



# Review of the Application of Design Guideline VDI 2230

## Using SR1 for PCS

Written By

By Ralph S. Shoberg, P.E.

## Preliminary Note

In the Guideline VDI 2230 Part 1, bolted joints are treated which have to transmit constant or alternating working loads and which are designed with high duty bolts. A bolted joint is a separable joint between two or more components using one or more bolts. The bolts must be designed such that the joined section fulfils its allotted function and withstands the working loads occurring. Calculation of the bolted joint aims at determining the required bolt dimensions, allowing for the following influential factors:

- Strength grade of the bolt
- Reduction of the assembly preload in the interface or in parts of the interface by the working load
- Reduction of the assembly preload by embedding
- Scatter of the preload during tightening
- Fatigue strength under an alternating load
- Compressive stress on the clamped parts due to the bolt head and/or the nut.

### 1 Range of Validity

The design rules established in this Guideline are valid for steel bolts. The values in the tables are given for the dimension range M4 to M39 and for clamp length ratios  $lk/d < 10$ . Bolts with smaller and larger dimensions can be calculated according to the Guideline. The given characteristics of materials apply only at room temperature. At higher or lower temperatures allowance must be made for their temperature dependence. Extreme operating stresses such as high and low temperatures outside the given limits, corrosion and impact stresses, are not considered in the present Part 1.

The following DIN standards, directives and special regulations should be observed where applicable:

- DIN 18800 Part 1 Steel Structures, Dimensioning and Construction
- DIN 18801 High Steel Constructions; Dimensioning, Design and Manufacturing
- DIN 15018 Part 1 to 3 Cranes; Steel Supporting Constructions, Calculation
- DIN 2505 (preliminary standard) Calculation of Flanged Joints
- DAST-directive 010 Application of High-strength Bolts in Steel Constructions (published by the German Committee for Steel Construction, Stahlbau-Verlag GmbH, Cologne)
- AD Note B7 (January 68) Calculation of Pressure Vessels; Bolts
- AD Note W 7 Materials for Pressure Vessels; Nuts and Bolts in Ferritic Alloy Steels
- TRD 309 Technical Regulations for Boilers; Bolts (published by the Vereinigung der Techn. Überwachungsvereine e.V. Essen, Beuth-Vertrieb, Berlin)
- DIN ISO 898 Part 1 and 2 Mechanical Properties of Fasteners
- DIN 267 Part 1 to 24 Mechanical Fasteners; Technical Delivery Conditions
- DV 804 (preliminary version) Directive for Railway Bridges and other Engineering Structures VEI

### 2 Choice of Calculation Approach for Bolted Joints

The calculation procedure for a bolted joint is fundamentally dependent on its geometry. More elaborately designed multi-bolted joints must be divided into geometry that is easily handled by the calculation. Fig. 1 shows joint geometry with the relevant applied loads, as well as the suggested calculation procedure. As the formulae are derived from different mechanisms acting during loading, the complexity of the calculation procedure increases according to the complexity of the joint.

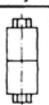
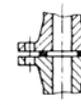
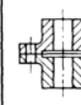
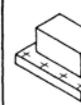
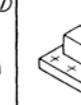
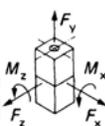
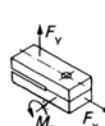
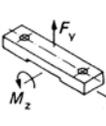
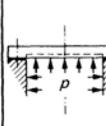
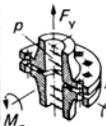
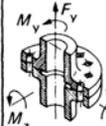
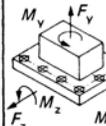
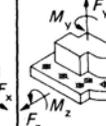
Single-bolted joint		Multi-bolted joint						Bolted joint
Concentric or eccentric		In a plane	Axial symmetry			Symmetrical	Asymmetrical	Bolt axes
Cylinder or prismatic body	Beam	Beam	Circular plate	Flange with sealing gasket	Flange with plane bearing face	Rectangular multi-bolted joint	Multi-bolted joint	
								Joint geometry
								Relevant loads
Axial force $F_A$ Transverse force $F_O$ Bending moment $M_B$	Axial force $F_A$ Transverse force $F_O$ Moment in the plane of the beam $M_z$	Axial force $F_A$ Transverse force $F_O$ Moment in the plane of the beam $M_z$	Internal pressure $p$	Axial force $F_A$ (Pipe force) Bending moment $M_B$ Internal pressure $p$	Axial force $F_A$ Torsional moment $M_T$ Moment $M_B$	Axial force $F_A$ Transverse force $F_O$ Torsional moment $M_T$ Moment $M_B$	Axial force $F_A$ Transverse force $F_O$ Torsional moment $M_T$ Moment $M_B$	Forces & moments
VDI 2230		Limited treatment by VDI 2230		DIN 2505 AD Note B7 VDI 2230	Limited treatment by VDI 2230			
Beam bending theory (VDI 2230) with additional conditions		Plate theory			Limited treatment using simplified models			Calculation procedure
Finite Element Method (FEM)								

Figure 1. Outline of the Bolted Joints

For all joints it is assumed that the bolt axes are parallel to each other in the unloaded state, and that they are also parallel to the normal to the interface line. Bulk plastic deformations are excluded. In the micro-geometrical range, plastic deformation is allowed for as preload loss due to embedding.

From an evaluation of previous theoretical and experimental work the mechanics of the joint geometry shown in Fig. 1 can be described in the following way.

1. Mechanics of the cylindrical single-bolted joint (geometry (1)).
2. Mechanics of the beam connection (geometry (2) and (3)).
3. Mechanics of the circular plate (geometry (4)).
4. Mechanics of the rotation of flanges (geometry (5)).
5. Mechanics of flanged joints with plane bearing face (geometry ((6), (7) and (8))).

From an understanding of the actual design of the joint being considered, the design engineer can select the most appropriate calculation method using the idealized models given.

### 3 Analysis of Force and Deformations

The working load on a bolted joint greatly affects the resilience of the entire joint. Therefore, optimum use of highly stressed bolted joints is only possible by means of thorough and exact consideration of the joint's state of forces and deformation. The behavior of forces and deformation in bolted joints is examined in the following terms:

- Elastic Resilience including the resilience of bolts and the resilience of superimposed clamped parts
- Load and deformation ratios for directly superimposed parts including the assembled state, the operating state, and special cases.

### 4 Calculation Steps

The calculation of a bolted joint is based on the external working load,  $F_B$ , acting on the joint. This working load, and the elastic deformations of the component parts which result, cause an axial force,  $F_A$ , a

transverse force,  $F_Q$ , a bending moment,  $M_B$ , and in some cases a torque moment,  $M_T$ , at the particular bolt location. The generally difficult and large-scale analysis of forces and deformations, involved in the determination of these initial quantities, cannot be addressed by this Guideline because of the large variety of designs of components and bolted joints. These calculations must be made by means of elasto-mechanics. Only for simple symmetrical and relatively stiff joints, can the initial quantities be obtained by a simple analysis of the working load. The values  $F_A$ ,  $F_Q$ ,  $M_B$ , and  $M_T$  are subsequently assumed to be known.

In Section 4.1, the method with calculation steps R1 to R10 is specified, which should mainly be carried out as a preliminary assessment (draft-phase) of bolted joints. It is based on the elementary consideration of the linear dependence of deformations and stresses (linear rule  $\Delta l/l = \sigma / E$ ). In contrast, a non-linear relationship for clamp load eccentricity, dependent on  $F_V$  and  $F_A$ , is derived in Section 4.2, so that all the forces on a bolted joint under equilibrium conditions can be determined.

#### 4.1 Calculation Procedure for the Basic Linear Case

For calculation of the required bolt dimensions, knowing the working load conditions, allowance is made for the following factors:

- A loss of preload,  $F_Z$ , in service due to embedding.
- The assembly preload,  $F_M$ , is reduced in service by a proportion of the axial bolt force,  $F_{PA} = (1 - \phi) F_A$ .
- To meet certain requirements, e.g., sealing functions, prevention of self-loosening, and prevention of one-sided opening of the interface, a minimum clamp load,  $F_{Kerf}$ , is necessary in the joint.
- Finally, allowance is made for the various levels of preload scatter, which will depend upon the assembly method. All of these factors are considered in the main dimensioning formula that forms the basis of the bolt calculation.

The assembly preload,  $F_M$ , in the bolt is a criterion for determining the bolt diameter.  $F_M$  is the load that, in combination with the thread friction torque produced during tightening, utilizes 90% of the minimum yield strength of the bolt. For a given material strength, the bolt will have clamp load,  $F_M$ , related to it, which is at least as high as the calculated maximum assembly preload,  $F_{Mmax}$ . For assembly, the tightening torque,  $M_A$ , is taken from Tables 1 to 4.

When the clamp load, together with the thread torque resulting from tightening, leads to a total stress which reaches 90% of the yield strength or 0.2% extension limit of the bolt material, the difference force,  $F_{SA}$ , resulting from axial force must not exceed  $0.1 R_{P0.2} A_S$ , so that the 0.2% extension limit of the bolt is not exceeded:

$$F_{SA} = \phi_n F_A \leq 0.1 R_{P0.2} A_S$$

If alternating stresses occur in service, the amplitude of the oscillating force  $\pm F_{SAa}$ , must not exceed the endurance limit of the bolt (see calculation step R9).

Finally, the calculation procedure also includes a check of the surface pressure under the bolt head or the nut. The limiting surface pressure of the material should not be exceeded in order to avoid preload loss due to creep.

The working load,  $F_B$ , with its axial force and transverse force components,  $F_A$  and  $F_Q$ , and under certain circumstances the bending moment,  $M_B$ , at the bolting point must be given as the initial conditions.

The design and assembly conditions can usually be selected or influenced; these determine the values to be inserted for embedding and scatter in assembly preload.

Taking account of the appropriate derivations and explanations in Section 3, the calculation procedure can be carried out as follows in the calculation steps R1 to R10:

R1 Rough determination of bolt diameter  $d$ , of clamping length ratio,  $l_K/d$ , and of the average surface pressure under the bolt head.

The appropriate assembly preload,  $F_M$ , for various bolt dimensions and strength grades can be found in Tables 1 to 4 together with Table 5.

Recommendations for maximum allowable surface pressure,  $p_G$ , of various materials are contained in a table. If  $p_G$  is exceeded, the design conditions must be modified (in the extreme case, by using a washer of adequate strength and dimensions). In this case  $l_K/d$  must be re-determined, and the rough dimension calculation must be verified.

R2 Determination of the tightening factor,  $\alpha_A$ , allowing for the selected tightening method and the lubrication of surface condition.

R3 Determination of the minimum required clamping load,  $F_{Kerf}$ , allowing for the following specific requirements:

- Friction grip to absorb a transverse load component,  $F_Q$ , or a torque,  $M_T$
- Sealing functions for known pressures and surfaces, as well as the material characteristics of the sealing elements
- No opening on one side in cases of eccentric loading and/or stressing. Subject to simplifying assumptions, a calculation formula for  $F_{Kerf}$  is possible
- The largest value determined for  $F_{Kerf}$  should be substituted in the design equations

R4 Determination of the load factor,  $\Phi$

Determination of the elastic resilience of the bolt,  $\delta_S$ , and estimation of the load introduction plane  $n * 1_K$

Determination of the elastic resilience of the clamped parts,  $\delta_P$

R5 Determination of the loss of preload due to embedding

R6 Determination of the required bolt size

- a) For all tightening techniques in the elastic range of the bolt, identify a bolt (diameter and strength grade) for which  $F_M \geq F_{Mmax}$

For Tables 1 to 4 are not applicable for specially profiled bolts, the clamping forces and the tightening torque must be calculated from equations. In this case, the coefficient of friction is dependent on the selected lubrication and surface states. The appropriate tightening torque  $M_A$  for the clamping load  $F_M$  can also be determined from the Tables.

- b) For tightening techniques that give or exceed the yield load identify a bolt (size and strength grade) for which  $F_M/0.9 \geq F_{Mmin}$ . The assembly preloads from Tables 1 to 4 are inserted here.

R7 Repetition of the calculation steps R4 to R6, if changes in bolt or clamping length ratio are necessary

R8 Check that the maximum permissible bolt force is not exceeded. The maximum permissible bolt force will not be exceeded if the differential load  $F_{SA}$ , is  $F_{SA} = \Phi F_A \leq 0.1 R_{P0.2} A_S$

For necked-down bolts we have accordingly  $\Phi F_A \leq 0.1 R_{P0.2} A_T$  - In the case of tightening techniques which indicate the yield load or exceed the yield load, this test will be replaced by tests to determine whether an additional plastic elongation of the bolt by the working load is permissible, to what extent the bolt can undergo plastic elongation, and how frequently the bolt can be reused.

**R9 Determination of the alternating stress endurance of the bolt**

For eccentric load application, allowance must also be made for bending stress. For an increasing working load, this calculation step is simplified.

Approximate values for the permissible stress deviation,  $\sigma_A$  (stress amplitude), of the endurance limit can be obtained. If this condition is not satisfied, the design must be improved if possible, or a bolt with a larger diameter or greater endurance limit must be used.

**R10 Checking calculation of surface pressure under the bolt head and nut bearing area**

In determining the bearing area,  $A_P$ , allowance must be made for chamfering of the hole.

Recommendations on the maximum allowable surface pressure,  $p_G$ , for various materials are given.

For tightening techniques that indicate the yield load and exceed the yield load there is a calculation.

## **5 Influencing Factors**

Allowance must be made for a number of factors that are dependent on the material and surface design of the clamping parts and the clamped surfaces, the shape of the selected bolts and nuts, and on the assembly conditions. The wide range of these influential factors is discussed in this section.

### **5.1 Strength of the Bolt**

The bolt size necessary for safe operation can be reduced by means of a bolt material with increased tensile strength and increased yield strength or 0.2% proof stress (saving of space and weight). The screw mechanical properties (strength and ductility) are according to the standard DIN ISO 898 Part 1.

Generally, dimensioning of the bolt is done so that the stress on the bolt, composed of the torsional and axial stresses during tightening (reduced stress  $\sigma_{red}$ ) together with the additional stress,  $\sigma_{SAb}$ , resulting from the working load, does not exceed the yield load or 0.2% proof stress of the bolt material.

However, with adequate bolt material ductility and bolt toughness, the yield load or 0.2% proof stress can be exceeded without danger up to certain limits. This occurs when techniques are used that tighten to yield or beyond, in order to obtain a specific assembly preload with a relatively low scatter.

Bolts whose mechanical properties meet the requirements of DIN ISO 898 Part I are generally capable of withstanding the high local stresses, which occur during the tightening process, without affecting the bolt's service performance. The maximum force, which the thread section can withstand before fracture, is calculated.

### **5.2 Minimum Thread Engagement**

#### **5.2.1 Bolt-Nut Strength Matching**

The full utilization of a bolted joint for tensile loading, i.e., its capability for loading up to the fracture point of the unengaged loaded thread section or reduced shank section, can only be achieved with an adequate length of engagement of the bolt thread in the nut thread.

The design principle which requires that overload fracture occurs in the loaded unengaged thread requires that the shear strength of the engaged screw and nut threads is at least as high as the tensile strength of the loaded unengaged thread. To ensure this, a minimum thread engagement and/or a

minimum nut height,  $m_{kr}$ , are necessary. The critical engagement length, for which the shear strength of the engaged threads is equivalent to the tensile strength of the unengaged loaded threads, depends on several factors, some of which interact with each other:

- The thread form
- The pitch
- The thread tolerance (flank envelope)
- The thread diameter
- The form of the nut (wrenching width), the bolt hole
- The strength and ductility of the bolt and nut materials (material selection, manufacturing process, heat treatment)
- The type of stress (tensile-, torsional-, bending stress)
- The friction coefficients (tightening torque), the number of tightenings

If the thread engagement is less than the critical value,  $m_{kr}$ , either the screw- or the nut thread, whichever has the lowest strength, will shear. Therefore, in this case the weak point of the thread is the shear strength of its engaged section. The unengaged loaded part of the thread is not damaged. In this particular range, the breaking load of the joint increases linearly with thread engagement length up to the point where the shear strength of the engaged bolt- and nut thread equals the tensile strength of the bolt.

The intersection of the increasing straight line of shear strength versus thread engagement with the line for thread tensile strength, which is independent of thread engagement, identifies the critical engagement length,  $m_{kr}$ .

It is true that an increase of the nut height exceeding the critical engagement length will increase shear strength of the engaged thread, but this has no influence on the tensile strength of the free part of the thread. That remains constant. Therefore, an increase of the holding power of the joint by increasing the nut height beyond the critical range is not possible for tensile loading.

### 5.2.2 Calculation of the Required Nut Height

The calculation model for the calculation of the required height of the nut allows for the geometry and mechanical properties of joint elements, nut and bolt, and predicts the type of failure caused by overloading, e.g., shearing of the screw or nut thread or breaking of the screw. For calculating the loading capacity of threaded joints, the following essential factors are allowed for which influence the tensile strength:

Dimensional values, such as tensile cross-section of the bolt thread, shearing area of the internal and external threads, the thread engagement length, pitch, dimensions of the thread, thread tolerances, thread form, nut form (wrenching width), and the clearance hole for the threaded bolt.

The effective thread engagement,  $m_{eff}$ , is calculated from the difference between the total height of the nut,  $m_{ges}$ , and the inside chamfer on both sides running into the bearing surfaces. It is assumed that the chamfered regions have only 40% of the loading capacity of the fully formed thread of the same height.

Expansion of the nut, caused by the radial component of the axial load being applied to the thread flanks, reduces the effective shearing areas of the bolt and nut threads.

The relationship of the shearing forces determines the amount of plastic deformation between the bolt thread and the nut thread. This plastic deformation of the threads reduces the effective shearing area and decreases the angle between the loaded flank of the thread and the bolt axis. Therefore, the radial component increases, the expansion of the nut is increased, causing the shear strength to be reduced.

The dynamic friction occurring due to relative movements between the thread flanks of the bolt and nut during tightening which has a lower friction coefficient than static friction, reduces the shear strength of the joint compared with its shear strength under purely axial load. This reduction can amount to 10 to

15%. In principle, however, the load capacity of the engaged loaded part of the bolt thread is also reduced under combined torsional and axial stresses. But the degree of reduction of shear strength by twisting the bolt or nut during tightening is about 5% lower than the degree of reduction in strength of the unengaged thread. In the limiting case, this may cause the failure mode 'shearing of the thread' occurring under pure tensile loading, to change to a fracture in the unengaged loaded thread during the tightening process.

### 5.3 Surface Pressure at the Bolt Head/Nut Bearing Areas

Surface pressures capable of causing creep processes (time-dependent plastic flow) giving a loss in preload (relaxation) should not be allowed in the bearing areas between the bolt head or nut and the clamped parts, either as a result of the preload or as a result of the maximum service load. Thus, the surface pressure that is calculated from the maximum load should not exceed the compressive yield point of the clamped material.

If creep cannot be avoided, estimation must be made of the remaining preload after the end of the relaxation process.

The initial preload must then be set high enough so that the remaining clamp load guarantees safe operation of the joint.

Experimentally determined limiting surface pressures are given.

If washers are used to reduce the surface pressure, care must be taken to ensure that they have sufficient strength and thickness.

Information on the computation of limiting surface pressures is given in Section 4.7 (calculation step R10).

### 5.4 Tightening Factor, $\alpha_A$ , (Assembly Technique)

Determination of the dimensions and strength grade of a bolt depends on the magnitude of the required preload. The assembly preload can be obtained with different tightening methods. In most cases, the bolts are subject to a torsional stress caused by pitch and friction in the thread, in addition to the axial tensile stress. Allowance must be made for this in determining the total stress in the bolt.

If the torsional stress increases - with a constant tensile stress - the total stress in the bolt increases. This can eventually require a larger diameter or higher strength grade. The tightening methods currently in use do not measure the preload produced directly, but indirectly, e.g., as a function of the tightening torque, of the elastic elongation, of the angle of rotation or by determination of the yield point of the bolt. In almost all these cases some uncertainty remains concerning the level of tensile/torsional stress. This is due to the relatively high scatter in friction coefficients and errors in the method of controlling the preload. Therefore, it is necessary to over-design by an amount reflected in the tightening factor,  $\alpha_A$ .

The value of tightening factor,  $\alpha_A$ , used in the design equation depends on the selected tightening method. With increasing confidence in level of preload attained,  $\alpha_A$  decreases, and with it the requirement for over-dimensioning. Thus, the assembly technique considerably influences the required bolt dimensions. Therefore, the selected tightening technique must be carefully considered, since it is taken as a basis for the calculation.

#### 5.4.1 Bolt Stresses during Tightening

In most cases the preload is generated by turning the nut or bolt. Thus, the bolt is subject to a torsional stress in addition to an axial stress. For some tightening methods, however, e.g., hydraulic tensioning, the bolt is stressed only axially during preloading.

The relationship between the assembly preload,  $F_M$ , and the torsional moment,  $M_G$ , due to preload in the thread can be derived from the mechanics of oblique planes.

During tightening, the yield point and highest tensile stress are influenced by the tensile stress,  $\sigma_M$ , and torsional stress,  $\tau$ , acting simultaneously. For calculation of tensile stress, the total stress resulting from these factors is reduced to an equivalent uniaxial state of stress (equivalent stress  $\sigma_{red}$ ) by means of a yield criterion.

#### 5.4.2 Scatter of the Assembly Preload During Tightening

The assembly preload in bolted joints is influenced by numerous factors, such as:

- The friction conditions (see Tables 5 and 6) in the surfaces moving relative to each other
- The geometrical shape of the joint
- The tightening method
- The tightening tool

Errors in estimating the friction coefficients scatter in friction coefficients within one lot of bolts, tightening methods varying in precision, as well as instrument/setting/reading error lead to scatter in these influential factors at various levels. Especially if the assembly preload is applied indirectly, e.g., by a tightening torque, these factors result in scatter of the preload achieved. Thus, the bolt must then be oversized, by an amount related to the tightening factor,  $\alpha_A$ .

The  $\alpha_A$  that accommodates the scatter in assembly preload between  $F_{Mmin}$  and  $F_{Mmax}$  as a result of the scatter in friction coefficients is introduced in equation form.

For a constant required minimum preload, an increasing tightening factor means that the bolt must be designed for a higher maximum assembly preload resulting from the higher scatter. This can be achieved for example by a proportionately larger bolt cross section.

#### 5.4.3 Assembly Techniques

##### 5.4.3.1 Torque Controlled Tightening

Torque controlled tightening generally implies tightening using controlled or indicating torque wrenches. In principle, however, motorized tightening using bolt installation spindles also comes within this concept since a compressed air spindle, for example, gives a measurable torque adjustment.

##### 5.4.3.2 Yield Controlled Tightening

For yield tightening with an electronic control system, the yield point of the bolt is automatically identified. This is done by measuring the torque and the rotation angle during tightening and by determining their difference quotient, which is equivalent to the slope of a tangent to the torque/angle curve. As soon as the yield point is reached, the gradient falls. This drop to a predetermined fraction of the stored maximum value obtained from the linear part of the torque/angle curve activates the shut-off signal. Since undulations occur in the early part of the curve, resulting from the elastic and plastic deformations as the interfaces are brought together which could lead to premature shut-off of the tightening spindle, a snug torque is required. Therefore, comparison of the gradient values and storing of the maximum gradient only starts after the snug torque has been exceeded.

##### 5.4.3.3 Angle-Controlled Tightening

Angle controlled tightening is indirectly a method of length measurement. Not only is the compression of the clamped parts measured at the same time as the elongation of the bolt, but also the plastic and elastic deformations occurring before complete closing of the interfaces which are irregular and cannot be predetermined. Therefore, in practice, the joint is preloaded - as for yield tightening - by means of a snug

torque until all the interfaces are completely closed. The angle of rotation is only measured from this snug point.

Practice has shown that this technique reaches its highest precision when the bolt is tightened into the plastic range because in this range the deformation line is roughly horizontal, and angle errors have almost no effect. In contrast to this, in the elastic range, angle errors can occur in the steep part of the deformation line, and would cause preload errors of similar order of magnitude to that of torque controlled tightening.

#### **5.4.3.4 Momentum-Controlled Tightening (Impact Wrenches)**

Impact wrenches transfer energy by means of momentum; it is thus almost impossible to measure the resulting torque. Like bolt installation spindles, impact wrenches must also be adjusted on the original bolted joint; this is done most accurately by measuring the elongation of the bolt and less accurately, using the re-tightening torque. The error considerations are essentially the same as those for bolt installation spindles. The tightening factors are so high that this tightening technique cannot be recommended for high duty bolted joints.

#### **5.4.3.5 Hydraulic Tensioning**

For hydraulic tensioning, the bolt is held at its free end above the nut position and is loaded in tension relative to the clamped parts. As this occurs, the nut is lifted off the clamped parts, and is repositioned against the clamped parts by rotating it. After unloading, a defined preload remains in the bolt. A disadvantage is the fact that the bolt must be tensioned beyond the specified preload since elastic and plastic deformation in the thread and the clamped parts occurs only after unloading and, thus, leads to a loss of preload. The position of the joining surfaces (thread and nut bearing face), which are not stressed during pre-tensioning, changes during removal of the pre-tension, and this can be compensated for by means of repeated re-tightening. Often a general re-tightening is specified in order to compensate for preload variation resulting from the preloading of the adjacent bolt.

Hydraulic tensioning, e.g., as it is employed for the construction of large pressure vessels, has the advantage that several bolted joints can be tightened to the same preload simultaneously. The precision that can be achieved for defined preloads depends mainly on the length of the screw. The tightening factor takes on values between 1.2 and 1.6, where the smallest values are for long screws and the largest values are for short screws.

### **5.5 Fatigue Strength**

For bolted joints, a high notch effect is present at the first stressed screw thread. The notch factor can have values between 4 and 10, depending on the configuration.

For alternating stress, therefore, the tensile load capability of bolted joints is reduced to less than 10% of their capability under uniform tensile stress.

This tensile load capability under alternating stress--the endurance limit--is determined by numerous factors.

On the basis of qualitative judgment, a table is provided that shows possible methods for improving the endurance limit of bolted joints.

Besides thread rolling after heat-treating, modifications of the design can effectively improve the endurance limit of bolted joints. Modifications of the design contribute to minimize the high local stresses that must be absorbed by the bolt for a given working load. These modifications can consist of causing a more even stress distribution within the nut, and reducing the portion  $F_s$ , of the working load component,  $F_A$ , thus reducing the additional stress felt by the bolt.

Modifications of the design generally do not improve the endurance limit of the thread but reduce the local stresses. This increases the endurance limit indirectly. A direct improvement of the endurance limit of the thread is only possible to a very limited extent because of the dominating influence of the notch effect (except by inherent compressive stresses induced during manufacturing by thread rolling after heat treatment).

## 5.6 Embedding

Besides the elastic deformations in the bolted joint, embedding also occurs which is mainly caused by the flattening of surface irregularities. The amount of embedding has an effect on forces in the joint and reduces the elastic deformation, thus reducing the assembly preload.

The amount of embedding is generally smaller than expected from the roughness of the clamped interfaces, because during tightening a considerable flattening has taken place, and even for alternating working loads, the stress variation at the interfaces remains relatively small.

According to currently available experimental results, the total amount of embedding is nearly independent of the number of interfaces, and of the surface roughness of the interfaces, but it increases with the clamping length ratio.

The amount of calculated embedding is only appropriate if the values given for the limiting surface pressure are not exceeded. Otherwise the clamped material at the head and/or nut bearing area is subject to creep and the amounts of embedding can increase uncontrollably.

## 5.7 Self-loosening and its Prevention

Under service stresses, the preload in a bolted joint can drop for two totally different reasons:

- Relaxation - as a result of embedding or creep in the bolted joint
- Rotational loosening (self-rotating) - as a result of relative movements between the mating surfaces

Thus, distinction can be made between 'embedding prevention features' to overcome creep and embedding and 'self-rotation prevention' which can arrest or prevent the 'internal' off torque; 'component loss prevention' cannot prevent partial self-rotation, but prevents the joint components from completely falling apart.

A bolted joint which is well designed for the working load, and which has been tightened up reliably does generally not need additional safety features. On the contrary, by improper use of safety elements the reliability of a bolted joint can be affected. If self-loosening cannot be prevented by means of joint design (e.g., high preload, long bolts), a locking element must be inserted which is suitable for the given stress. The locking elements currently available on the market can be classified in five groups according to their function and in only three groups according to their efficiency.

Bolted joints that are loaded only axially are generally not in danger, i.e., they do not need to be secured. Only for very short bolts of the lower strength grades can the use of elastic elements be recommended, which are clamped within the joint (for prevention of embedding). The elastic effect, however, must also be present under full preload and at the highest working load.

In the case of stresses that are predominantly at 90 degrees to the bolt axis, transverse slippage between the clamped parts can occur as a result of insufficient preload, which leads to relative movements in the thread. Hence, the friction grip in the joint is neutralized. Under the wedge effect of the thread, an internal off-torque occurs which can lead to self-rotating under full preload. Efficient prevention of self-rotation can neutralize this internal off torque or can prevent it, e.g., serrated bearing face bolts or adhesive locking elements.

However, due attention must be paid to the fact that for bolted joints that are permanently subject to transverse displacements, there is a danger of a fatigue failure due to bending.

If a certain drop in preload can be tolerated by removing the self locking but it is required that the joint components do not fall apart (suffer loss) then loss-prevention devices can be considered, e.g., clamping elements. After installation and stressing, the remaining preloads can be 60% (good loss-prevention) or only 10% (poor loss-prevention) of the original preload.

## 6 Calculation Example

### 6.1 Calculations of a Bolted Joint between a Piston and a Piston rod in a Hydraulic Cylinder as an Example of Concentric Clamping and Concentric Loading

#### 6.1.1 Initial Conditions

The bolted joint shown in Fig. 2 is calculated as a concentrically clamped concentrically loaded joint. At an internal pressure of  $P_{max} = 5.5 \text{ N/mm}^2$  and a stressed surface area of  $A = (80 - 25)^2 \pi/4 = 4,536 \text{ mm}^2$ , the axial load is calculated as  $F_A = 24.9 \cdot 10^3 \text{ N}$ .

Measurements in mm:

$$l_1 = 24 \text{ from DIN 912}$$

$$l_2 = 0$$

$$l_K = 42$$

$$l_3 = l_K - l_1 = 18$$

$$L = 60$$

$$t = 13 \text{ (fixed)}$$

$$D_i = 25$$

$$D_A = 80$$

The cylinder is part of a press with 300 working strokes per hour. The axial load should thus be considered as a dynamic working load.

Because the bolt is also required to perform a sealing function, then the residual clamping load after unloading by the working load should be not less than  $F_{KR} = 10^3 \text{ N}$  for safety reasons.

C45V is specified as the piston material.

It is assumed that the joint is tightened using an indicating torque wrench.

#### 6.1.2 Calculation Procedure

The joint is calculated following the calculation steps R1 to R10.

R1 Rough determination of the bolt diameter and the clamping length ratio  $l_K/d$  as well as the rough determination of the surface pressure under the bolt head.

For a concentrically applied dynamic working load  $F_A = 24.9 \cdot 10^3 \text{ N}$  acting in the axial direction of the bolt, a required minimum preload of  $F_{Mmin} = 40 \cdot 10^3 \text{ N}$  is estimated. From the possible diameter-strength combinations, a DIN 912 - M 12 x 60 - 10.9 cylindrical screw was chosen.

The lengths  $l_1$ ,  $l_3$ , and  $t$  are thus fixed as  $l_1 = 24 \text{ mm}$ ,  $l_3 = 18 \text{ mm}$ ,  $t = 13 \text{ mm}$ .

The clamping length is  $l_K = l - t - 5 = 42 \text{ mm}$ .

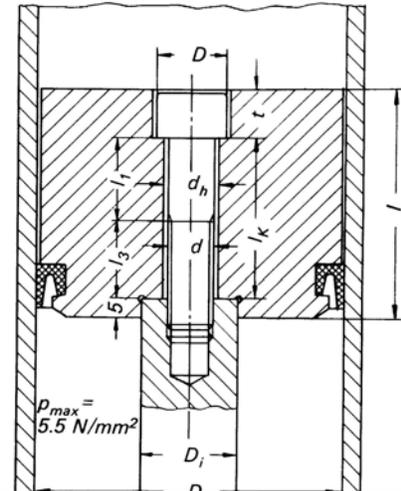


Figure 2. Bolted Joint Example: Hydraulic cylinder with central bolted joint between

The clamping length ratio is  $l_K/d = 42/12 = 3.5$

The surface pressure under the bolt head is:  $p = \frac{F_M / 0.9}{A_p} \leq p_G$

$F_M = 56 * 10^3$  N from Table 1 for  $\mu_G = 0.14$ .  $A_p = 90 \text{ mm}^2$  with  $d_W = 17.23$  (from DIN 912) and  $d_h = 13.5$  (from DIN ISO 273).  $p$  thus becomes:

$$p = \frac{56 * 10^3}{0.9 * 90} = 691 \text{ N/mm}^2$$

The limiting surface pressure for C45V becomes:  $p_G = 700 \text{ N/mm}^2 \Rightarrow 691 \text{ N/mm}^2$

R2 Determination of the tightening factor  $\alpha_A$

It is assumed that the bolt is tightened using an indicating torque wrench, thus  $\alpha_A = 1.6$ .

R3 Determination of the required minimum clamp load  $F_{Kerf}$

The required minimum clamp load must be at least equal to the minimum residual preload as stated in the initial conditions ( $F_{Kerf} = F_{KR}$ ); we thus have  $F_{Kerf} = 10^3$  N.

R4 Determination of the load factor  $\Phi$

$$\Phi = \frac{\delta_p}{\delta_p + \delta_s}$$

With the equations in Section 3.1.1, the elastic resilience of the bolt is determined to:

$$\delta_s = \delta_K + \delta_1 + \delta_2 + \delta_{GM}$$

$$\delta_K = \frac{0.4 * d}{E_s * A_N} = 0.207 * 10^{-6} \text{ mm/N}$$

$$\delta_1 = \frac{l_1}{E_s * A_N} = 1.032 * 10^{-6} \text{ mm/N}$$

$$\delta_3 = \frac{l_3}{E_s * A_{d3}} = 1.152 * 10^{-6} \text{ mm/N}$$

$$\delta_{GM} = \delta_G + \delta_M$$

$$\delta_{GM} = \frac{l_G}{E_s * A_{d3}} + \frac{l_M}{E_s * A_N} = 0.592 * 10^{-6} \text{ mm/N}$$

$$\delta_s = 2.99 * 10^{-6} \text{ mm/N}$$

With the following:

$$l_1 = 24 \text{ mm}, l_3 = 18 \text{ mm}$$

$$l_G = 0.5 d = 6 \text{ mm}$$

$$l_M = 0.4 d = 4.8 \text{ mm}$$

$$d_3 = 9.853 \text{ mm}, A_N = 113 \text{ mm}^2$$

$$A_{d3} = 76.2 \text{ mm}^2$$

$$E_s = 205 * 10^3 \text{ N/mm}^2$$

With equation the elastic resilience of the clamped parts  $\delta_p$ , is calculated as:

$$\delta_p = \frac{l_K}{A_{ers} * E_P} = 0.39 * 10^{-6} \text{ mm} / \text{N}$$

With  $A_{ers}$  from equation, with  $E_P = 205 * 10^3 \text{ N/mm}^2$  and with  $D_A = d_W + l_K$

$$A_{ers} = \frac{\pi}{4} (d_W^2 - d_h^2) + \frac{\pi}{8} * d_W * l_K [(x+1)^2 - 1]$$

$$A_{ers} = \frac{\pi}{4} (17.23^2 - 13.5^2) + \frac{\pi}{8} * 17.23 * 42 [(0.591+1)^2 - 1]$$

$$A_{ers} = 522.4 \text{ mm}^2$$

With the following:

$$x = \sqrt[3]{\frac{l_K * d_W}{(d_W + l_K)^2}} = 0.591$$

Thus, we have:

$$\Phi = \Phi_K = \frac{\delta_p}{\delta_p + \delta_s} = 0.115$$

It is estimated that the load introduction planes are at a distance  $n l_K = 0.3 l_K$ . Therefore  $\phi = \phi n = 0.3 \Phi_K = 0.035$

#### R5 Determination of the loss of preload due to embedding

For a clamping length ratio of 3.5 there is a total amount of embedding of  $f_Z = 5 * 10^{-3} \text{ mm}$  (rounded).

Thus, the preload loss is:

$$F_Z = f_Z \frac{1}{\delta_s + \delta_p} = 1.48 * 10^3 \text{ N}$$

#### R6 Determination of the required bolt size

According to equation, we have

$$\text{req. } F_{Mmax} = \alpha_A [F_{Kerf} + (1 - \Phi) F_A + F_Z] = 42.4 * 10^3 \text{ N}$$

For the cylindrical screw DIN 912 - M 12 x 60 - 10.9, with  $\mu_G = 0.14$ , an assembly preload is determined of  $F_M = 56 * 10^3 \text{ N} > \text{req. } F_{Mmax}$ .

The selected bolt meets the requirements.

The corresponding tightening torque for  $\mu_{Kmin} = 0.10$  is  $M_A = 104 \text{ Nm}$ .

- R7 Repetition of calculation steps R4 to R6 in order to effect a required change of bolt and/or clamping length is omitted.

Checking that maximum permissible bolt force is not exceeded. According to equation, the following must be fulfilled:

$$\Phi_n * F_A \leq 0.1 R_{p0.2} * A_S$$

$$(R_{p0.2} = 940 \text{ N/mm}^2 \text{ from DIN ISO 898 Part 1; } A_S = 84.3 \text{ mm}^2)$$

Substituting numerical values, this relationship becomes:

$$0.035 * 24.9 * 10^3 \text{ N}$$

$$< 0.1 * 940 \text{ N/mm}^2 * 84.3 \text{ mm}^2$$

$$872 \text{ N} < 7924 \text{ N}$$

The conditions of the equation are satisfied, i.e., the maximum bolt force is not exceeded.

- R9 Determination of the alternating stress endurance of the bolt. We have the following:

$$\sigma_a = \pm \frac{\sigma_{SA}}{2} = \pm \frac{\Phi_n F_A}{2 A_{d3}} \langle \sigma_A$$

$$\sigma_a = \pm 51 \text{ N/mm}^2$$

$$\sigma_a = \pm \frac{0.035 * 24.9 * 10^3}{2 * 76.2} = \pm 6 \text{ N/mm}^2 \langle \sigma_A$$

- R10 Checking calculation of the surface pressure below the bolt head bearing surface.

See proof in R1.

$$p = \frac{F_M + \Phi F_A}{A_p} = \frac{56 * 10^3 + 0.035 * 24.9 * 10^3}{90} = 632 \text{ N/mm}^2 \langle p_G = 700 \text{ N/mm}^2$$

Result: The bolted joint from Fig. 2 is secure for the estimated working load.

For more information about this and other related Fastening Technology information and products, please contact:

**RS Technologies, a Division of PCB Load & Torque, Inc.**

24350 Indoplex Circle  
Farmington Hills, MI 48335  
Toll-Free in the USA: 888-684-2894  
24 Hour SensorLine™: 716-684-0001  
Fax: 248-888-8266  
Email: [rsinfo@pcbloadtorque.com](mailto:rsinfo@pcbloadtorque.com)  
[www.pcbloadtorque.com](http://www.pcbloadtorque.com)





**3425 Walden Avenue, Depew, NY 14043 USA**

pcb.com | info@pcb.com | 800 828 8840 | +1 716 684 0001

© 2021 PCB Piezotronics - all rights reserved. PCB Piezotronics is a wholly-owned subsidiary of Amphenol Corporation. Endevo is an assumed name of PCB Piezotronics of North Carolina, Inc., which is a wholly-owned subsidiary of PCB Piezotronics, Inc. Accumetrics, Inc. and The Modal Shop, Inc. are wholly-owned subsidiaries of PCB Piezotronics, Inc. IMI Sensors and Larson Davis are Divisions of PCB Piezotronics, Inc. Except for any third party marks for which attribution is provided herein, the company names and product names used in this document may be the registered trademarks or unregistered trademarks of PCB Piezotronics, Inc., PCB Piezotronics of North Carolina, Inc. (d/b/a Endevo), The Modal Shop, Inc. or Accumetrics, Inc. Detailed trademark ownership information is available at [www.pcb.com/trademarkownership](http://www.pcb.com/trademarkownership).

WPL\_22\_1121