling the bolt, and the extra parts;
- the bolt is weakened;
- preliminary calibration is required.

Ultrasonic measuring (US) method

This consists in measuring the time it takes an ultrasound wave to travel down and back the longitudinal axis of the bolt.

The bolts are not drilled but they must be top-quality, and careful calibration is required.

The method requires qualified, skilled operators.

Constant improvements are making this method increasingly attractive, in particular for small-size screws (diameters under 20 mm).

Strain-gauge method

Strain gauges are attached to the bolt and connected to a Wheatstone bridge; the variation
in the signal – which corresponds to the strain variation in the bolt – is measured. Preliminary calibration is required.

This is strictly a laboratory method and can not be used in industrial applications.

In conclusion, the methods described above require specialised technicians and can be long to implement.

Furthermore their accuracy is directly proportional to their cost. In addition, they do not directly measure the tension tightening load, but rather the variation in the bolt elongation.

B

The sensor washer

The sensor washer offers a major advantage compared with other methods, since it measures the tightening load directly.

The "sensor" washer (→ **fig. 9**) is an instrumented washer installed under the nut on which the torque is applied.

It is highly recommended to install a common thin washer between the sensor washer and the nut, to avoid high friction on the sensor washer during tightening and untightening.

This washer acts as a load-cell sensor.

The accuracy is good and the method is easy to use.

When a torque wrench is used, the friction forces may vary greatly from one bolt to another for the same assembly. If good accuracy is required, each assembly bolt should be fitted with a sensor washer.

Furthermore, this method provides easy periodic or permanent measurement and recording of the bolt tensile stress when the assembly is both on and off.

The "sensor" washer

Fig. 9

Changing from torque wrenches to hydraulic bolt tensioners

Using a hydraulic bolt tensioner entails knowing (F_0) the residual load applied to the bolt. However, when tightening with the torque wrench method, only the torque is recommended. It is usually expressed in Newton-metres (Nm). The following theoretical formula provides a preliminary approximation of the residual tension load when a torque value is applied to a bolt. This formula was obtained by taking into account the friction of the threads and the friction of the nut face against the flange.

$$F_0 = \frac{T}{0{,}16\ p + \mu_1\ 0{,}582\ d_2 + \mu_2\ r_m}$$

where:

F_0 = residual tightening load
T = tightening torque
p = thread pitch
μ_1 = friction coefficient of the bolt/nut threads
μ_2 = friction coefficient at the nut face/flange
d_2 = equivalent diameter of the bolt
r_m = average radius of the nut face.

The **Chapter E** "Comparison between the torque wrench and hydraulic bolt-tightening" describes a real application of this formula.

Tightening with heater rod

This method consists of elongating the bolt by heating it with a heater rod inserted down the bolt centre. It then suffices to turn the nut under low torque force until it is in contact with the flange.

Upon cooling, the bolt will contract lengthwise, thereby tightening the nut. Simultaneous tightening of several bolts is theoretically possible. The method is theoretically accurate but in fact has several disadvantages:

- A hole must be drilled down the centre of the bolt to receive the heating rod.
- Heating systems, electrical connections, temperature-control devices and handling means are required, especially in the event of simultaneous tightening.
- The method is exceedingly slow, due to the time required to heat the bolts, and the final tightening load can only be checked after the bolts have cooled down, which takes even longer.

The process cycle includes: heating the bolt, advancing the nut, cooling down the parts, and measurements. This cycle must be repeated several times in order to adjust the tightening.

The temperature required to reach suitable elongation is often so high that it could modify the mechanical properties of the equipment. As a result, when thermal elongation is insufficient, additional torque tightening must be performedand verified by measuring the nut angle.

This thermal elongation technique is fairly rarely used, and is generally only applied to large sized bolts (diameter > 100 mm).

Changing from heater rods to hydraulic bolt tensioners

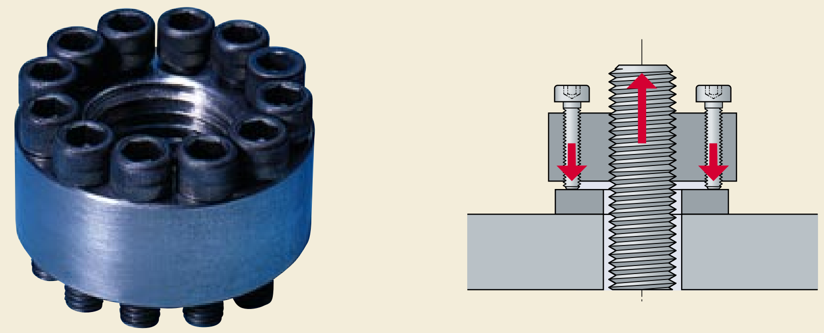
As we have seen, the use of hydraulic bolt tensioners requires knowing the residual load applied to the bolt. The technical data known for heating rod tightening are as follows: the temperature increase (Δt), and the angle of rotation (θ) to apply to the nut once the temperature is reached. The technique is based on the type of strain: the "natural" deformation due to the heat of the bolt, which depends on the expansion coefficient (α) of the steel, and the deformation due to torque tightening, i.e. the rotation of the nut. The following theoretical formula allows estimating the residual tension load (F_0) in the bolt when this technique is applied:

$$F_0 = \alpha\, S_E \Delta t + \frac{\theta p S_E}{l}$$

where:

θ = angle of rotation of nut
Δt = temperature increase
α = expansion coefficient of the bolt steel
p = thread pitch
S = section of bolt
l = tightened length of bolt
E = elastic modulus (Young's modulus) of the bolt steel

Fig. 10



Tightening by mechanical elongation

Tightening by mechanical elongation

With this method, the tension load is directly applied to the bolt (→ **fig. 10**).

In general, the body of the nut is provided with a set of small thrust screws located symmetrically around the main threaded hole. These screws apply – either directly or through a washer – a bearing pressure on the contact surface of the flange.

They are turned one by one and step by step using a low torque load until a suitable tension load for the bolt is reached.

The bolt elongation is most often measured using one of the previously mentioned methods. In spite of the fact that this method eliminates torsion stress in the bolt, it has several drawbacks:

- Simultaneous tightening is not easy to carry out: only a step-by-step tightening process is reasonably possible, from one bolt to the next. This is both tedious and time-consuming, and the result is pseudo-simultaneous tightening.
- To precisely determine whether tightening was carried out correctly, an additional measurement means must be provided, such as the elongation method or the use of load-measuring washers.
- The nuts are generally expensive, since they are bigger and require several small thrust screws and machining of several threaded holes
- The process is very slow because to properly thighten each bolt, the small screws have to be hand-tightened in several passes.

For all of these reasons, the mechanical elongation method is not used frequently.

Tightening with hydraulic bolt tensioners

Description

This tightening method uses Pilgrim HYDROCAM bolt tensioners (→ **fig. 11**).

The bolt must have an end that protrudes above the tightening nut. Cold extension is applied to the bolt by means of an annular hydraulic cylinder placed around it. The bolt undergoes an axial traction load only.

The stress-free nut is then turned down with very little effort and does not transmit any torque to the bolt. When the fluid pressure is released in the tensioner, the major part of the hydraulic load on the tensioner is transferred into the nut, and tightening is completed (→ **fig. 12**).

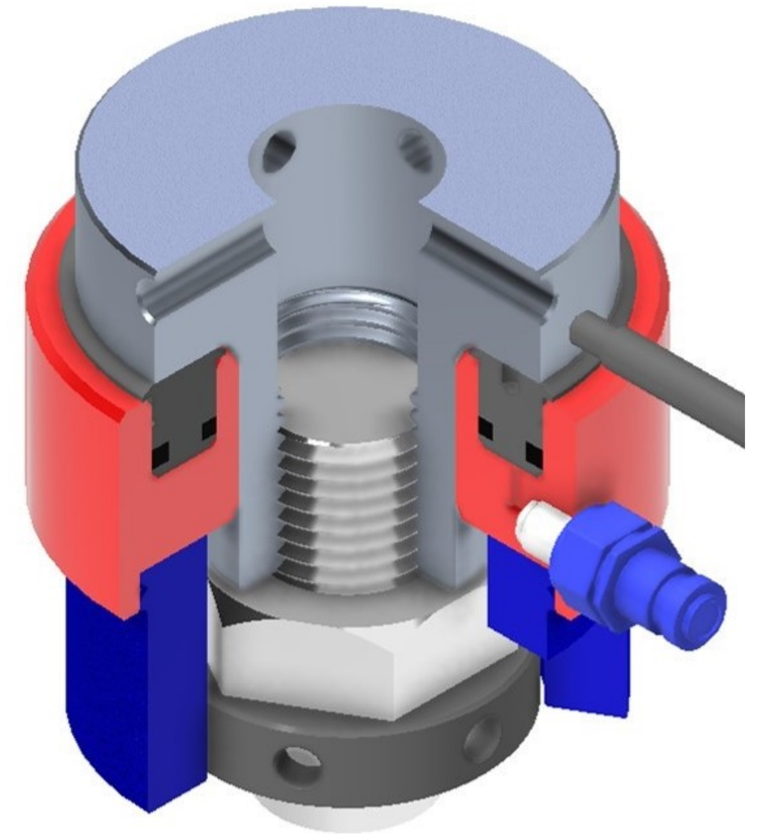
For optimum accuracy, it is recommended to perform traction of the bolt and turning-down of the nut twice.

In effect, the first turning-down operation compensates for clearances, compresses the roughness of the surfaces and sets the load balance, while the second operation serves primarily to obtain the required accuracy of residual load inthe bolt.

This double turn-down operation simply consists of repeating stages 3, 4 and 5 (→ **fig. 12**).

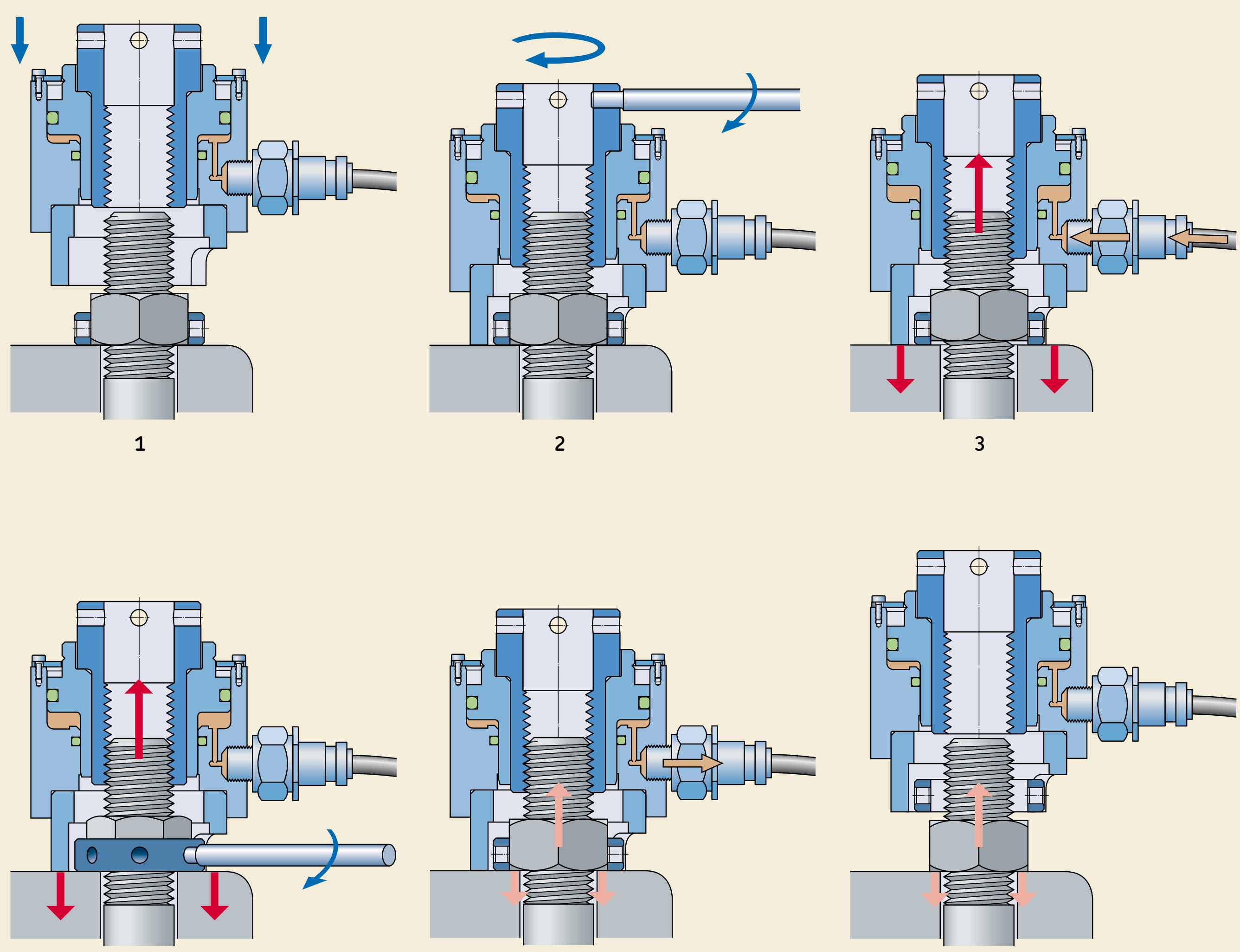
When performed in a workmanlike manner, this method represents the best method of meeting the quality requirements of correct tightening, as defined in the introduction.

Fig. 11



Pilgrim HYDROCAM bolt tensioner

Fig. 12



__Operating principle of the HYDROCAM bolt tensioner__

1 *The turndown socket is placed over the nut and the hydraulic tensioner grasps the bolt.*
2 *The brace/retraction unit is screwed onto the protruding end of the bolt.*
3 *After the hydraulic connections, the tensioner is pressurised and applies the required tractive force on the bolt.*
4 *While the pressure is maintained, the nut is turned down without loading, using the socket and the tommy bar.*
5 *Their pressure is released and the piston is pushed back. The tightening load is now exerted through bolt tension.*
6 *The tensioner and the socket can be removed.*

Advantages of hydraulic bolt-tensioning

No torsion stress

The method eliminates all "parasite" torsion or bending stress in the bolt (→ **fig. 13**).

Good accuracy

It is accurate because the most important parameter, namely the traction load, is perfectly controlled through the hydraulic pressure in the tensioner. The load does not depend on the various friction coefficients in the assembly. The only uncertainty in the method arises from the degree of torque applied when driving down the nut. However, this torque's influence is, by definition, of secondary importance.

Through simple precautions of good workmanship, good uniformity can be obtained in the driving-down operation. In addition, the F_h/F_0 ratio must be taken into account (hydraulic load/residual load after pressure release). This ratio is described in detail on (→ **page 21**). Proper understanding of this ratio is important, for there are means available to obtain an accurate ratio for each assembly.

Easy implementation

This method is easy to perform and requires no physical effort, even for very large bolts. Occupational hazards and physical exertion are significantly reduced.

Suits a wide range of bolt diameters

The method can be used for a very wide range of bolt diameters, from 5 to 500 mm!

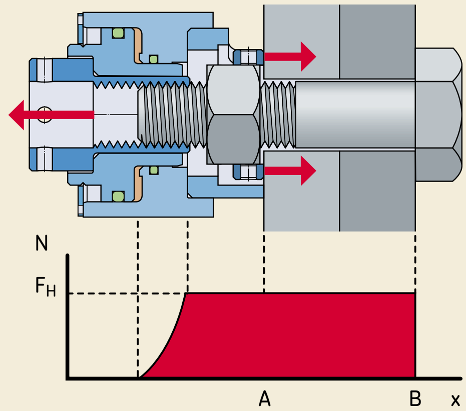
Material Variety

Many different bolt materials, such as stainless steel, titanium, composite materials and others, can very easily be tightened with the hydraulic tensioning.

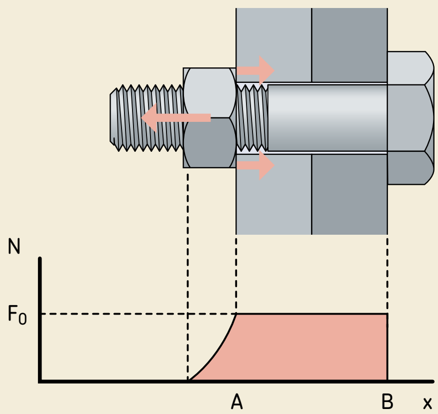
No damage to components

Internal stresses are controlled, and no friction is generated under heavy bearing pressure. Therefore this method protects the individual components of the assembly (→ **fig. 14**).

Fig. 13



During tightening
F_H = hydraulic tension load ⇒
σ_H = traction stress due hydraulic tension F_H



After tightening
F_0 = final tension load in the bolt ⇒
σ_0 = final traction stress after turning down at the nut and pressure release

Easy untightening

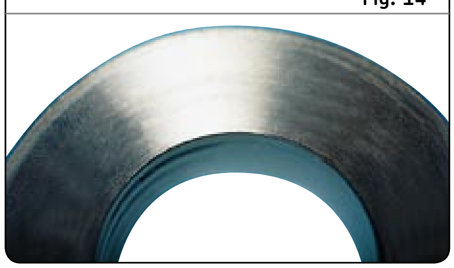
The untightening operation is extremely easy: generally the required hydraulic effort is approximately one percent greater than that required in the previous tightening phase.

Simultaneous tightening is possible

The HYDROCAM tensioner tightening method allows simultaneous tightening of several or all of the bolts in a given assembly.

No damage to components
Tightening with a hydraulic bolt tensioner preserves the condition of the components, no matter how many successive tightening and untightening operations occur.

Fig. 14



The advantages are:

- equal tightening of all bolts in the assembly,
- simple process,
- reduced work time (see **chapter F** "Simultaneous tightening by hydraulic traction").

Process automation is possible

The tightening and untigthtening operations can be automated, thereby providing:

- optimisation on the simultaneous operations,
- better tightening accuracy,
- even distribution of tightening forces,
- reduced tightening time,
- better working conditions for the operators in cases of difficult access,
- remote control.

The remote control feature allows operators to control all phases of the tightening or untightening procedures, from a safe area. This significantly reduces exposure to poor or dangerous working conditions, such as radiation, hazardous media, high temperatures, high noise levels, risk of component failure.

Measurement devices for hydraulic bolt tensioning

Several different control methods can be used to check the hydraulic bolt tensioning, depending on the required accuracy for the assembly:

Measuring the hydraulic pressure

Once you have determined the F_h/F_0 ratio, the precise measurement of the pressure applied to the hydraulic cylinder provides an 8 to 10% accuracy level for the final tightening load, which is often quite acceptable.

The "double pressurisation" method

Once the bolt has been tightened by the initial cycle of pressurisation on the tensioner, turning down the nut and releasing of pressure, a second pressurisation is applied to take a more accurate measurement of the bolt load. During this second pressurisation, the rising pressure value is recorded and plotted versus the displacement of the top part of the bolt. The resulting tension load of the bolt can be determined graphically using the change in the slope of the graph. (→ **fig. 15**).

This method improves control over the final residual load.

Final tightening load accuracy may be on the order of 5 to 8%.

Elongation measurement

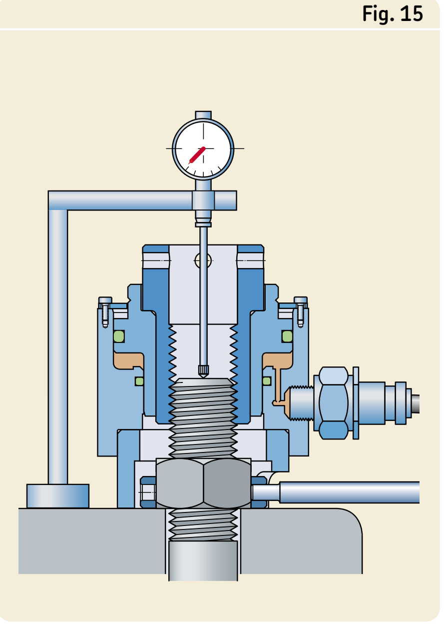
This measurement can be performed with one of the previously described methods dial gauge or LVDT (→ **fig. 16**), or the ultrasonic method (→ **fig. 17**).

With precisely machined bolts and accurate, well-calibrated instruments, the uncertainty in the final tightening load can be as low as 1 to 5%.

C

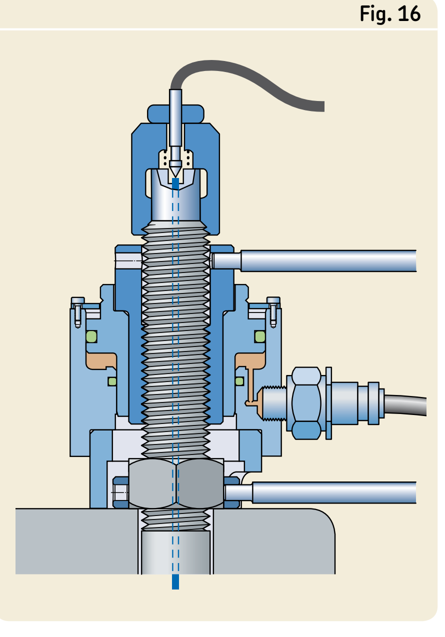
Measurement of the bolt tip displacement with a dial gauge.

Fig. 15



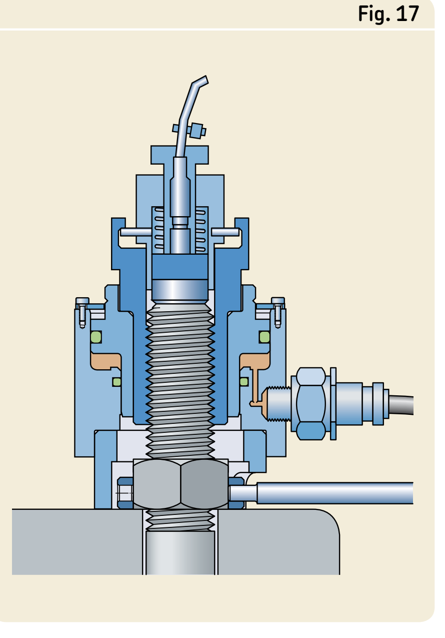
Measurement of the bolt elongation with an internal rod and an LVDT sensor.

Fig. 16



Ultrasonic measurement of the bolt elongation.

Fig. 17



The “sensor washer”

This method, which was previously described for torque tightening, is perfectly suited to hydraulic tensioning and provides an accurate measurement of the final tension load in the bolt.

Since this sensor washer remains in the assembly, it is particularly useful for periodical or continuous monitoring of load variations over time, where required.

Unlike torque tightening, hydraulic tensioning, which is reproducible, does not require one sensor washer for every bolt. One washer every two, three, four or eight bolts etc. may suffice, depending on the requirements.

With sensor washers, the measurement precision as to the bolt load is about five percent, but it can be improved to two percent by careful machining of the assembly components.

Note that no additional classic washers are required between the tightening nut and the sensor washer, since hydraulic tensioning generates no surface friction.

The hydraulic bolt-tightening method provides good control over the tension stress. If the bolt is long enough, the final tension stress can be safely brought very close to the yield point without any risk of exceeding the latter.

This point is very important for the quality of the joint. Contrary to what has been commonly believed, case studies show that the closer the bolt stress to the yield point:

- the better the behaviour of the joint,
- the longer the life of the bolts under cyclic loads (→ **diagram 3**).

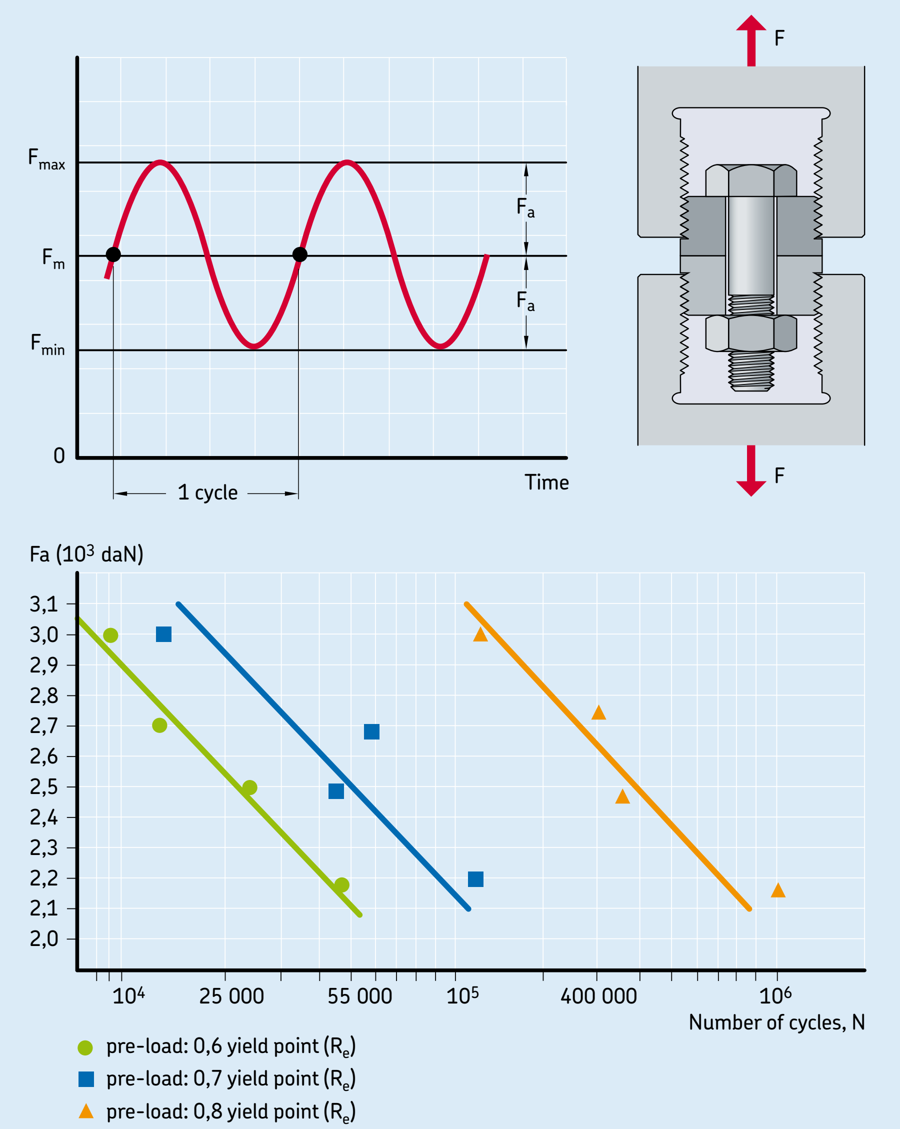
Controlling the stress in the bolt enables material choice and dimensioning to be optimised at the design stage.

Compared with traditional methods using torque-tightening, hydraulic bolt-tensioning reduces the dimensions of the assembly and or the number of bolts. This in turn offers three major advantages:

- a reduction in overall dimensions,
- a reduction in weight,
- a reduction in cost.

Influence of the bolt-tightening rate on the dynamic performance of the bolt.

Diagram 3



(Source : CETIM "Assemblages vissés conception et montage". Reproduced by permission of CETIM.)

Technical analysis of bolt-tensioning

Whatever the method used to tighten a bolt such as the assembly shown in **fig. 18**, the goal is in fact to apply a traction load to the bolt, and a compression load to the assembled components. In general, the bolt has a relatively low stiffness compared with that of the structural parts on which the compression stress is applied.

Diagram 4 shows that line D_1, which corresponds to the bolt, has only a slight slope, whereas the slope for line D_2, which corresponds to the structure (→ **diagram 5**) is steep.

Lines D_1 and D_2 can be drawn on the same graph, and the final tightening load F_0 is shown by the intersection of the two lines (→ **diagram 6**).

Naturally, the tension in the bolt has the same value but exactly the opposite sign as the compression of the structure (→ **fig. 19**).

The elongation of the bolt is δ_B and the compression of the structure is δ_S.

As soon as external traction load F_E is applied to the assembly, the tension in the bolt is not increased by F_E but only by F_1 because the compression of the structure is reduced by F_2, this gives (→ **fig. 20**):

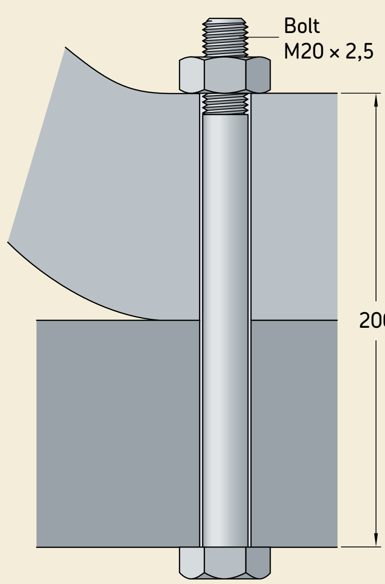
$$F_E = F_1 + F_2$$

It can be seen on the graphs that not all of the external load is applied to the bolt, but only a fraction of it (→ **diagrams 7** and **8**).

D

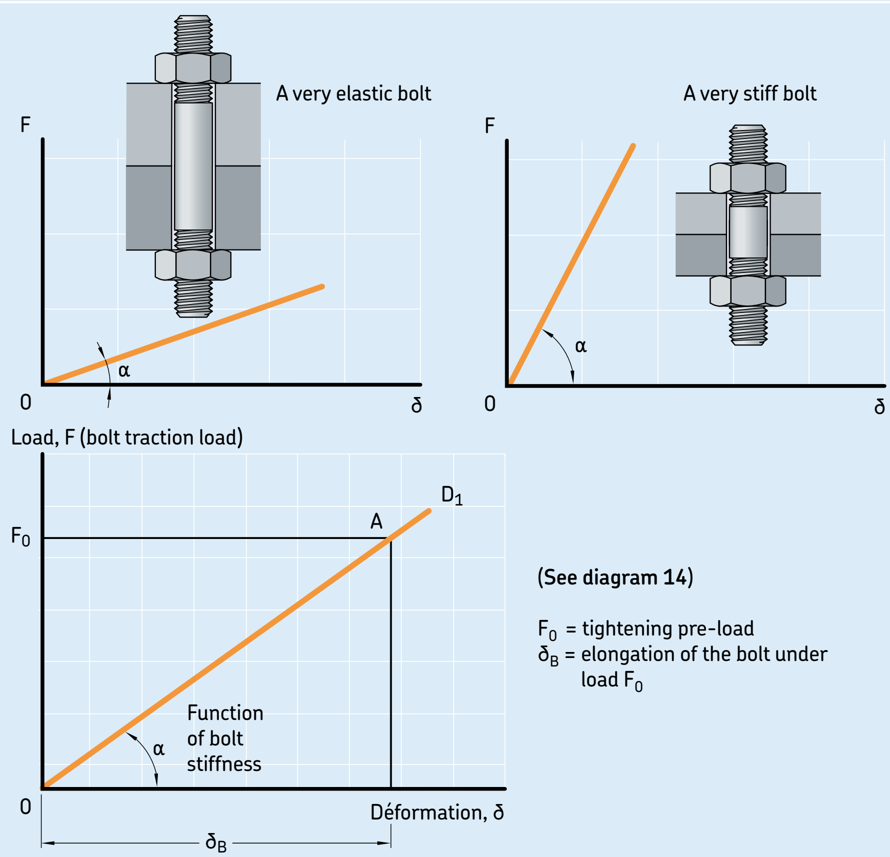
Example of bolted assembly

Fig. 18



Tensile diagram of the bolt

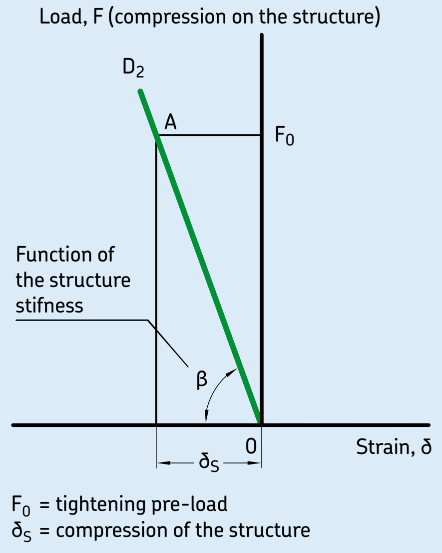
Diagram 4



(See diagram 14)

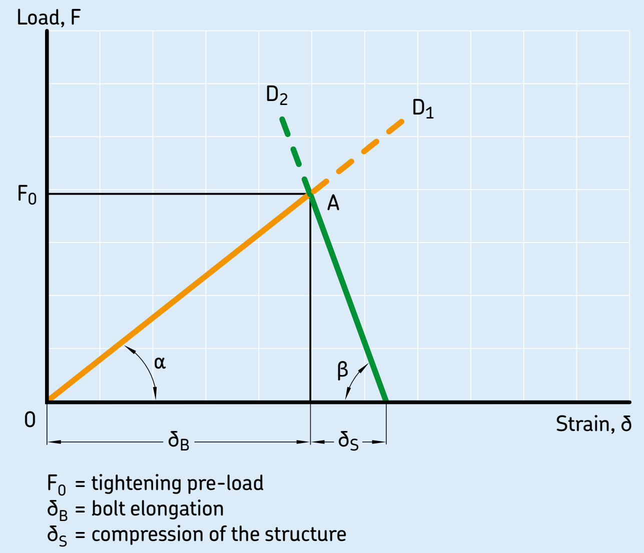
F_0 = tightening pre-load
δ_B = elongation of the bolt under load F_0

Diagram 5



Compression load/deformation diagram of the structure

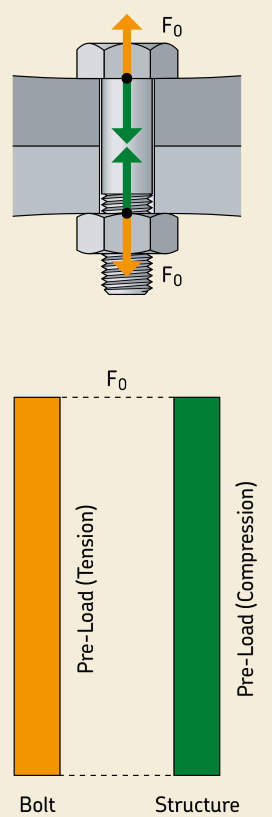
Diagram 6



Load/deformation diagram of a bolted joint

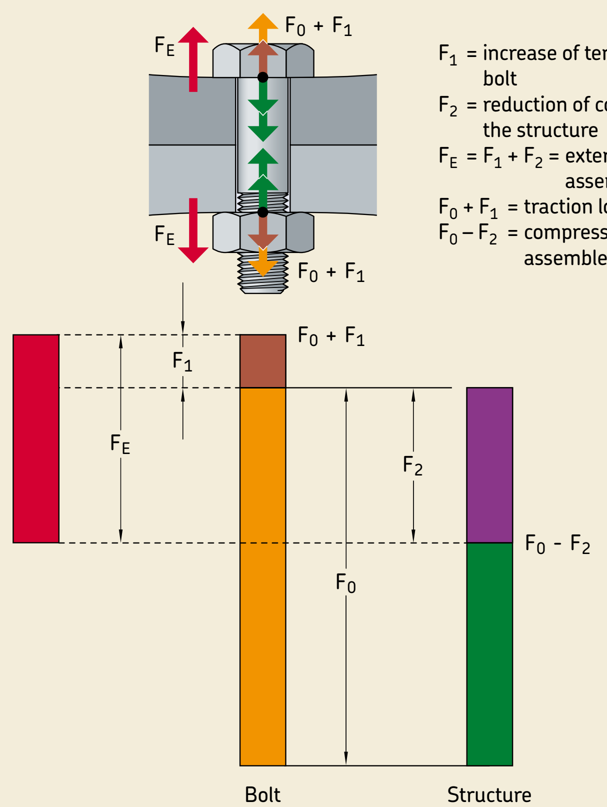
Bolted joint tightened to a pre-load F_0

Fig. 19



Bolted joint with tightened pre-load F_0 and external traction load F_E

Fig. 20



F_1 = increase of tension load applied to the bolt
F_2 = reduction of compression load applied to the structure
$F_E = F_1 + F_2$ = external traction load on the assembly
$F_0 + F_1$ = traction load on the bolt
$F_0 - F_2$ = compression load between assembled parts

For a compressive external load F_E the **diagram 7** is replaced by the **diagram 9**.

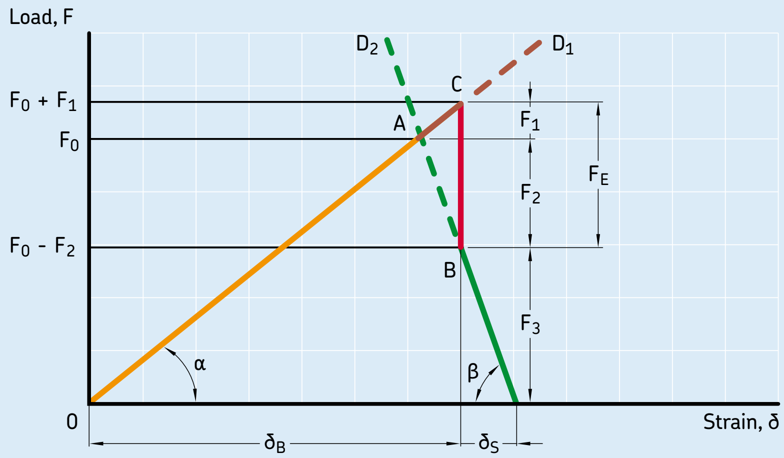
The F_1 part of the load which is taken up by the bolt can be calculated as follows and depends on the bolt stiffness R_B and the R_S, stiffness structure. This gives:

$$F_1 = F_E \, R_B / (R_B + R_S)$$

If the tightening load is insufficient compared with the external load, tightening is lost (→ **diagrams 10** and **11**).

In the case of cyclic external loads, the **diagrams 12** and **13** show that the actual alternating load on the bolt is very low. This is very important since we know that the alternating fraction of the load has a strong influence on the fatigue of the material (→ **diagram 3**).

Diagram 7

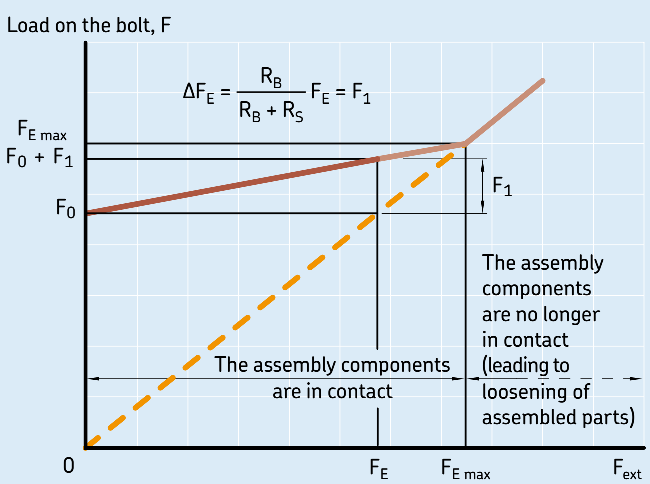


F_0 = tightening pre-load
F_E = external traction load on the assembly
$F_0 + F_1$ = traction load on the bolt
$F_0 - F_2$ = compression load between assembled parts

Diagram showing the effect of an external traction load F_E on a bolted assembly with a pre-load F_0

Relationship between the total load introduced in the bolt and the external load applied to the assembly

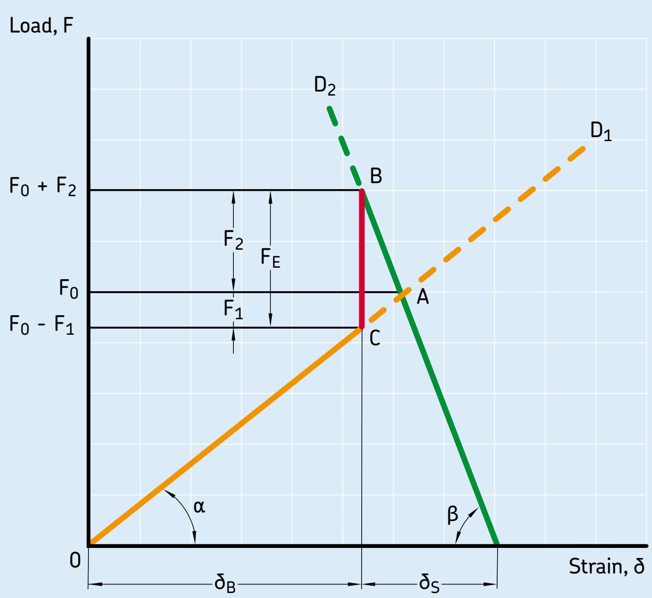
Diagram 8



F_0 = tightening pre-load
$F_0 + F_1$ = traction load on the bolt
F_E = external traction load on the assembly
$F_{E\,maxi}$ = maximum admissible external load on the assembly before looseness
ΔF_E = traction of the external load taken up by the bolt
R_S = stiffness of the structure
R_B = stiffness of the bolt

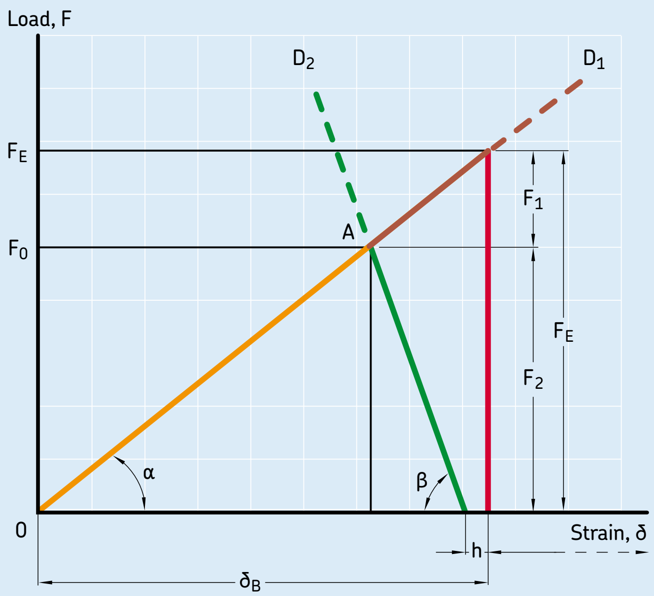
A compressive external load F_E on the assembly induces a reduction in the bolt tension load and an increase in the compression strain in the structure

Diagram 9



F_0 = tightening pre-load
F_E = external compression load on the assembly
$F_0 - F_1$ = traction load on the bolt
$F_0 + F_2$ = compression load on the structure

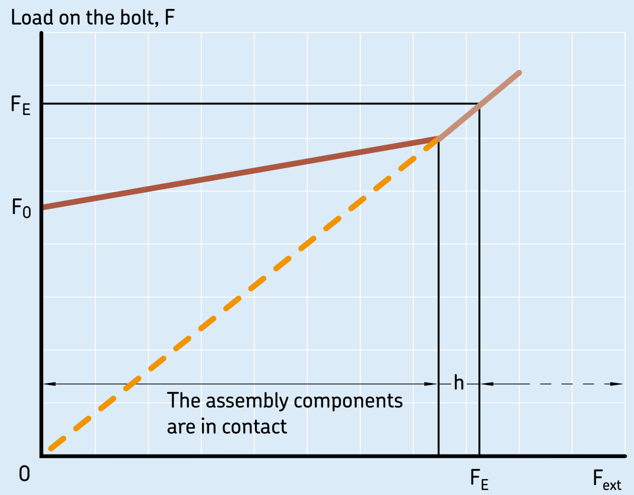
Diagram 10



F_0 = tightening pre-load; F_E = traction load on the bolt
h = the assembly components are no longer in contact
(leading to loss of tightening)

When the tightening load is insufficient relative to the external load
Tightening is lost.

Diagram 11

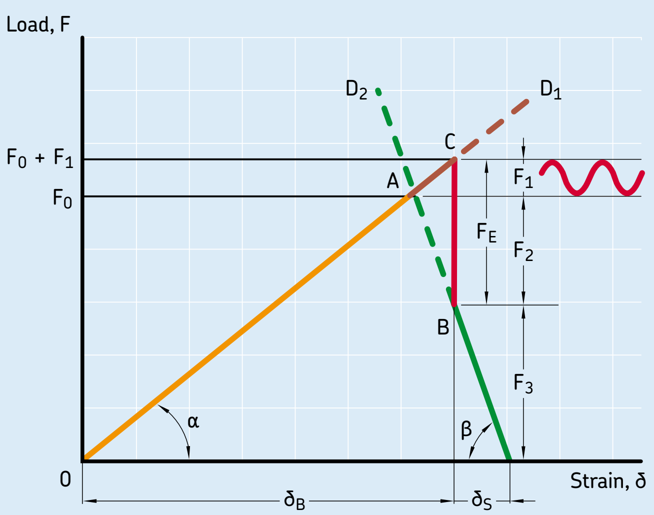


F_0 = tightening pre-load
F_E = traction load on the bolt
h = the assembly components are no longer in contact
(leading to loosening of assembled parts)

Load on the bolt when the external traction load is too high relative to the assembly tightening pre-load

Load diagram in the event of a cyclic external load

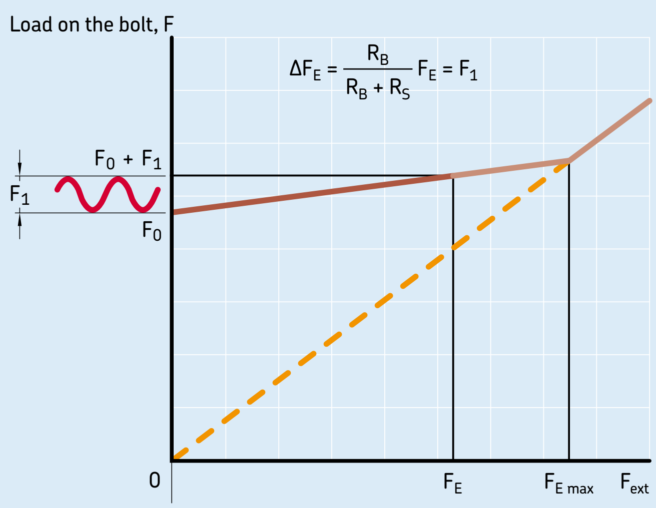
Diagram 12



F_0 = tightening pre-load
$F_0 + F_1$ = traction load on the bolt

Cyclic external load
Variation in the bolt load when a cyclic external traction load F_E is applied to the assembly

Diagram 13



F_0 = tightening pre-load
$F_0 + F_1$ = traction load on the bolt

L/d ratio, F_1/F_0 ratio for hydraulic tightening

It is clear that the greater the difference in stiffness between the bolt and the structure, the smaller the fraction of the external load of the assembly which affects the bolt.

Therefore, it is always better to use bolts with a high length to diameter (L/d) ratio, for stiffness is lowest in this case.

Furthermore, this (L/d) ratio is also the main parameter determining the ratio between the hydraulic load F_h applied during tightening, and the final residual load F_0 (→ **diagram 14**).

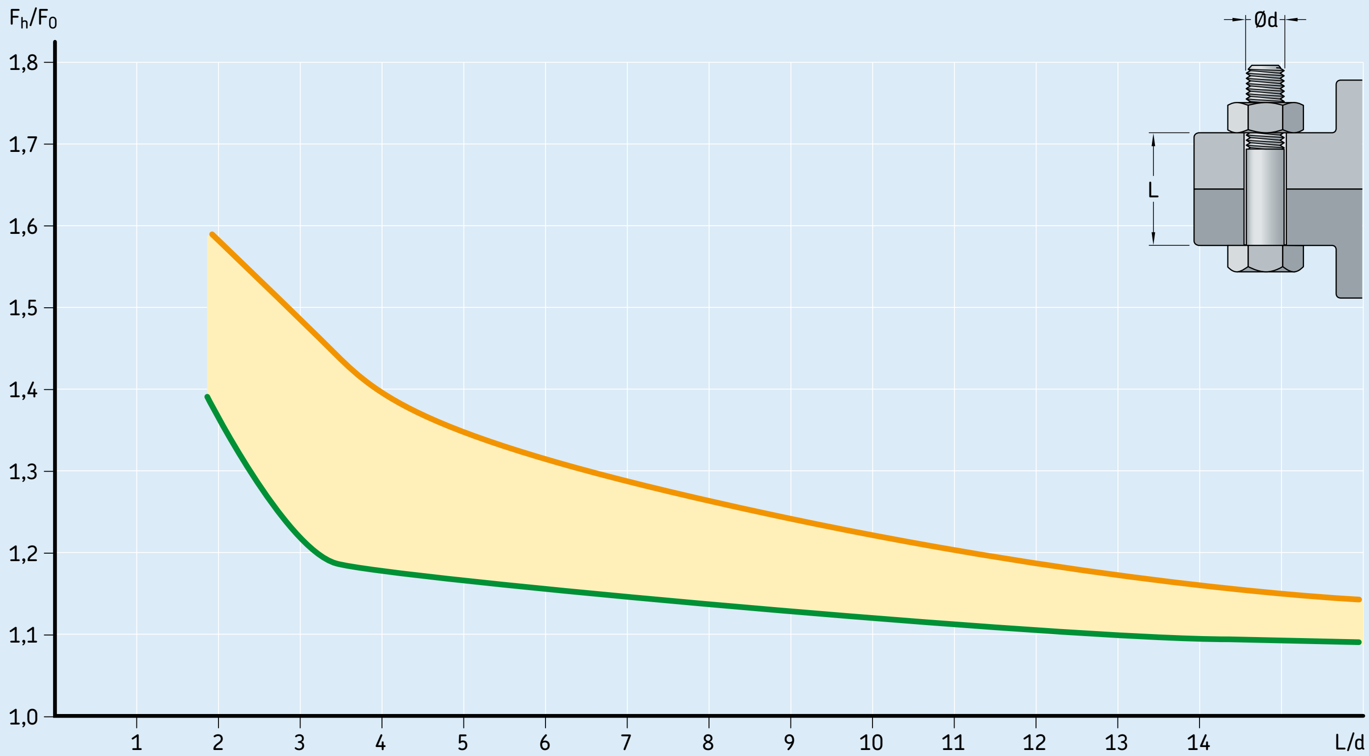
It is clear that the higher the ratio L/d, the lower the relation F_h/F_0. The benefits in a high L/d ratio are therefore twofold.

When the L/d ratio is low (< 1,5), the hydraulic effort required to obtain the final tightening load can be very high, close to the yield point of the material. In these conditions, the use of a top-quality hydraulic torque wrench may be an acceptable solution.

However, in this case, the choice between the two tightening methods will also depend on other parameters of the application, in particular the required accuracy and distribution of tightening, accessibility, the need for simultaneous tightening etc.

D

Diagram 14



F_0 = residual load after tightening
F_h = hydraulic load
Upper limit: wide pitch thread
Lower limit: narrow pitch thread

The F_h/F_0 ratio as a function of bolt aspect ratio L/d for commonly usedbolting steels

Notes

Comparison between torque wrench tightening and hydraulic bolt-tensioning

This comparison will be made by taking two cases: tightening an existing assembly, and design of a new assembly.

Tightening an existing assembly

First consider two flanges with an outer diameter of 600 mm. The bolted joint comprises sixteen M20 × 2,5 bolts on a PCD of 500 mm (→ **fig. 18**).

Each bolt has a clamped length of approximately 200 mm, which means a length/diameter ratio L/d = 10, a frequent choice in mechanical applications.

The pitch of the threads is 2,5, in compliance with the ISO standard, and the class of material is 10-9.

This bolt size has been deliberately chosen for this comparison because these bolts can be fairly easily tightened with a calibrated manual torque wrench (torque < 700 Nm).

Naturally, tightening with a hydraulic tensioner is even easier.

Fig. 21 gives the dimensions of the M20 × 2,5 ISO threads:

d = 20 mm
d_2 = 18,376 mm
d_3 = 16,933 mm
d_{eg} = 17,655 mm
A_S = 244,5 mm^2.

For calculation purposes, the screw is considered as a cylindrical bar with an equivalent diameter d_{eg} and a cross-section A_S.

Class 10-9 material has the following characteristics:

- ultimate tensile strength: $R_m \geq 1\,000$ MPa
- yield point at 0,2%: $R_{e\,(0,2\%)} \geq 900$ MPa.

It is decided not to exceed a stress of 90% of the yieldpoint 0,2% of the material:

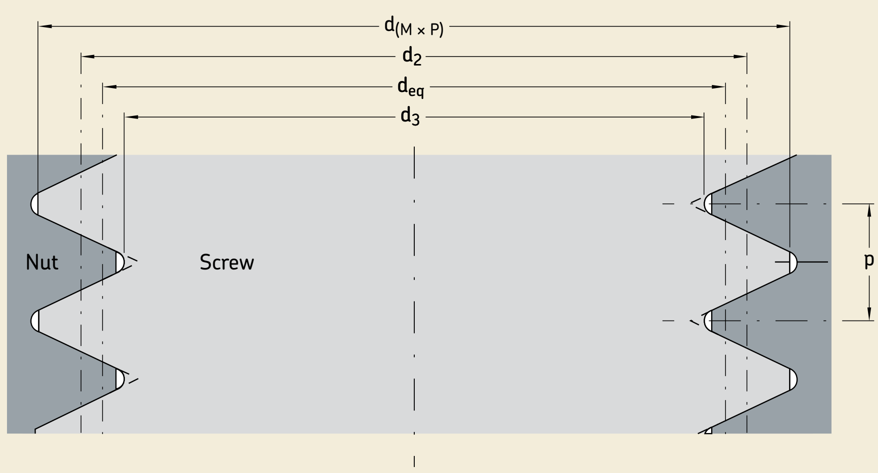
$\sigma_{max} = 0,9\ R_e = 810$ MPa

Therefore the maximum admissible load on the bolt will be:

$F_{max} = 0,9\ R_e\ A_S$
$F_{max} = 198\,000$ N

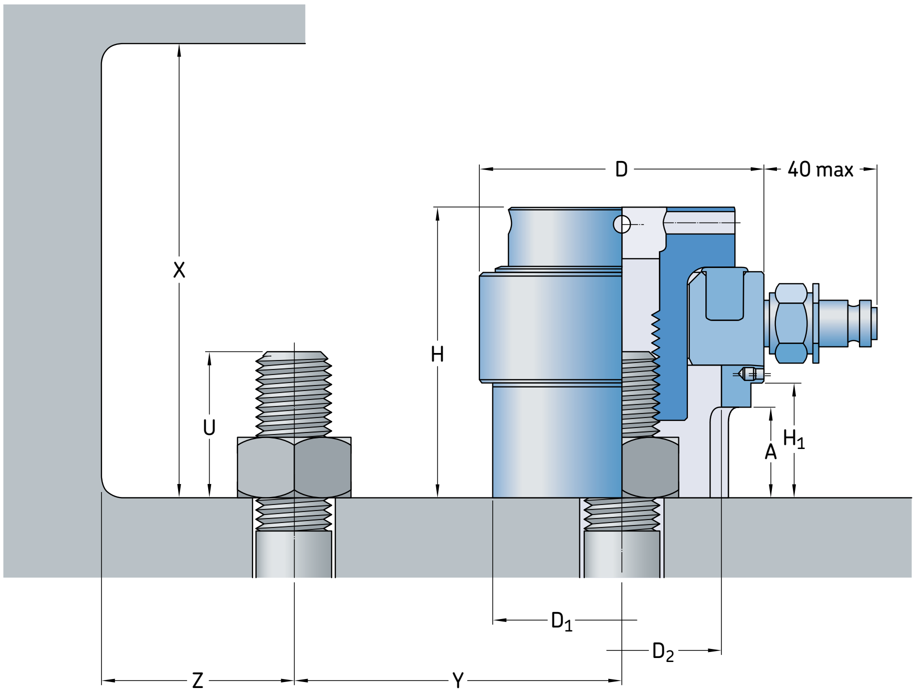
Dimensions of a M20 × 2,5 ISO thread

Fig. 21



d = outer thread diameter
d_2 = average thread diameter = $d - 0,6495\ p$
d_3 = root thread diameter = $d - 1,2268\ p$
d_{eq} = equivalent diameter = $\dfrac{d_2 + d_3}{2}$ = $d - 0,9382\ p$
p = thread pitch
A_S = equivalent section = $\dfrac{\pi d_{eq}^2}{4}$

HTA



Type	Bolt dimension Diameter	Pitch	Max. pressure
	mm	mm	MPa
HTA 20 M20 × 2,5	M20	2,5	150
HTA 20 M22 × 2,5	M22	2,5	150
HTA 20 M24 × 3	M24	3	150
HTA 20 M27 × 3	M27	3	150
HTA 35 M27 × 3	M27	3	150
HTA 35 M30 × 3,5	M30	3,5	150
HTA 35 M33 × 3,5	M33	3,5	150
HTA 35 M36 × 4	M36	4	150
HTA 50 M36 × 4	M36	4	150
HTA 50 M39 × 4	M39	4	150
HTA 50 M42 × 4,5	M42	4,5	150
HTA 50 M45 × 4,5	M45	4,5	150
HTA 60 M42 × 4,5	M42	4,5	150
HTA 60 M45 × 4,5	M45	4,5	150
HTA 60 M48 × 5	M48	5	150
HTA 60 M52 × 5	M52	5	150
HTA 90 M45 × 4,5	M45	4,5	150
HTA 90 M48 × 5	M48	5	150
HTA 90 M52 × 5	M52	5	150
HTA 90 M56 × 5,5	M56	5,5	150
HTA 90 M60 × 5,5	M60	5,5	150
HTA 130 M60 × 5,5	M60	5	150
HTA 130 M64 × 6	M64	6	150
HTA 130 M68 × 6	M68	6	150
HTA 130 M72 × 6	M72	6	150
HTA 130 M76 × 6	M76	6	150
HTA 160 M72 × 6	M72	6	150
HTA 160 M76 × 6	M76	6	150
HTA 160 M80 × 6	M80	6	150
HTA 200 M80 × 6	M80	6	150
HTA 200 M85 × 6	M85	6	150
HTA 200 M90 × 6	M90	6	150
HTA 200 M95 × 6	M95	6	150
HTA 200 M100 × 6	M100	6	150
HTA 250 M100 × 6	M100	6	150
HTA 250 M110 × 6	M110	6	150
HTA 250 M120 × 6	M120	6	150
HTA 250 M125 × 6	M125	6	150
HTA 310 M125 × 6	M125	6	150
HTA 310 M130 × 6	M130	6	150
HTA 310 M140 × 6	M140	6	150
HTA 310 M150 × 6	M150	6	150

HTA .. T ..

HTA .. C

HTA .. J

optional

Multi-purpose

The dimensions and the traction load suit this tensioner to many different applications.

Reference example:

- For a complete tensioner for M48 bolts: HTA 90 M48 × 5
- For the M48 × 5 brace for the HTA 90 tensioner: HTA 90T M48 × 5

This page is taken from the Pilgrim catalogue "HYDROCAM Bolt Tensioners – Industrial Tightening Systems", which describes the entire range of HYDROCAM tensioners and their industrial applications.

Hydraulic area	Max. hydraulic load	Piston stroke	Dimensions										Total tensioner weight
			D	H	H_1	D_1	D_2	A	U	X	Y	Z	
cm^2	kN	mm	mm										kg
20	300	8	86	100	30	74	56	26	38	138	56	44,5	3
20	300	8	86	100	30	74	56	26	42	142	57	44,5	3
20	300	8	86	100	30	74	56	26	46	146	59	44,5	3
20	300	8	86	100	30	74	56	26	52	152	62	44,5	3
35	525	8	109	116	40	97	73	31	52	168	73,5	56	4,8
35	525	8	109	116	40	97	73	31	57	173	76,5	56	4,8
35	525	8	109	116	40	97	73	31	63	179	79	56	4,8
35	525	8	109	116	40	97	73	31	69	185	81	56	4,8
50	750	8	128	128	49	116	90	38	69	197	91	65,5	7,5
50	750	8	128	128	49	116	90	38	74	202	94	65,5	7,5
50	750	8	128	128	49	116	90	38	80	208	97	65,5	7,5
50	750	8	128	128	49	116	90	38	86	214	100	65,5	7,5
60	900	8	137	140	54	133	102	40	80	220	91	69,5	9
60	900	8	137	140	54	133	102	40	86	226	92	69,5	9
60	900	8	137	140	54	133	102	40	92	232	94	69,5	9
60	900	8	137	140	54	133	102	40	99	239	96	69,5	9
90	1 350	8	166	154	65	154	114	42	86	240	119	84,5	15,3
90	1 350	8	166	154	65	154	114	42	92	246	122	84,5	15,3
90	1 350	8	166	154	65	154	114	42	99	253	124,5	84,5	15,3
90	1 350	8	166	154	65	154	114	42	107	261	127,5	84,5	15,3
90	1 350	8	166	154	65	154	114	42	114	268	130,5	84,5	25
130	1 950	8	198	179	82	187	137	50	114	293	147	101	25
130	1 950	8	198	179	82	187	137	50	122	301	150	101	25
130	1 950	8	198	179	82	187	137	50	130	309	153	101	25
130	1 950	8	198	179	82	187	137	50	137	316	155,5	101	25
130	1 950	8	198	179	82	187	137	50	145	324	158,5	101	31
160	2 400	10	215	190	86	203	145	50	137	327	163,5	109	31
160	2 400	10	215	190	86	203	145	50	145	335	166,5	109	31
160	2 400	10	215	190	86	203	145	50	152	342	169,5	109	31
200	3 000	10	244	217	106	232	180	60	152	355	184,5	124	31
200	3 000	10	244	217	106	232	180	60	162	365	187,5	124	39
200	3 000	10	244	217	106	232	180	60	171	374	193	124	39
200	3 000	10	244	217	106	232	180	60	181	384	196	124	39
200	3 000	10	244	217	106	232	180	60	190	393	202	124	54
250	3 750	10	284	245	131	272	223	73	190	425	222	144	54
250	3 750	10	284	245	131	272	223	73	209	444	227,5	144	54
250	3 750	10	284	245	131	272	223	73	228	463	236	144	54
250	3 750	10	284	245	131	272	223	73	238	473	242	144	54
310	4 650	10	325	273	156	313	260	86	238	506	262,5	164,5	75
310	4 650	10	325	273	156	313	260	86	247	515	265	164,5	75
310	4 650	10	325	273	156	313	260	86	266	534	274	164,5	75
310	4 650	10	325	273	156	313	260	86	285	553	280	164,5	75

Tightening with a hydraulic tensioner

Lets take a look at Hydraulic Tensioning.

The HTA 20 tensioner, which suits the characteristics of the bolts being used, is selected from the Pilgrim HYDROCAM standard product range (→ **table 3**).

Uncertainty factors in hydraulic tensioning

The various factors of uncertainty generally encountered in hydraulic bolt tensioning are listed and analysed below.

Deviation due to dimensions and tolerances of bolts and parts of assembly

Diagram 14 helps determine the hydraulic tension load that must be applied to the bolt in order to obtain the desired final tightening load.

It is seen that for a ratio L/d = 10, the F_h/F_0 ratio between the necessary hydraulic load F_h and final residual tightening load F_0 is approximately 1,10 and 1,20, viz: 1,15 ±4,5%.

However, the graph shows a general deviation (for L/d = 10 : ±4,5%) which takes into account the various types of assembly, and the various shapes and characteristics of components, threads in particular, most often met in mechanics.

However, we know from experience that for a given assembly, the F_h/F_0 ratio will vary by ±2% or less for single bolt-tightening, since the dimensional tolerances, geometric faults and deviations in material characteristics need to be considered solely for a given part or a given batch.

Therefore, we take 1,15 as the average value and 1,18/1,12 as maximum and minimum respectively.

Uncertainty in the hydraulic tension load

Uncertainty as to the hydraulic tension load first depends on the tensioner itself and on the accuracy on the hydraulic pressure measurement.

Pilgrim HYDROCAM tensioners have excellent efficiency: 98% ±1%.

The uncertainty attributable to the tensioner is thus only ±1%.

The deviation on the pressure value is generally ±2%. It depends on the accuracy of the measurement instrument (manometer or sensor) and on the accuracy of the operator's reading.

In the final analysis, the accuracy of the hydraulic tension load comes to ±3%.

We retain our preliminary condition of not exceeding a tension load on the bolt of F_{max} = 198 000 N (to avoid exceeding a maximum stress of σ_{max} = 810 MPa).

The hydraulic tension load will therefore be limited to the same value:

$F_{h\ max}$ = 198 000 N

Taking the "hydraulic" uncertainty into consideration, the minimum hydraulic tension load comes to:

$F_{h\ min}$ = 198 000/1,06

so: $F_{h\ min}$ = 186 800 N

And the required average hydraulic load is:

$F_{h\ m} = (F_{h\ max} + F_{h\ min})/2$ = 192 400 N

Since the Pilgrim HYDROCAM HTA 20 tensioner has a hydraulic area of 20 cm^2, and allowing for the efficiency (≈98%), the nominal pressure will be: 98 MPa.

Uncertainty due to the clamping moment of the nut

When the nut is turned down manually there is uncertainty as to the clamping moment, which generally leads to a ±3% uncertainty as to the final tightening load.

The General Catalogue Pilgrim HYDROCAM describes simple precautions to limit turn-down deviation.

In our example, the deviation factor is ±3% on the turning down.

Consequences of uncertainties on the final load

The total uncertainty on the final tightening load can now be calculated.

The final tightening load is maximal when the hydraulic tension load is maximal, the clamping moment to turn down the nut is maximal, and ratio F_h/F_0 is minimal.

The maximum final tightening load will therefore be:

$F_{0\ max}$ = 182 000 N (198 000 × 1,03/1,12)

The final tightening load is minimal when hydraulic tension is minimal, the clamping moment to turn down the nut is minimal and ratio F_h/F_0 is maximal.

The minimum final tightening load will therefore be:

$F_{0\ min}$ = 154 000 N (186 800 × 0,97/1,18)

The average value of the tightening load is calculated as (**diagram 15**):

$F_{0\ m} = (F_{0\ max} + F_{0\ min})/2$ = 168 000 N

F_0 = 168 000 N ±8,5%

It is seen that we are within the tolerance previously indicated for "usual" use of a tensioner without any additional measurement means.

With time and practice, it is perfectly feasible to do much better on the same assembly: ±6% and even better.

Maximum admissible external load on the assembly

Now that our assembly is correctly fastened with the hydraulic tensioner, let us see what maximum external load can be applied on each bolted point without exceeding the set limit of:

F_{max} = 198 000 N

For this purpose, the stiffnesses of the bolts and of the assembly are required.

The stiffness of the bolts can be calculated from the known dimensions:

$R_B = A_S\ E/L$ = 244,5 × 210 000/200
= 256 700 N/mm

We have not described a precise design for the structure, therefore we refer to the general experience of mechanical engineers and arbitrarily set the stiffness of the structure at 5 to 10 times that of the bolt.

Assuming a value of 8 for this coefficient, this gives:

R_S = 2 000 000 N/mm

It was previously shown that, for an external load F_E, only one part, F_1 is applied to the bolt:

$F_1 = F_E\, R_B/(R_B + R_S)$

It is therefore easy to calculate the maximum admissible external load which can be supported by the bolts without exceeding the set limit of 810 MPa.

Thus, the total load on each bolt – even includingthe external load – must not exceed:

F_{max} = 198 000 N

The value of this maximum load is:

$F_{T\,max} = F_{0\,max} + F_1 = F_{0\,max} + F_E\, R_B/(R_B + R_S)$

Thus the maximum admissible F_E can be calculated (→ **diagram 15**) :

$F_E = (F_{t\,max} - F_{0\,max})\,(R_B + R_S)/R_B$

F_E = 140 000 N

The total external load which can be supported by the assembly described on **page 18** is:

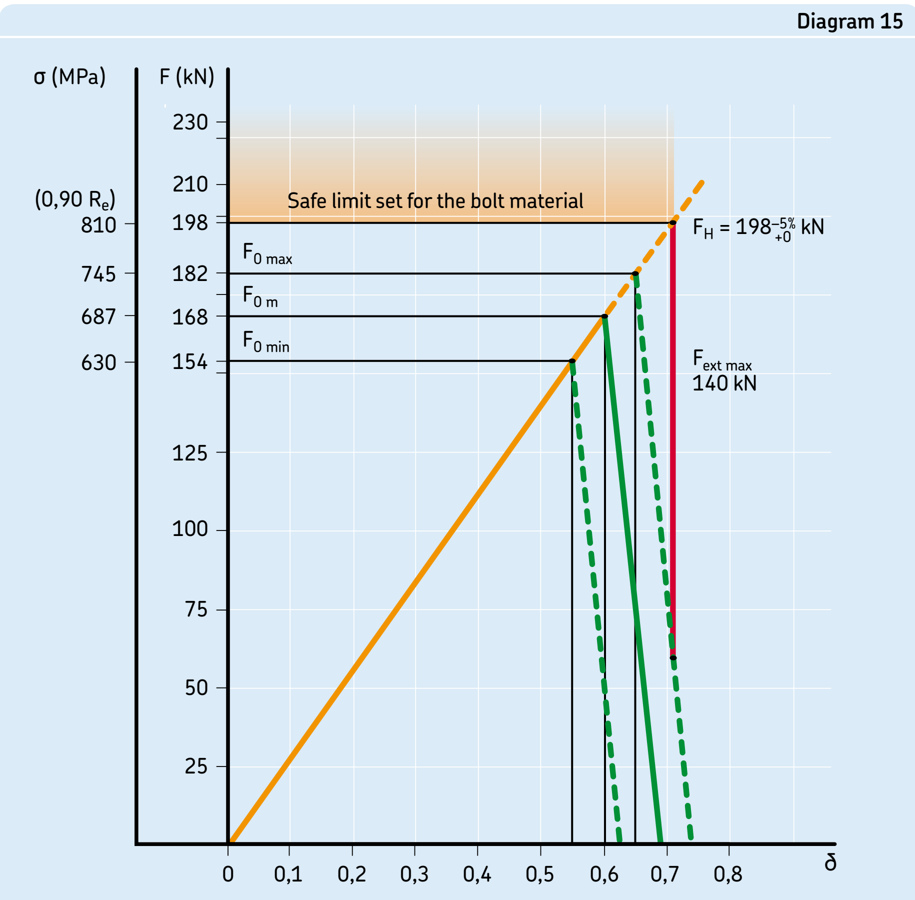
16 × 140 000 = 2 240 000 N

The distribution of the load on the bolts is assumed to be uniform.

We know that bolts will all “work” in almost the same way with very similar stress levels. There is great homogeneity.

However, for higher levels of external load, there is the risk of exceeding the safety limit set for the bolts.

Load/elongation diagram of a M20 × 2,5 lg 200 mm ISO bolt tightened with a bolt tensioner



Tightening with a torque wrench

Let us now tighten the same set of bolts with a torque wrench.

It is decided to use a manual calibrated torque wrench.

Uncertainty factors with torque wrench tightening

Let us examine these factors and the values they can reach.

Uncertainty as to the torque itself

A calibrated torque wrench generally has a deviation of: ±5%.

However, it should be pointed out that with the commonly used torque tightening equipment, this deviation is often much higher than ±5%.

Uncertainty due to bolt and assembly tolerances

These deviations are primarily due to dimensional tolerances, geometrical defects or variations in material characteristics.

Since there is no F_h/F_0, ratio,the influence of this deviationis lower than for the hydraulic tensioner. This value is estimated at: ±1%.

Uncertainty as to friction coefficients

Two friction coefficients need to be taken into account in the case of the torque wrench:

μ_{th} = friction coefficient in the nut/bolt threads

μ_{fl} = friction coefficient of the nut face on the structure.

For steel parts originating from a given batch,the first value generally lies between 0,08 and 0,12 (0,10 ±20%) and the second is 0,10 to 0,15 (0,125 ±20%).

E

Analysing the components of the tightening torque

The tightening torque T_T delivered by the wrench needs to overcome two load moments:

- load moment in the T_{th} bolt/nut threads (this is the torquewhich generates bolt torsion) (→ **fig. 5**)
- load moment in the nut/structure bearing surfaces T_{fl}.

Therefore, we can calculate: $T_T = T_{th} + T_{fl}$

Equations commonly used for these torques are:

$T_{th} = T'_{th} + T''_{th} = F\,(p/2\pi) + F\,(\mu_1\, r_r)$ and
$T_{fl} = F\,(\mu_2\, r_m)$

where:
p = thread pitch (2,5 for M20)
$1/2\pi = 0{,}16$
μ_{th} = friction coefficient in threads
μ_{fl} = friction coefficient in bearing surfaces
d_2 = average diameter of threads (18,376 for M20)
r_r = $0{,}583\, d_2$ = average radius of threads
r_m = average radius of bearing surface of the nut (13 for nut M20)
F = tightening load.

This can be expressed as:

$T_{th} = T'_{th} + T''_{th} = F\,(0{,}16\, p) + F\,(\mu_{th}\, 0{,}583\, d_2)$
$T_{fl} = F\,(\mu_{fl}\, r_m)$

In fact only the part T'_{th} of torque T_T is used to elongate the bolt: the other torque components could be called "parasite" torques.

The aim is to obtain the same final average tightening load as with the hydraulic bolt tensioner:

$F_{0\,m} = 168\,000$ N

Torque to be applied

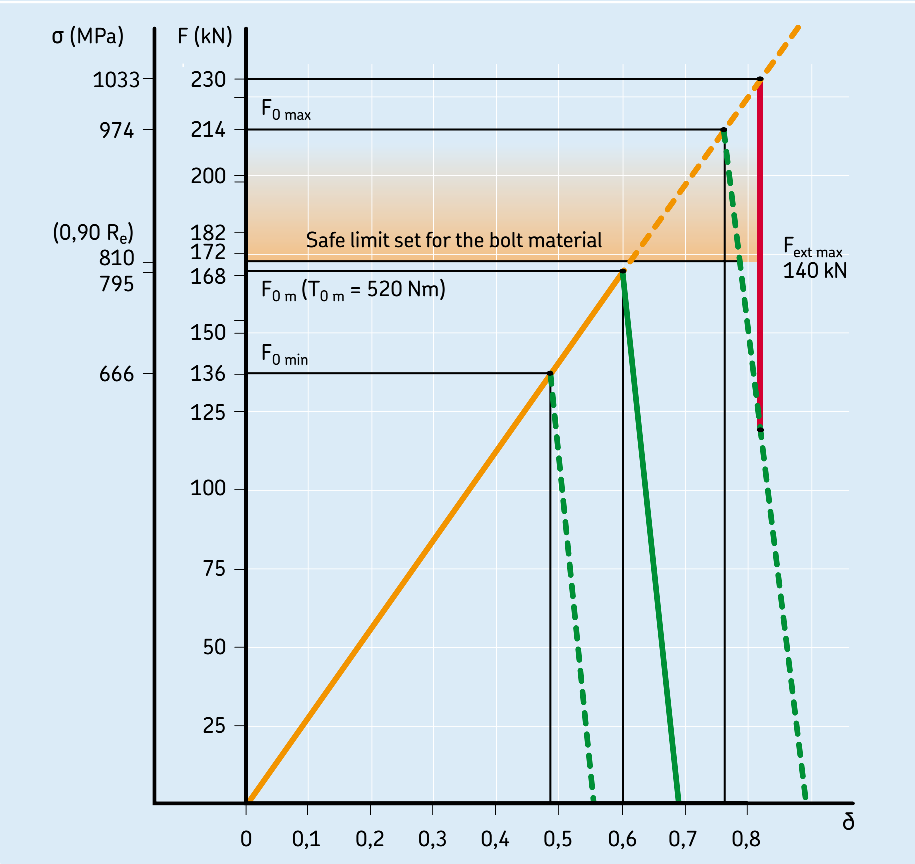
Let us calculate the necessary average torque to obtain this required average load.

The load moment in the threads is:

$T_{th\,m} = 168\,000\,(0{,}16 \times 2{,}5)$
$\quad + 168\,000\,(0{,}10 \times 0{,}583 \times 18{,}376)$
$T_{th\,m} = 67\,200 + 180\,000 = 247\,200$ Nmm

Load/elongation diagram of a M20 × 2,5 lg 200 mm ISO bolt tightened using a torque wrench

Diagram 16



The load moment on the bearing surfaces is:

$T_{fl\,m} = 168\,000\,(0{,}125 \times 13)$
$T_{fl\,m} = 273\,000$ Nmm

The required average torque will be:

$T_{T\,m} = T_{th\,m} + T_{fl\,m} = 520\,200$ Nmm
520,2 Nm
52,02 daNm

Only 67 200 Nm (just 13%) of this is actually used for tightening !

Given the accuracy of the tool itself, the real torque will be within the following two values:

$T_{T\,min} = 520\,200 \times 0{,}95 = 494\,190$ Nmm
$T_{T\,max} = 520\,200 \times 1{,}05 = 546\,210$ Nmm

Consequence of uncertainty as to the final load

The corresponding minimum and maximum tension loadsin the bolt will be:

$F_{min} = T_{T\,min} / (0{,}16 \times 2{,}5 + \mu_{th\,max}\, 0{,}583\, d_2 + \mu_{fl\,max}\, 13)$
$F_{min} = 494\,190 / (0{,}16 \times 2{,}5 + 0{,}12 \times 0{,}583 \times 18{,}376 + 0{,}15 \times 13)$
$F_{min} = 135\,930$ N

$F_{max} = T_{T\,max} / (0{,}16 \times 2{,}5 + \mu_{th\,min}\, 0{,}583\, d_2 + \mu_{fl\,min}\, 13)$
$F_{max} = 546\,210 / (0{,}16 \times 2{,}5 + 0{,}08 \times 0{,}583 \times 18{,}376 + 0{,}10 \times 13)$
$F_{max} = 213\,610$ N

It is already clear that the real average load will be over the targeted average load:

$F_{0\,mr} = (135\,930 + 213\,610)/2 = 174\,770$ N
Which is: +4%

Expressing the tolerance as a function of the expected average tightening load, the tension load in the bolt can be expressed as follows (→ **diagram 16**):

$F_0 = 168\,000$ N (+27% to −19%)
(or 174 770 N ±22%)

The stress analysis is detailed in the following pages, but it can already be seen that, because of the additional torsion stress, to induce a tensioning load of 168 kN in the bolt tightened by torque wrench method leads to an equivalent stress of 795 MPa and therefore the safe limit of 810 MPa leaves only a maximum tension load (including the external load F_E) of 172 kN.

It is also seen that even with an accurate tightening tool, the tolerance range on the final tightening tension load is, in our example, almost three times greater with a torque wrench than with a hydraulic bolt-tensioner.

Effect on stresses in the bolt

Case where stresses are maximum

Let us now see what happens in the bolt in the case of “stacking” of tolerances, that is, when the actual values of the various parameters simultaneously contribute to maximising the resulting loads.

The loads induce the following stresses in the bolt: – traction stress:

- traction stress:
  $\sigma_{max} = F_{max}/A_S = 213\ 610/244{,}5$
  $\sigma_{max} = 873{,}6$ MPa
- torsion stress:
  $\tau_{max} = 16\ T_{th\ max}/(\pi\ d_{eq}^3)$
  we know that
  $T_{th\ max} = 213\ 610\ (0{,}16 \times 2{,}5 + 0{,}08 \times 0{,}583 \times 18{,}376)$
  $T_{th\ max} = 268\ 520$ Nmm
  therefore $\tau_{max} = 248{,}5$ MPa
- the equivalent Von Mises stress is calculated as:
  $\sigma_{eq\ max} = \sqrt{\sigma_{max}^2 + 3\ \tau_{max}^2}$
  $\sigma_{eq\ max} = 974$ MPa

It is seen that in this case, where loads are maximal, the torsion stress – which attains 28 percent of the traction stress – leads to an increase in the equivalent stress of more than 11%.

In this case, the yield point of the material will probably be exceeded as of the tightening operation itself, let alone the 810 MPa limit which we had set ourselves.

External load on the assembly when stresses are maximum

Now let us assume that an external load of 140 000 N is applied to one bolt in the assembly after torque-tightening has resulted in maximum loads.

We have already seen that the load taken up by the bolt is:

$F_1 = F_E\ R_B/(R_B + R_S)$
$F_1 = 15\ 900$ N

The maximum tension load on the bolt increases to a level of:

$F_{max} = 213\ 610 + 15\ 900 = 229\ 510$ N

- The traction stress becomes:
  $\sigma_{max} = 939$ MPa
- The torsion stress does not change:
  $\tau_{max} = 248{,}5$ MPa
- The equivalent Von Mises stress comes to:
  $\sigma_{eq\ max} = 1\ 033$ MPa

We can clearly see that the external effort makes things worse.

In our example, there is a very high risk that bolt stricture will occur during the tightening operation, and that the bolt will break under working conditions!

Case where stresses are minimum

We can also examine what happens in the assembly when the actual values of the various parameters all simultaneously contribute to minimising the resulting loads.

These minimum loads generate the following stresses in the bolt:

- traction stress:
  $\sigma_{min} = F_{min}/A_S = 135\ 930/244{,}5$
  $\sigma_{min} = 556$ MPa
- torsion stress:
  $\tau_{min} = 16 \times T_{th\ min}/(\pi \times d_{eq}^3)$
  we know that:
  $T_{th\ min} = 135\ 930\ (0{,}16 \times 2{,}5 + 0{,}12 \times 0{,}583 \times 18{,}376)$
  $T_{th\ min} = 229\ 120$ Nmm
  so $\tau_{min} = 212$ MPa
- and the equivalent Von Mises stress will be:
  $\sigma_{eq\ min} = \sqrt{\sigma_{min}^2 + 3\ \tau_{min}^2}$
  $\sigma_{eq\ min} = 666$ MPa

It can be seen that in “negative stacking” conditions, where the resulting loads are minimal, the torsion stress attains 38% of the traction stress and leads to an increase of almost 20% in the equivalent stress!

Fortunately, in this case, the loads are such that stresses are far below the yield point of the material.

However, it will be noticed that this “minimum” stress obtained by the torque method is very close to the pure traction stress foreseeable in average tightening conditions (666 MPa versus 687 MPa).

This serves to confirm that when a torque wrench is used, the torsion induced by this tightening method must be carefully taken into account.

External load on the assembly when stresses are minimal

Let us now apply the external load of 140 000 N to the torque wrench tightened assembly under (chance) minimal load conditions.

It is clear that from the stress viewpoint, the consequences are not very severe.

We know that the additional load on the bolt is:

$F_1 = 15\ 900$ N

The tension load is expressed by:

$F = 135\ 930 + 15\ 900 = 151\ 830$ N

- the traction stress is:
  $\sigma = 621$ MPa
- the torsion stress is:
  $\tau = 212$ MPa
- the equivalent Von Mises stress will be:
  $\sigma_{eg} = 721$ MPa

We approach, but do not reach the yield point of the material.

In fact in this case, the problem lies not in terms of the stresses, but more with the tightening itself as will be shown.

In our example, the external load on the bolt is:

$F_E = 140\ 000$ N

For “minimal” tightening conditions, the resulting tightening load is 135 930 N, whereas the external load which leads to looseness is:

$$F_E = F_0 \, R_S / (R_B + R_S)$$

which is: 153 345 N !

In the case of a slightly higher external load (+10%), there is a real risk of looseness, unscrewing and leakage (if it is a sealing application).

Furthermore, we also know that, under cyclic loads, fatigue failure is more likely to occur when the tightening is not optimal.

Risks when tightening with torque wrench

With torque tightening, the two following situations can both occur on a given application:

- excessive stress, resulting in bolt failure due to over-stress,
- insufficient tightening, with the risk of looseness, potential leakage and even fatigue failure.

To avoid these risks, the tendency is often to oversize the bolts.

Design engineers often think that higher tightening loads can thus be applied.

However, this approach both worsens the situation in terms of fatigue resistance, and increases the weight of the assembly, which in turn increases construction cost (more, bigger or heavier bolts, larger flanges, etc). It can also impact service and maintenance costs, which have to allow for heavier and larger parts.

Conclusion of the comparison

The comparison between the torquing and tensioning methods was based on tightening of a single isolated bolt. The benefits of using the tensioner are immediately clear.

As described in the introduction, hydraulic tensioners are even more advantageous when several bolts need to be tightened, which is, of course, the situation which is most often encountered.

In this case, ease of use, safety, reliability, repeatability, and the possibility of performing simultaneous tightening, not to mention better accuracy are just some of the advantages.

In addition, accuracy can be further improved by using a measurement system such as the previously described sensor washer.

Design of a new assembly

Let us now examine the case involving the design of a new assembly.

Based on the preceding work, it is clear that design will differ greatly depending on whether the bolt-tightening method involves torquing or the use of hydraulic tensioners.

Let’s look again at the example of two flanges with an outer diameter of 600 mm, assembled using sixteen M20 × 2,5 bolts on a PCD of 500 mm (→ **fig. 18**).

Tightening with hydraulic tensioners

In the previous example, we optimised the size of the bolts and the value of the loads was optimised for hydraulic tensioning.

Nothing needs to be added in this case.

Tightening with the torque method

Let us now examine the design considerations to allow use of the torque-tightening method while ensuring the same safety margin as with the bolt tensioner.

Three solutions exist at the initial design stage:

- selecting a higher class of bolt material
- using a greater number of bolts
- increasing the size of the bolts.

Let us analyse each of these three solutions in turn.

Selecting a higher class of bolt material

Given that the maximum stress is $\sigma_{eq\ max}$ = 1 033 MPa, we conclude that the class of material for the bolt must be 12-9. In this case, the ultimate tensile stress is R_m > 1 200 MPa and the yield point is $R_{e\ (0,2\%)}$ > 1 080 MPa.

Yet, even with this solution, the safety margin will not be the same as with the bolt tensioner because 90% of the yield point means a maximum stress of 972 MPa.

Therefore, in this example, changing the class of the material is a solution only if we accept a lower safety margin.

Furthermore, the additional cost of using better material may exceed 30%, and in the final analysis, the previously described problems related to the risk of loosening in the minimal case, still remain.

Finally the problems mentioned above, related to loosening in the minimal case.

Inversely, however, an assembly designed with 16 M20 × 2,5 class 12-9 bolts, tightened by the torque method, can be re-designed to use 16 M20 × 2,5 class 10-9 bolts, tightened by the hydraulic tensioning method, and the safety margin of the assembly will ever be greater.

Increasing the number of bolts

This solution can only be retained if there is sufficient space available around the full periphery (PCD) of the assembly.

The space between two successive bolts must be such that the socket of the torque wrench can be placed without interference with the adjacent bolts.

In our example, the overall diameter of the nut is approximately 36 mm and the outer diameter of the socket is 50 mm.

This means that the straight distance between two successive bolts must be greater than 43 mm.

We assume a value of 50 mm to allow sufficient clearance. With sixteen bolts over a diameter of 500 mm, the distance between two bolts equates to 97,55 mm, which is quite sufficient.

It would even be possible to use 30 bolts, since the distance between adjacent bolts would still be 52,26 mm.

For 16 bolts the maximum stress is: $\sigma_{eq\ max}$ = 1 033 MPa, whereas we intended not to exceed 810 MPa ; therefore, we need to reduce the stress by more than 20%.

If we simply apply this coefficient to the number of bolts, we find that 20 bolts are needed instead of sixteen.

Let us examine the situation of 20 bolts tightened with a torque wrench.

As we have already seen, the average tightening load is:

$$168\,000 \times 16 = 2\,688\,000 \text{ N}$$

Therefore with 20 bolts:

$F_{0\ m}$ = 134 400 N per bolt.

Now, with the total external load distributed over 20 bolts, the load per bolt equates to:

$$2\,240\,000/20 = 11\,200 \text{ N}$$

and, if we take the stiffness into account, the additional load on each bolt is:

F_1 = 12 720 N

The calculation which we previously made for 16 bolts is repeated for the case using 20 bolts.

Average torque required to reach the average tightening load

Average load moment in the threads:

$$T_{th\ m} = 134\,400\,(0{,}16 \times 2{,}5) + 134\,400\,(0{,}10 \times 0{,}583 \times 18{,}376)$$
$$T_{th\ m} = 53\,760 + 144\,000 = 218\,400 \text{ Nmm}$$

Average load moment on bearing surfaces of the nuts:

$$T_{fl\ m} = 134\,400\,(0{,}125 \times 13)$$
$$T_{fl\ m} = 218\,400 \text{ Nmm}$$

Average torque required:

$$T_{T\ m} = T_{th\ m} + T_{fl\ m} = 416\,160 \text{ Nmm} = 416{,}16 \text{ Nm} = 41{,}616 \text{ daNm}$$

Allowing for the accuracy of the tool, the actual torque will lie somewhere between:

$$T_{T\ min} = 416\,160 \times 0{,}95 = 395\,350 \text{ Nmm}$$
$$T_{T\ max} = 416\,160 \times 1{,}05 = 436\,970 \text{ Nmm}$$

Corresponding minimum and maximum tension loads in the bolt

$$F_{min} = T_{T\ min} \big/ (0{,}16 \times 2{,}5 + \mu_{th\ max}\, 0{,}583\, d_2 + \mu_{fl\ max}\, 13)$$
$$F_{min} = 395\,350 \big/ (0{,}16 \times 2{,}5 + 0{,}12 \times 0{,}583 \times 18{,}376 + 0{,}15 \times 13)$$
$$F_{min} = 108\,740 \text{ N}$$

$$F_{max} = T_{T\ max} \big/ (0{,}16 \times 2{,}5 + \mu_{th\ min}\, 0{,}583\, d_2 + \mu_{fl\ min}\, 13)$$
$$F_{max} = 436\,970 \big/ (0{,}16 \times 2{,}5 + 0{,}08 \times 0{,}583 \times 18{,}376 + 0{,}10 \times 13)$$
$$F_{max} = 170\,890 \text{ N}$$

Actual average tension load:

$$F_{0\ mr} = (108\,740 + 170\,890) \big/ 2 = 139\,815 \text{ N}$$

(4% more than the expected value)

Uncertainty in the tension load in the bolt:

F_0 = 134 400 N (+27% at –19%)

Stresses in the bolt in the case of maximum loads

- traction stress:
  $\sigma_{max} = F_{max}/A_S = 170\,890/244{,}5$
  $\sigma_{max} = 699$ MPa
- torsion stress:
  $\tau_{max} = 16\, T_{th\ max} \big/ (\pi\, d^3_{eq})$
  $T_{th\ max} = 170\,890\,(0{,}16 \times 2{,}5 + 0{,}08 \times 0{,}583 \times 18{,}376) = 135\,944$ Nmm
  $\tau_{max} = 126$ MPa
- equivalent stress (Von Mises criterion):
  $\sigma_{eq\ max} = \sqrt{\sigma_{max}^2 + 3\,\tau_{max}^2}$
  $\sigma_{eq\ max} = 732$ MPa

E

Applying the external effort

As we have previously seen, the fraction of the external load which is applied in every bolt is:

$F_1 = 12\,720$ N

Maximum tensioning load in the bolt:

$F_{max} = 170\,890 + 12\,720 = 183\,610$ N

Stress when external load is applied

- traction stress:
  $\sigma_{max} = 751$ MPa
- torsion stress:
  $\tau_{max} = 126$ MPa
- equivalent stress (Von Mises):
  $\sigma_{eq\,max} = 782$ MPa

The objective is achieved. Therefore, using 20 bolts instead of 16 can be a solution.

However, the cost of the bolts increases by 25% to which must be added the extra machining costs. Furthermore, there is still a risk that problems will arise due to insufficient tightening.

(It should to be pointed out that a calculation with 18 bolts yields a maximum stress of: $\sigma_{eq\,max} = 862$ MPa which does not meet our safety requirement.)

Inversely, however, an assembly designed with 20 M20 × 2,5 bolts, tightened by the torque method, can be re-designed to use 16 M20 × 2,5 bolts tightened by the hydraulic tensioning method.

Increasing the bolt diameter

As with the increase in the number of bolts, we first simply apply the 20% coefficient to the square of the bolt diameter. Given the standard dimensions, we find that a 24 mm bolt is needed instead of a 20 mm one.

The dimensions of M24 × 3 ISO bolts are:

$d = 24$ mm
$d_2 = 22{,}0508$ mm
$d_3 = 20{,}320$ mm
$d_{eq} = 21{,}1854$ mm
$A_S = 352{,}5$ mm^2.

We still need an average load on each bolt of:

$F_{0\,m} = 168\,000$ N per bolt.

Let us repeat the torque calculation for M24 bolts.

Average torque required to obtain the average load

Load moment in the threads:

$T_{th\,m} = 168\,000\,(0{,}16 \times 3) + 168\,000\,(0{,}10 \times 0{,}583 \times 22{,}0508)$
$T_{th\,m} = 80\,640 + 215\,970 = 296\,610$ Nmm

Load moment on the bearing surfaces:

$T_{fl\,m} = 168\,000\,(0{,}125 \times 16)$
$T_{fl\,m} = 336\,000$ Nmm

Average torque required:

$T_{T\,m} = T_{th\,m} + T_{fl\,m} = 632\,610$ Nmm (632,61 Nm or 63,261 daNm)

Because of the uncertainty induced by the tool, the real torque value will randomly lie between the following:

$T_{T\,min} = 632\,610 \times 0{,}95 = 600\,980$ Nmm
$T_{T\,max} = 632\,610 \times 1{,}05 = 664\,240$ Nmm

Corresponding minimum and maximum tension loadsin the bolt

$F_{min} = 600\,980 \big/ (0{,}16 \times 3 + 0{,}12 \times 0{,}583 \times 22{,}0508 + 0{,}15 \times 16)$
$F_{min} = 135\,890$ N

$F_{max} = 664\,240 \big/ (0{,}16 \times 3 + 0{,}08 \times 0{,}583 \times 22{,}0508 + 0{,}10 \times 16)$
$F_{max} = 213\,690$ N

Stresses in the bolt in the maximum case

- traction stress:
  $\sigma_{max} = F_{max}/A_S = 213\,690/352{,}5$
  $\sigma_{max} = 606{,}5$ MPa
- torsion stress:
  $\tau_{max} = 16\,T_{th\,max} \big/ (\pi\, d_{eq}^3)$
  ($T_{th\,max} = 213\,690\,(0{,}16 \times 3 + 0{,}08 \times 0{,}583 \times 22{,}0508) = 322\,340$ Nmm)
  $\tau_{max} = 173$ MPa
- equivalent stress (Von Mises criterion):
  $\sigma_{eq\,max} = \sqrt{\sigma_{max}^2 + 3\,\tau_{max}^2}$
  $\sigma_{eq\,max} = 676$ MPa

Application of the external load

We know that distributed load on each bolt is:

$F_E = 140\,000$ N

However, given the larger size of the bolt, there is higher stiffness.

Therefore the fraction of external load supported by the bolt will be greater. It actually is:

$F_1 = 21\,860$ N (instead of 15 900 N)

The maximum tension load on the bolt will be:

$F_{max} = 213\,690 + 21\,860 = 235\,550$ N

With the external load, the stresses are the following

- traction stress:
  $\sigma_{max} = 668$ MPa
- torsion stress:
  $\tau_{max} = 173$ MPa
- equivalent stress (Von Mises):
  $\sigma_{eq\,max} = 732$ MPa.

Once again, we have achieved our objective: we can see that using M24 × 3 bolts instead of M20 × 2,5 bolts is an acceptable solution.

However, the increased cost of these bolts is often as highas 40%, and the risks of their coming loose in the minimum case still remain.

It is seen that the calculation with M22 × 2,5 bolts (which is an unusual dimension), leads to maximum stress of $\sigma_{eq\,max} = 839$ MPa.

Therefore, if this solution is chosen we would have to accept a lower safety margin.

Inversely, however, an assembly designed 16 M24 × 3 bolts tightened by the torque method can be re-designed to use 16 M20 × 2,5 bolts tightened by the hydraulic tensioning method.

Notes

E

Simultaneous tightening with bolt tensioners

Simultaneous tightening consists in tensioning several bolts, or even all of the bolts, of an assembly at the same time.

Some simultaneous tightening examples will be found at the end of the chapter.

We have already seen that this tightening process presents important, even decisive, advantages:

- great homogeneity in tightening all the bolts of the assembly,
- straightforward implementation,
- short process times.

Hydraulic bolt tensioning is the most convenient method for obtaining simultaneous tightening.

The following information is of a general nature, and provides a series of examples. Results on actual application may vary depending on the dimensions and tolerances of the components, the quality of the tensioning tooling, and the implementation procedure. A particular application can, if necessary, be examined in greater detail.

Simultaneous tightening of 100% of the bolts

This is the most accurate and quickest method: all the bolts are tightened at the same time (→ **diagram 17**).

However, it requires as many tensioners as there are bolts. Therefore in our previous example, 16 tensioners would be required, including the necessary hoses and hydraulic connections.

The tightening operation is quite simple: each bolt is capped by a tensioner. All the tensioner oil-feed lines are connected to a common source of high-pressure oil. All the bolts are simultaneously stretched (preferably in two steps) to the required load. In our example:
residual load F_0 = 168 000 N for a hydraulic load F_h = 192 400 N using Pilgrim HYDROCAM HTA 20 tensioners pressurised to 98 MPa.

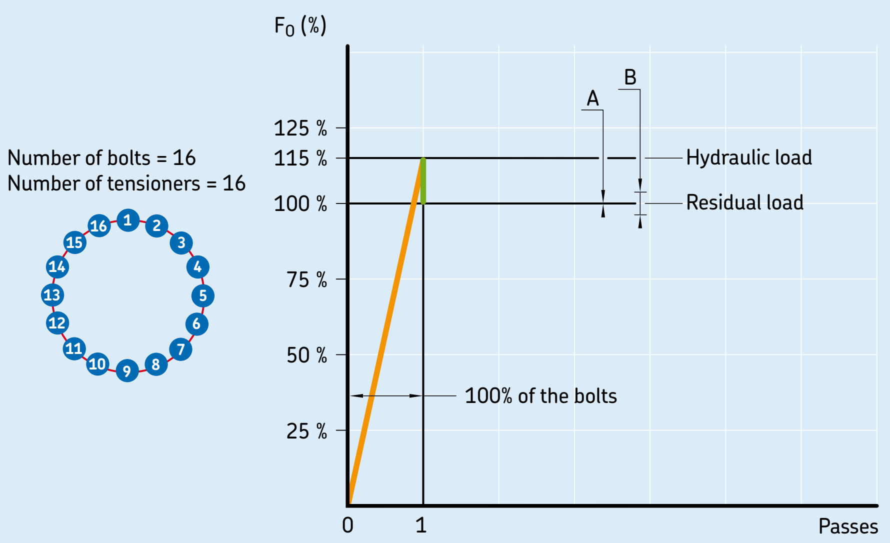
This method provides excellent tightening homogeneity over all the bolts. We know from experience that the previously described uncertainties are significantly reduced:

- the uncertainty on the F_h/F_0 ratio is reduced ±1%
- the uncertainty on the hydraulic load depends on the tensioners only, and is therefore reduced to ±1%
- the uncertainty in the nut clamping moment is à ±2%.

The resulting dispersion in the tightening loads of the different bolts will be as low as ±4%.

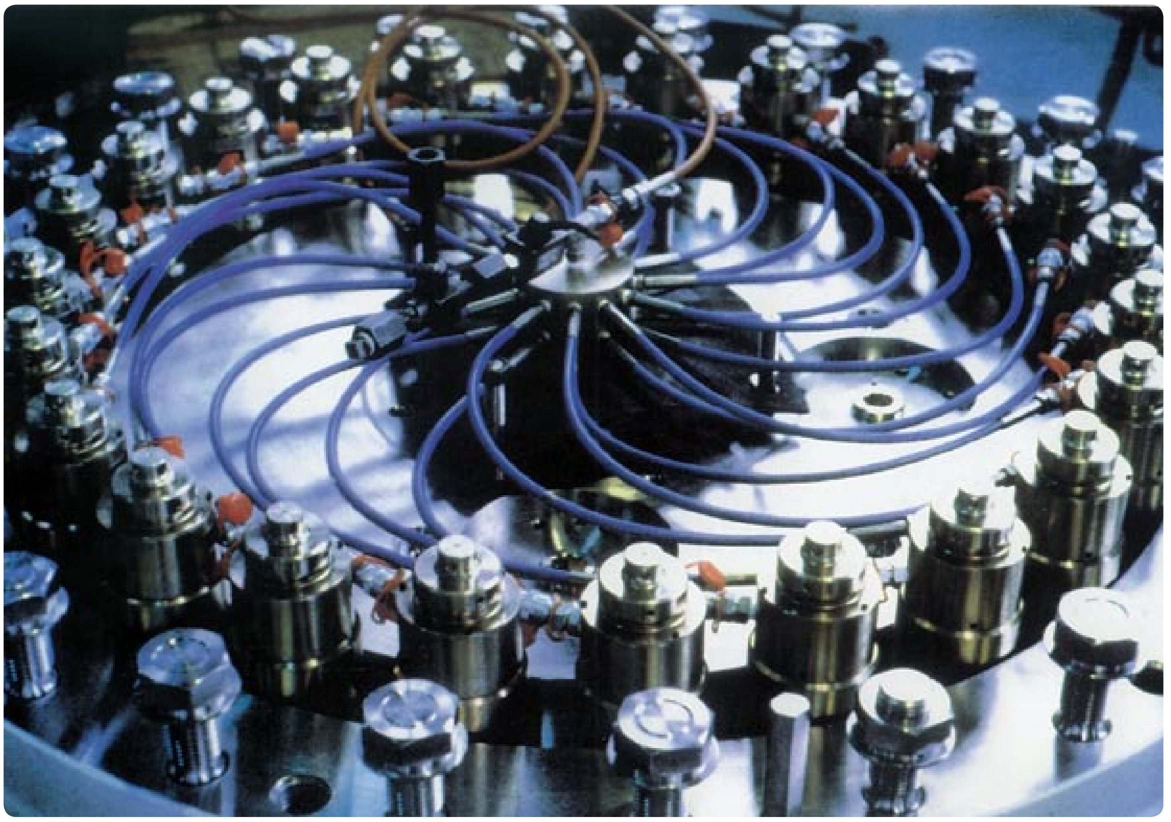
Simultaneous tightening of 100% of the bolts

Diagram 17



A = Dispersion in the average value of the residual tension obtained after tightening, in relation to the desired residual tension: < ±1%
B = Total dispersion in all residual tension values after tightening, in relation to the average value: ±4%

Simultaneous tightening with monobloc ring



Simultaneous tightening with individual tensioners connected to a single oil feed

F

Segmented Ring Tensioning Unit for Primary / Secondary Manway of a Steam Generator

Simultaneous tightening of 50% of the bolts

In this case, every other bolt is tightened in one step, (→ **diagram 18**), so that the number of tensioners required is halved.

In our example, only eight tensioners are required with the necessary hoses and hydraulic connections.

The tightening process requires two rounds of tightening on the first half of the bolts (which we call "even") and at least one round on the second half (which we call "odd") which means three and possibly four tightening rounds.

Assuming that during the first round we apply a hydraulic load of 192 400 N to the even bolts, and settle to a residual load of 168 000 N.

A first round on the second half of the bolts – the odd bolts – obtains the same residual load of 168 000 N.

However, tightening the odd bolts releases some of the load taken by the even ones, and reduces the load by 10%, to an average tightening value of 151 000 N.

A second tightening round is thus required to bring the load back to 168 000 N.

However, this second step will once again cause the second half to loosen by approximately 6%, reducing the tightening load to 158 000 N.

If this level is acceptable, the tightening operations can stop at this point. However, sometimes the second half requires a second round to bring residual load back upto 168 000 N.

This round will affect the first lot by approximately 3%, or an average residual tightening value of 163 000 N, which is often perfectly acceptable.

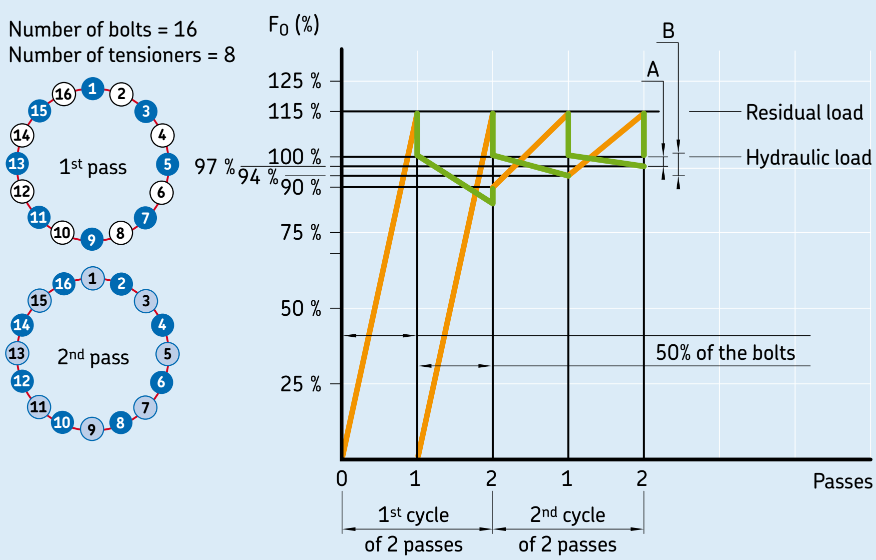
The load homogeneity is not as good as in the case of 100% simultaneous tightening, but the variation remains within an acceptable range of +5%/–7%.

In many applications this deviation can be considered as acceptable.

Naturally, the intervention time is longer than in the preceding case.

Simultaneous tightening of 50% of the bolts

Diagram 18



A = Dispersion in the average value of the residual tension obtained after tightening, in relation to the desired residual tension: +1%/–3%

B = Total dispersion in all residual tension values after tightening, in relation to the average value: +5%/–7%

Example of tool for semi-simultaneous tightening (50%) of cover head on a chemical vessel

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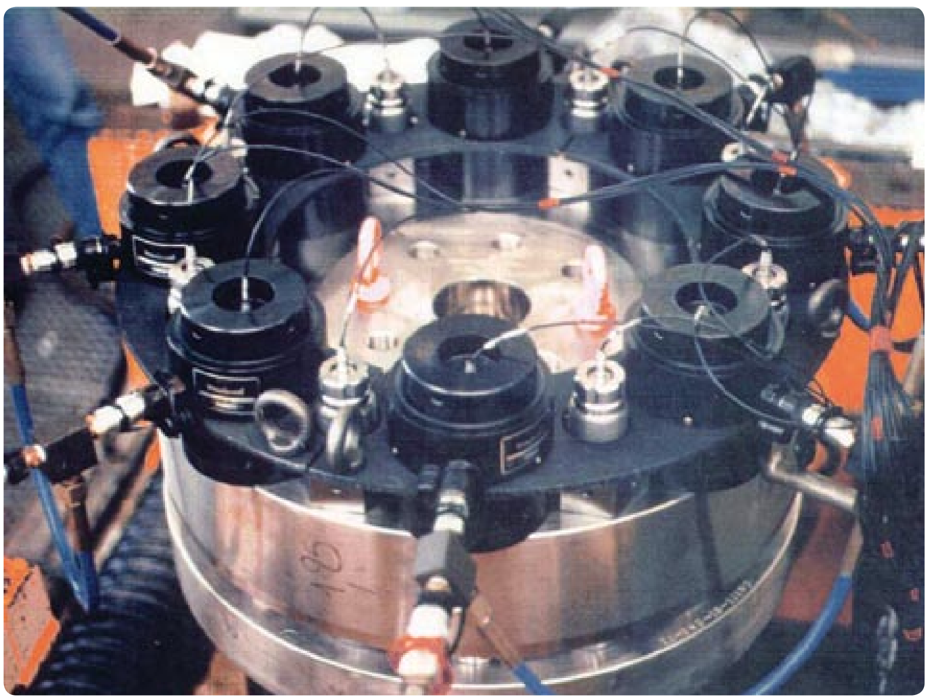


Example of tool for semi-simultaneous tightening (50%) of cover head on a chemical vessel

Simultaneous tightening of 25% of the bolts

In this case, only one bolt in four is tightened at each step (→ **diagram 19**).

The number of tensioners required is one-fourth the total number of bolts, namely four in our example.

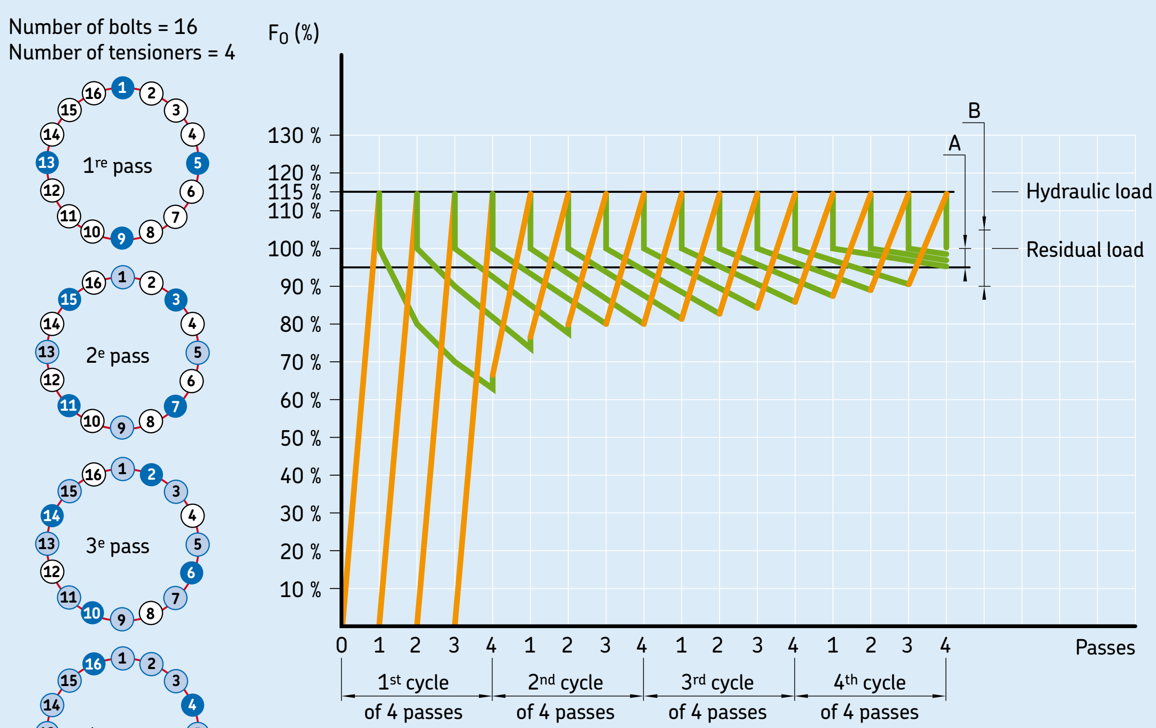
Each set of four bolts will need at least four tightening steps, which means that sixteen steps are required to guarantee acceptable load homogeneity.

The first set of bolts (say numbers 1, 5, 9 and 13) is first tightened with a residual load of 168 000 N, but then will lose 20%, 30% and finally 35% of the load (109 000 N) while the neighbouring sets are successively tightened.

Example of partial simultaneous tightening

Simultaneous tightening of 25% of the bolts

Diagram 19



A = Dispersion in the average value of the residual tension obtained after tightening, in relation to the desired residual tension: +1%/–5%

B = Total dispersion in all residual tension values after tightening, in relation to the average value: +6%/–10%

Four rounds are required before the first set remains within 5% of the nominal load after adjustment of its neighbours.

The load is then 159 600 N. If this variation is compatible with the application, we may stop the operations at this stage.

However, the load homogeneity is obviously not as good as in the case of 100% simultaneous tightening.

The level of variation will lie within the range +6%/–10%.

One more round on the first set of bolts will bring the variation within the range +5%/8%.

Clearly, the use of control means such as smart washers can reduce the uncertainties described above.

Since there are more rounds, the intervention time increases proportionally.

The technical and economic requirements of each application will guide the designer and the installation contractor in the choice of the number of tensioners and the use of measuring devices. However, in the vast majority of cases, simultaneous tightening with hydraulic tensioners is clearly the optimal and most accurate solution.

Overleaf: simultaneous tightening equipment for big diesel engine cylinder head

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Conclusion

The quality of a bolted assembly depends on two interdependent parameters, namely:

- the design of the assembly,
- the method used to tighten the bolts.

It is therefore necessary to choose the suitable tightening method at the beginning of the design phase.

Tightening bolts by the torque method is simple, especially for bolts of reasonable dimensions.

However, the torquing process generates "parasite" torsion stress in the bolt, which weakens the assembly. More importantly, uncertainty regarding the final load remains very high.

Although the use of top-quality hydraulic torque wrenches has slightly improved this situation, the inherent drawbacks of the method are incompatible with the requirements in many applications.

Tightening by heater rods or by mechanical elongation offers some improvement compared with torque-tightening, but these two methods are rarely used due to their costand the time-consuming processes involved.

Hydraulic tensioner tightening offers the best compromise between quality, reliability, safety and accuracy on the one hand, and ease of use, cost and time-saving on the other.

Particularly with regard to costs, this method is not only competitive from the tooling-investment viewpoint but it also provides further savings through optimised design of assemblies and the low cost of site operations.

To summarise, hydraulic bolt-tensioning offers the following advantages:

- no "parasite" torsion stress in the bolts,
- high load precision simply by controlling the hydraulic load,
- fast, safe and easy operation,
- the integrity and the reliability of threads and bearing surfaces are maintained,
- readily compatible with simultaneous tensioning of all the bolts in an assembly (or even several assemblies),
- tightening and measuring operations can be partially or fully automated,
- compatible with a wide variety of bolt materials (stainless steel, titanium, composite materials ...).

In addition, the HYDROCAM bolt tensioners also feature:

- use with a wide range of bolt sizes: 5 to 500 mm,
- easy adaptation due to modular design,
- greater homogeneity due to excellent efficiency (very important for simultaneous tightening),
- easy implementation of simple measuring methods, in particular the "smart washer", which directly measures the final tightening load.

Hydraulic bolt tensioners can generally be used on assemblies which were not initially designed to be tightened with tensioners, provided that it is possible to use bolts with a sufficient protrusion on top of the nut. Nonetheless, it is strongly recommended to make allowance for the use of bolt tensioners from the beginning of the assembly design process: under such circumstances, the full benefits of the method can be achieved.

Bolted assemblies specifically designed to be tightened with bolt tensioners are particularly suited to the stringent requirements of applications requiring a high level of quality and safety, and they are optimised for dimensions and weight.

The Pilgrim HYDROCAM General Catalogue introduces:

- products and services,
- applications,
- tensioner selection criteria
- general instructions and operating precautions.

A full list of these products and services is presented on the following page.

Bibliography :

- French Standard: "Norme AFNOR NF E 25-030 (1984)"
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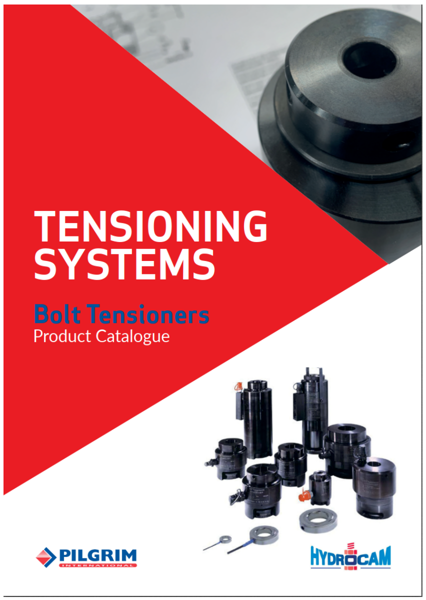

Pilgrim International Ltd

The Engineering Department of Pilgrim develops, designs and manufactures in the UK the complete range of HYDROCAM bolt tensioners.

Backs up the quality of its services and products in the form of ISO 9001/2015 certification.

Bolt-tightening products and services provided by Pilgrim

HYDROCAM bolt tensioners

- Complete standard range including six different types covering a large range of bolt sizes: M 8 to M 160 and loads of 100 kN to 8 500 kN
- Standard tensioners adaptable according to application interface
- Special tensioners designed for dedicated applications, extending the range of bolt sizes from M 5 to M 500.

Sensor washers for measuring tightening loads

Accessories

- Manual pumps delivering various pressure ranges: 700, 1 000 bar or 1 500 bar
- air-driven hydraulic power units delivering various pressure ranges: 700, 1 000, 1 500, 2 000 bar or 3 000 bar
- electrically driven hydraulic power units delivering various pressure ranges: 700, 1 000, 1 500 bar or 2 000 bar
- high-pressure hoses of all lengths; distribution blocks
- pressure intensifiers.

Simultaneous tightening machines and systems (with optional automation)

Standard and customised automatic remote-control systems

Services

- assistance in design of bolted joints,
- assistance in selecting most appropriate tightening method,
- expertise and experimentation,
- industrial partnerships,
- checks and tests,
- training,
- technical assistance/on-site intervention,
- installation and commissioning,
- maintenance and repairs in our workshop or on site,
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