BOLTED JOINTS TECHNOLOGY

Design & Calculation Guide

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I. Objective:

The objective of this training material is to introduce the systematic approach and methods of calculation of high duty bolted joints connections.

Engineers and Technicians attending this training will be exposed to methods of joint calculation extracted from the VDI 2230 (February 2003) applied as a guideline to the design of bolted joints connections of steel bolts of 60 degree thread forms in high-duty and high-strength joints, ranging from M4 to M39 thread sizes. The concepts presented can also be extended analogically to bolts made of other materials and lower strengths also called property classes.

II. The Development of Bolted Joints Technology:

The development of the methods of calculating and designing bolted joints has happened in phases over the past 200 years. The advance of the industrialization on the 19th Century, pushed for a need of systematic calculations of structural equipment and machinery i.e.: railroad, bridges and boilers where one of the structural parts were the bolts. Until then, the calculation of bolts was only made by dividing the load to what that bolt would be submitted, by the smaller sectional area of that element. At that time, this approach would be satisfactory as long as the calculated stress was lower than the ultimate stress resistance of the bolt times a safety factor.

The second phase came when Woller found out that under alternated loads the materials had its strength reduced, and by C. Von Bach who in 1891 who first distinguished static and dynamic loads and also made the discover that bolts are submitted to a multi-axial state of tension during assembling. He then proposed the combined load to be introduced in the calculations of bolts.

At the beginning of the 20th Century, Von Camerer made inferences that external workloads were also taken by the bolts as well as that the joint could decompress under these workloads. Beyond that, he also considered the joint and bolt resilience and its effect on the magnitude of partial forces. Oddly, his discoveries and considerations were neglected by decades after.

In 1927, based on the principals developed by C. Von Bach e Camerer, Rotscher presented the Bolted Joints Diagram, allowing the separation of the workload in partial loads. This diagram is still used by fastening designers as a baseline to calculate bolted joint connections.

In the 1930s Dr. Thun and his students concentrated efforts on the studies of bolts and joints subjected to fatigue. The book of Wiegand and Hass, published in 1940 for the first time separates static and dynamic load resistances. These calculations are basically and structured in the concept that the bolt dimension must be calculated in function of a pre-load that is extended as a multiple of a workload and safety factors are recommended arbitrarily. For the utilization of the bolt due to the pre-load, the torsional tension developed during the tightening is recognized by the hypothesis of energy of deformation (Huber, Von Mises, Hencky).

Using the calculus of the resilience of the bolt and the joint, the bolted joint diagram is determined by the workload acting on the bolt, which can promote the joint decompression. The next step is to verify that the external workload in the bolt does not overcome the bolt Yield Strength and do not promote joint interface separation.

Post war and up to the 60s, the bolted joints technology was more focused on the development of bolts as isolated elements, especially in the USA. Most of the developments happened in the area of materials and manufacturing processes i.e.: thread rolling after heat-treatment, modifications on the radius of the thread root etc.

In Germany, between the 60s and 70s, several studies were done about the resilience of joints revising Bach's and Rotscher's concepts. In addition, studies were introduced about the clamp-load dispersion and pre-load loss due to relaxation. At the same time, studies were developed at the Technical University of Berlin about the behavior of bolted joints eccentrically clamped and eccentrically loaded, what introduced a revision on the concept that the pre-load had no influence in the additional load of the bolt. This lead to modifications on the first conceptual diagram of bolted joints.

All these concepts were introduced on the VDI 2230 as calculation recommendations on its first edition of December of 1974 and then revised in 1977, 1983, 1986, 2001 and again in 2003.

III. Preliminary Notes:

This training material is, when possible, based on concepts extracted from the VDI 2230 and treats on a summarized approach the behavior of Bolted Joints Connections.

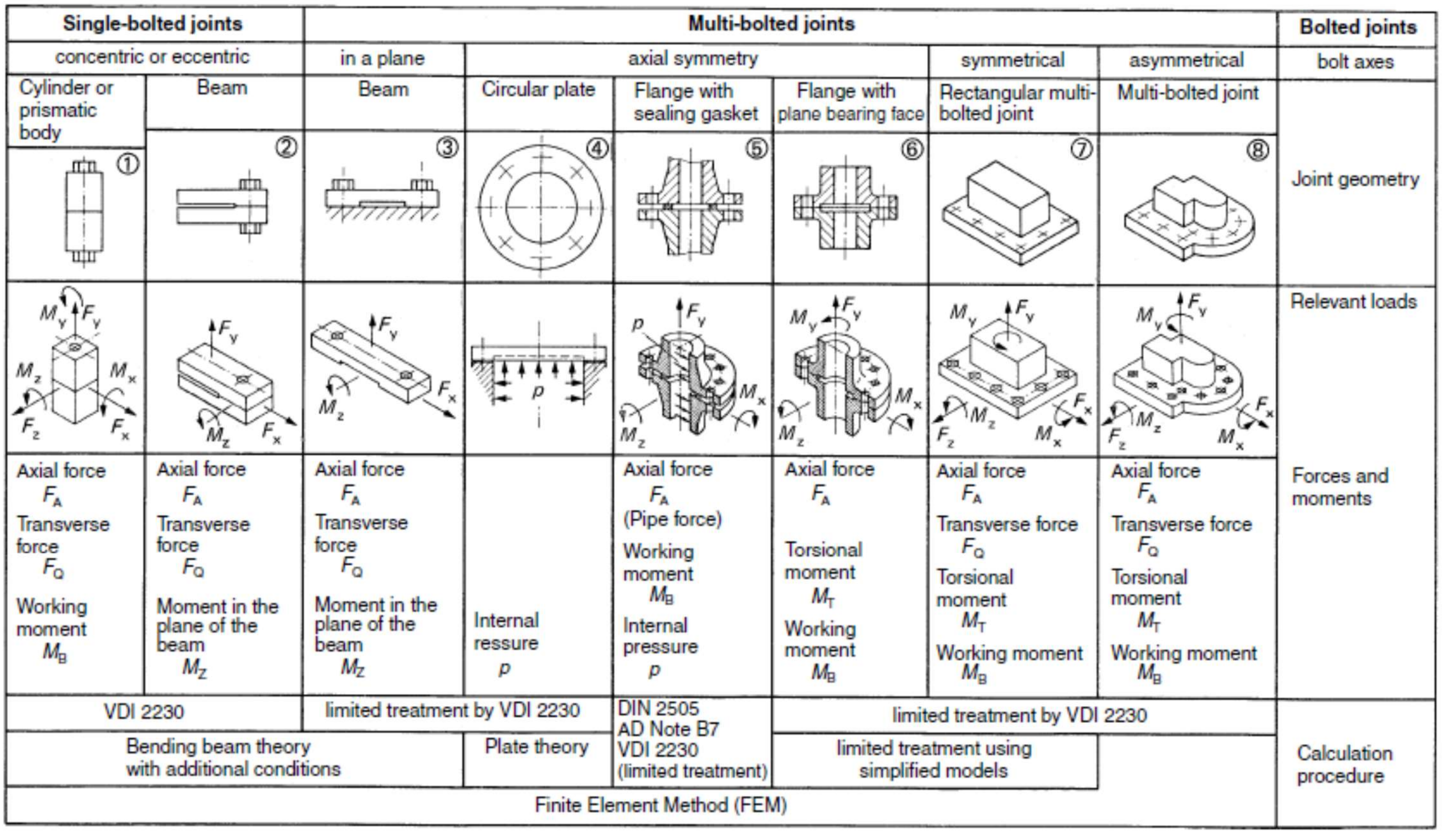
A Bolted Joint is a connection of two or more parts using one or multiple bolts as connecting element(s). The bolts must be designed to clamp these parts together on a manner that they will remain rigidly connected even under external workloads, vibration and temperature variations.

The design of a hard joint must consider the bolt dimensions as a resultant of the following factors:

- Bolted Joint's Resilience
- Pre-load loss in the joint interface and parts of the joint interface due to external workloads
- Pre-load loss due to plastic deformations
- Clamp-load dispersion during assembly processes
- Fatigue stress due to cyclical workloads
- Stress on the interfaces in contact with the bolt head and/or nut, washer and etc..

IV. Types of bolted joints:

The procedures for calculating a bolted joint will take in consideration several aspects of the joint which fundamentally depend on the joint's geometry. Multi-connected bolted joints for example, must be divided in less complex geometries in order to make it possible the calculations. All formulas used come from the different mechanisms acting in the joint due to the workloads and the complexity of the calculation procedures grows with the complexity of the joint configuration. Plastic deformations are not accounted in the calculations, except for those of microscopic magnitudes, which are accounted in order to estimate the pre-load loss during the assembling processes. From a theoretical evaluation and experimental work, we can classify different types of bolted joints as follows:



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

Figure 1

IV.1 Single-bolted Joints

Single-bolted joints are calculated based on the elastic behavior of the joint and immediate surroundings of the bolt axis. In both, assembling and under-service loads, this region has direct effect on the deformation and loading of the bolt. They can be considered as a simple mechanical spring model in which the bolt and clamped parts can be analyzed as tension and compression springs with elastic resiliencies δ_s and δ_p, as shown on the figure 2 below.

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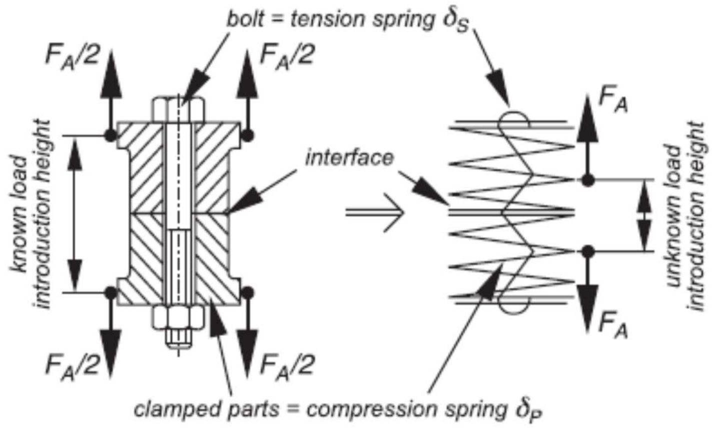

Figure 2.

A clamp-load F_K in the joint interface is a resultant of the assembly preload F_M generated during the tightening process. An external workload F_A introduced to the joint via the clamped parts is proportionally transmitted via the clamped region (interface) but also via the bolt. The amount of force acting on the bolt in addition to the bolt preload is known as F_{SA}, whereas the remaining proportion F_{PA}, relieves the clamped parts. These forces on the joint interface or on the bolt will directly depend on factors like the elastic behavior of the parts and the location of action of the external workforces. Figure 3 illustrates, i.e. on a form of a joint diagram how the position of the workforce can affect the proportionality of the forces in the interface and in the bolt itself.

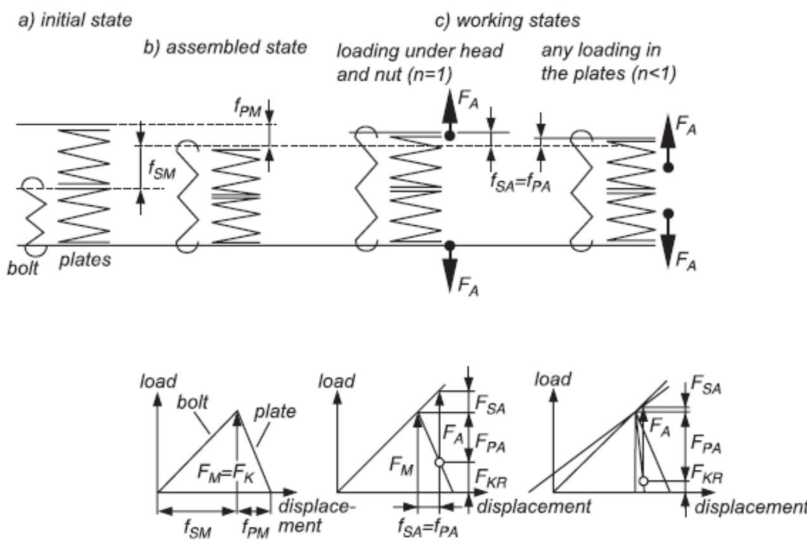

Figure 3

The single spring model however, is no longer sufficient when performing extremely extensive joint analysis, and it will not cover all possible effects of an additional workload acting in the joint. At this point, the bending resilience β_s and β_p of the bolt and clamped parts also need to be taken into account.

The Equation used to calculate the additional load F_{SA} takes into account that the bolt can be stretched by the workload F_A and moment M_B as shown in below:

$$F_{SA} = \frac{n \cdot d_P \cdot (b_P + b_S) - m_M \cdot b_P \cdot g_P}{(d_P + d_S) \cdot (b_P + b_S) - g_P^2} \cdot F_A + \frac{n_M \cdot d_P \cdot (b_P + b_S) - m \cdot b_P \cdot g_P}{(d_P + d_S) \cdot (b_P + b_S) - g_P^2} \cdot M_B$$

Equation 1

The quantities n, m, n_m and m_m take into account the effect of load or moment introduction point. The influencing factors δ, β and γ stand for displacements or skewness in account of unit loads or moments:

γ_P	*Skewness of the bolt head relative to the bolt axis on account of an imaginary additional bolt load F_{SA} = 1 N*
N	*Load introduction factor describing the effect of the working load on the displacement of the bolt head*
M	*Moment introduction factor for describing the effect of a working moment on the skewness of the bolt head. ($m = \beta_{VA} / \beta_P$)*
n_M	*Moment introduction factor for describing the effect of a working moment on te displacement of the bolt head. ($n_M = \gamma_{VA}/\delta_P$)*
m_M	*Load introduction factor for describing the effect of the working load on the skewness of the bolt head. ($m_M = \alpha_{VA} / \beta_P$)*
δ_{VA}	*Axial displacement of the bolt head on account of an imaginary working load F_A = 1 N*
β_{VA}	*Skewness of the bolt head relative to the bolt axis on account of an imaginary working moment M_B = 1 Nm*
γ_{VA}	*Axial displacement of the bolt head on account of an imaginary working moment M_B = 1 Nm*
α_{VA}	*Skewness of the bolt head relative to the bolt axis on account of an acting working load F_A = 1 N*

In this case, for a Cylinder or Prismatic body, when introducing the load factor Φ, we can re-write the previous formula as follows:

$$F_{SA} = \Phi_{en}^{*} \cdot F_A + \Phi_m^{*} \cdot \frac{M_B}{s_{sym}}$$ *Equation 2*

F_{SA}: Axial additional bolt load
Φ_{en}*: Load factor for eccentric clamping and eccentric load introduction via the clamped parts
Φ_m*: Load factor for pure moment loading M_B and concentric clamping.
S_{sym}: Distance of the bolt axis from the axis of the imaginary laterally symmetrical deformation.

In general, the remaining clamp force in the interface of the joint F_{KR} must be sufficient in order for the joint to work properly.

$$F_{KR} = F_M - F_{PA} = F_M - (F_A - F_{SA})$$ Equation 3

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On the other hand, bolted joints can also be loaded by compressive forces, which will act by increasing the FKR and reducing the amount of load between the bolt head and/or nut and the plates, which could lead to an disengagement of these parts. This is also illustrated on the joint diagram shown on Figure 3 (page 7).
The residual bearing load can be represented as follows:

$$F_{SR} = F_M + F_{SA} \quad \text{where } F_{SA} < 0$$

Equation 4

IV.1.1 Concentrically Clamped single-bolted Joint

A good example of a concentrically clamped joint is when an imaginary compression cone starting from the head of the bolt can be completely formed around the bolt body axis or when this imaginary cone is symmetrically restricted laterally in the plane of the bolt axis. In other words, during the tightening process, the reaction forces acting on the bolt head is not on an angle with axis of the bolt and the bolt head do not bent leaving the influencing factor $\Upsilon_P=0$. In both cases of a concentric or eccentric application of force, the relationship comes from a basic equation where:

$$F_{SA} = n \cdot \frac{\delta_P}{\delta_P + \delta_S} \cdot F_A$$

Equation 5

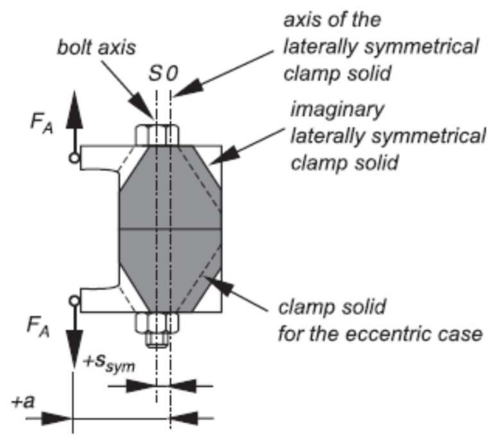

Figure 4

IV.1.2 Eccentrically Clamped single-bolted Joint

On the contrary, in the eccentrically clamped bolted joint, the bolt axis does not coincide with the symmetrical clamp solid. In this case, S_{sym} designates the distance S from the bolt axis 0 to the clamp solid axis. The parameter ɑ indicates the distance of the substitutional line of action A of the axial load F_A up to the axis of imaginary laterally symmetrical clamp solid 0. It is important to remember that ɑ must be always entered with a positive sign, so does S_{sym} (A and S are on the same side as relative to the bolt axis), or it must be then given a negative sign If they are on opposite sides. This is valid, given the assumptions of a pure application of force, forcing M_B = 0 and a flat cross section. Thus, the following formula can be used and will remain valid for small eccentricities, while other models for determining the influencing factors must be applied in case of large eccentricities of the joint.

$$F_{SA} = n \cdot \frac{\delta_P^z \cdot \left\{1 + s_{sym} \cdot a \cdot \frac{(\beta_P^z / \delta_P^z)}{\left[1 + (\beta_P^z / \beta_S)\right]}\right\}}{\delta_S + \delta_P^z \cdot \left\{1 + s_{sym}^2 \cdot \frac{(\beta_P^z / \delta_P^z)}{\left[1 + (\beta_P^z / \beta_S)\right]}\right\}} \cdot F_A$$

Equation 6

This equation takes into account the bending of the bolt which can in fact be ignored due to its high resilience. Hence the bending resilience β_P^z can be approximately determined from the moment of gyration I_{BERS}:

$$\beta_P^z \approx \frac{l_K}{E_P \cdot I_{Bers}}$$

Equation 7

The calculation of the additional bolt-load can now be determined using the following equation:

$$F_{SA} = n \cdot \frac{\delta_P^z + s_{sym} \cdot a \cdot \frac{l_K}{E_P \cdot I_{Bers}}}{\delta_S + \delta_P^z + s_{sym}^2 \cdot \frac{l_K}{E_P \cdot I_{Bers}}} \cdot F_A$$

Equation 8

IV.2 Calculation Steps

Besides some assumptions can be made in order to safety calculate a bolted joint utilizing the equations previously presented, these cannot be applied to some large scale analysis of forces and deformations to every variety of designs and components. In some cases, the application of other methods of calculation for example FEA, is necessary to achieve better results.
On a simplified but effective way, the calculation of bolted joint will be treated here in terms of the elastic deformations and forces acting in the joint, and the calculations will take into account certain boundary conditions of each joint i.e.: geometry, materials, loading, joint function, strength grade, tightening techniques and tools utilized on the assembling process.

IV.2.1 Calculation Steps on a linear basic case:

On a basic calculation of the bolt dimensions, assuming a known workload the steps are taken based on the following factors:

1. Pre-load loss due to joint relaxation F_Z;

2. Clamp-load FM reduction due to the workload proportion $F_{PA} = (1 - \Phi).F_A$;
3. Joint Function (sealing, self-loosening prevention, one-side opening prevention, minimum residual force F_{Kerf} in the joint);
4. Finally, various dispersion levels of force are considered, which will be a function of the tightening methods.

All the factors above are accounted in the equation and shown in form of joint diagram below:

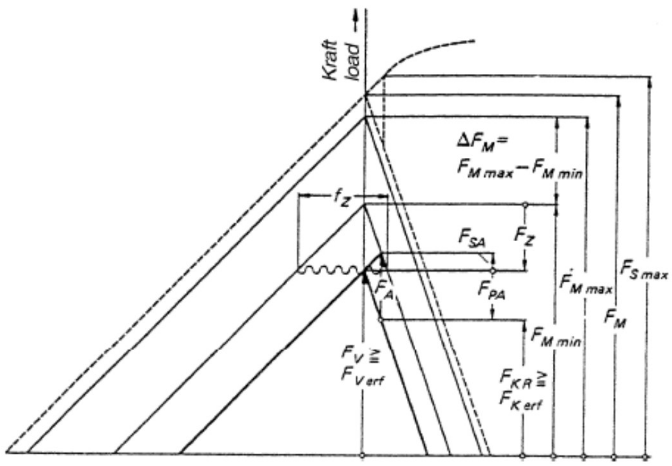

Figure 9

$$F_{Mmax} = \alpha_A.F_{Mmin} = \alpha_A.[F_{Kerf} + (1 - \Phi).F_A + F_Z] \qquad \textit{Equation 9}$$

The bolt load F_{sp} is one criteria used to calculate the bold diameter. F_{sp} is the bolt load in the multi-axial stress condition (tension and torsion) that corresponds to 90% of yield. For a given property class, the bolt will be submitted to a corresponding F_{sp} which will be the same as the necessary tension strength F_{Mmax}.

In order to determine the dispersion factor of forces α_A, we must assume a torque scattering on the order of 10%. What leads to a calculated assembling torque $M_A = 0.9.M_{sp}$.

If the tensile strength is equivalent to 90% of yield for the material, F_{SA} due to the workload F_A cannot surpass $0.1.R_{P0.2}.A_S$, hence the 0.2% of plastic deformation will be respected.

$$F_{SA} = \Phi_n.F_A \leq 0.1.R_{P0.2}.A_S \qquad \textit{Equation 10}$$

When alternated loads occur, the amplitude of load $\pm F_{SAa}$ must not exceed the fatigue resistance of the bolt. Moreover, the calculation procedure should also take into account the limiting surface pressure underneath the bolt head where the joint material must resist the surface pressure in order to avoid preload loss due to joint relaxation.

The calculations will then follow the upcoming 10 steps (R01 to R10):

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R01 – Bolt Nominal Diameter d, for the loaded portion relationship l_K/d and surface pressure underneath the bolt head and/or nut.

$$p = \frac{F_{sp}/_{0.9}}{A_P} \leq p_G \qquad \textit{Equation 11}$$

An appropriate load F_{sp} can be taken from section V of this Guide, as well as the surface pressure limits. If p_G is exceeded, the design conditions must be modified and in some cases it will be necessary the use of flat washers of adequate resistance and dimensions. In this case, l_k/d also needs revision.

R02 – Dispersion of forces (tightening factor α_A) as a function of the tightening methods and friction coefficients.

R03 – Determining the residual force F_{Kerf} to avoid one side opening of the joint interfaces. Also the friction coefficients between the clamped parts resisting to the transversal forces F_Q or a moment M_T and considerations of joints with sealing functionalities. The residual force F_{Kerf} must sustain these most critical conditions and can be calculated using the equation:

$$F_{Kerf} = \frac{(a-s).u}{K_B^2+s.u}\,.F_A \qquad \textit{Equation 12}$$

R04 – Determining the force factor Φ, bolt resilience δ_S (equation 13) and estimated load plane $n.l_K$. Determining the elastic resilience of the joints δ_P (equation 14), the factor Φ_k for concentric loaded joints (equations 15 and 16) for joints eccentrically clamped and eccentrically loaded.

$$\delta_S = \delta_K + \delta_1 + \delta_2 + \cdots \delta_{GM} \qquad \textit{Equation 13}$$

$$\delta_p = \frac{l_K}{A_{ers}\,.E_p} = \frac{f}{F} \qquad \textit{Equation 14}$$

$$\Phi_n = \eta.\Phi_k = \eta.\frac{\delta_P}{\delta_S+\delta_P} \qquad \textit{Equation 15}$$

$$\Phi_{en} = n.\frac{\delta_p^{**}}{\delta_S+\delta_P} = n.\frac{\delta_P.\left(1+\frac{a.s.A_{ers}}{I_{Bers}}\right)}{\delta_S+\delta_p.\left(1+\frac{s^2.A_{ers}}{I_{Bers}}\right)} = n.\Phi_{eK} \qquad \textit{Equation 16}$$

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R05 – Determining the pre-load loss due to joint relaxation F_Z (equation 17) with the determination of f_Z using the equation 18.

$$F_Z = \frac{f_Z}{\delta_S + \delta_p} = \frac{f_Z}{\delta_S}.(1 - \Phi_K) \qquad \textit{Equation 17}$$

$$f_Z \approx 3.29.\left(\frac{l_K}{d}\right)^{.34}.10^{-3} mm \qquad \textit{Equation 18}$$

R06 – Determining the bolt size:
In all types of tightening methods in the elastic zone, we have according to equation 9:

$$F_{Mmax} = \alpha_A.\left[F_{Kerf} + (1 - \Phi).F_A + F_Z\right]$$

According to Part V, $F_{SP} \geq F_{Mmax}$

a) At this time, the bolt clamp-load is calculated per equation 19 shown below with the clamp-load and the tightening torque dependent on the friction coefficients and obtained from the appropriate tables on Part VI of this material.

$$\sigma_M = \frac{v.R_{p0.2}}{\sqrt{1+3.\left[\frac{3}{2}.\frac{d_2}{d_0}\left(\frac{p}{\pi.d_2} + 1.155.\mu_G\right)\right]^2}} \qquad \textit{Equation 19}$$

$$M_A = 0.9.M_{sp}$$

b) If the tightening methods exceed the elastic limit we have:

$$F_{Mmin} = \left[F_{Kerf} + (1 - \Phi).F_A + F_Z\right] \qquad \textit{Equation 20}$$

Select the bolt (diameter and property class) using:

$$F_{sp}/0.9 \geq F_{Mmin} \qquad \textit{Equation 21}$$

When dispersion on the friction coefficients are disregarded, the clamp load is obtained using $\mu_G = 0.12$

R07 – Repeat the steps 4 to 6 if changes on the lk/d ratio are necessary.

R08 – The additional bolt load F_{SA} as result of the external force may not exceed the bolt admissible load. $F_{SA} = \Phi.F_A \leq 0.1\ R_{p0.2}.A_S$. For tightening methods on yield, testing will be necessary to determine the additional plastic deformation on the bolt due to F_{SA} and if it is possible to re-use the bolt.

R09 – Determining the fatigue stress in the bolt: $$\sigma_a = \Phi.\frac{F_{A0}-F_{AU}}{2.A_3} \leq \sigma_A \quad \textit{Equation 22}$$

A_3 being the cross section area of the bolt's core (bolt minor diameter).

For eccentrically clamped and eccentrically loaded joints, use equation 23 presented in the next page. Approximate values of fatigue resistance for bolts can be obtained from Kloss and Thomala equations with results represented on Figure 10 below.

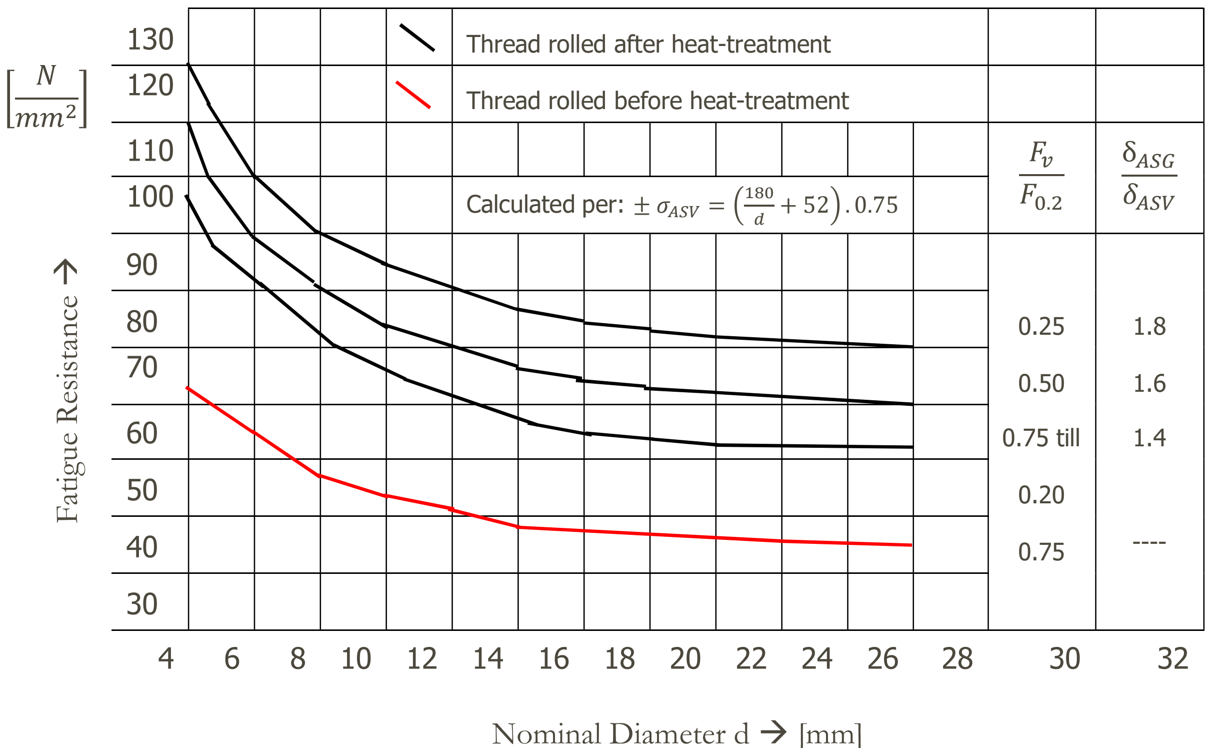


Figure 10 - Standard values of fatigue resistance for heat-treated bolts with property class 8.8, 10.9 and 12.9 according to Kloss and Thomala.

If the conditions are not satisfactory, you might need to choose a larger bolt diameter or a higher property class of for the bolt in order to increase the fatigue resistance. Notice also that bolts with threads rolled after heat-treatment have a higher fatigue resistance as compared to the ones of threads rolled before heat-treatment.

$$\sigma_{Sab} = \left[1 + \left(\frac{1}{\Phi_{en}} - \frac{s}{a}\right) . \frac{l_K}{l_{ers}} . \frac{E_S}{E_P} . \frac{a.\pi.d_3^3}{8.I_{Bers}}\right] . \frac{\Phi_{en}.F_A}{A_3}$$ *Equation 23*

R10 – Using the equation 24 below, check the surface pressure underneath the bolt head and/or nut.

$$p = \frac{F_{sp} + \Phi.F_A}{A_P} \leq p_G$$ *Equation 24*

In determining the contact area, the chamfers of the passing holes must be accounted in the calculations. In the part V of this guide, you can find a table with recommended maximum admissible values of surface pressure.

When using tightening methods that achieve or surpass Yield, values are calculated from the formula represented below as equation 25:

$$p = 1.2.\frac{F_{sp} + \Phi.F_A}{A_P} \leq p_G$$ *Equation 25*

V. Tables & References

V.1.1 Mechanical Properties of bolts per ISO 898/1

Section	Mechanical Properties		Property Class 8.8 d ≤ 16 mm	Property Class 8.8 d > 16 mm	Property Class 9.8	Property Class 10.9	Property Class 12.9
1 & 2	Tensile strength - R_m (N/mm²)	Nominal	800	800	900	1000	1200
		Minimum	800	830	900	1040	1220
3	Hardness Vickers – HV (F ≥ 98 N)	Minimum	250	255	290	320	385
		Maximum	320	335	360	380	435
4	Hardness Brinell – HB (F = 30.D²)	Minimum	238	242	276	304	366
		Maximum	304	318	342	361	414
5	Hardness Rockwell - HRC	Minimum	22	23	28	32	39
		Maximum	32	34	37	39	44
6	Surface hardness - HV 0.3		Maximum for property class 10.9 is 390 HV. Maximum difference between surface and core is 30 pts.				
7	Yield – $Rp_{0.2}$ (N/mm²)	Nominal	640	640	720	900	1080
		Minimum	640	660	720	940	1100
8	Stress – S_p/R_{eL} or $S_p/R_{p0.2}$ on proof load	Ratio	0.91	0.91	0.90	0.88	0.88
		N/mm²	580	600	650	830	970
9	Post-fracture elongation – A%	Maximum	------				
		Minimum	12	12	10	9	8
10	Tensile strength on edge testing		Must meet the same values presented on section 1 (test is not applicable for studs)				
11	Impact resistance - J		------				
		min	30	30	25	20	15
12	Hammer test resistance – Bolt head		------				
		Min	No cracks or rupture of the head				
13	Height of non-decarburized thread zone, E, mm		$\frac{1}{2} H_1$			$\frac{2}{3} H_1$	$\frac{3}{4} H_1$
	Depth of complete decarburization, G, mm		0.015				

V.1.2 Thread Tolerances per DIN 13

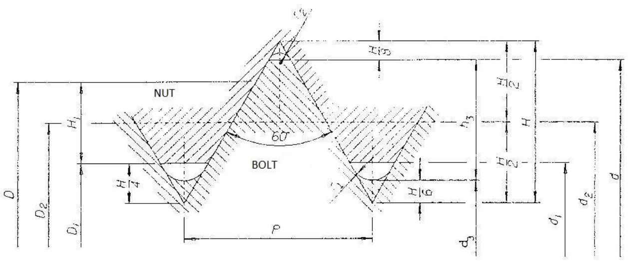
Thread		Nut Thread (mm)						Bolt Thread (mm)						
Nom. Dia.	Pitch	Class	Major Dia.	Pitch Dia.		Minor Dia.		Class	Major Dia.		Pitch Dia.		Minor Dia.	
				Min	Max	Min	Max		Max	Min	Max	Min	Max	Min
5	0.8	5H	5	4.408	4.58	4.134	4.294	4h	5	4.905	4.480	4.42	4.019	3.901
		6H			4.605		4.334	6g	4.976	4.826	4.456	4.361	3.995	3.842
		7H			4.64		4.384	8g	4.976	4.74	4.456	4.306	3.995	3.787
								6e	4.94	4.79	4.420	4.325	3.959	3.806
6	1	5H	6	5.35	5.468	4.917	5.107	4h	6	5.888	5.350	5.279	4.773	4.63
		6H			5.5		5.153	6g	5.974	5.794	5.324	5.212	4.747	4.563
		7H			5.54		5.217	8g	5.974	5.694	5.324	5.144	4.747	4.495
								6e	5.94	5.76	5.290	5.178	4.713	4.529
7	1	5H	7	6.35	6.468	5.917	6.107	4h	7	6.888	6.350	6.279	5.773	5.63
		6H			6.5		6.153	6g	6.974	6.794	6.324	6.212	5.747	5.563
		7H			6.54		6.217	8g	6.974	6.694	6.324	6.144	5.747	5.495
								6e	6.94	6.76	6.290	6.178	5.713	5.529
8	1	5H	8	7.35	7.468	6.917	7.107	4h	8	7.888	7.350	7.279	6.773	6.63
		6H			7.5		7.153	6g	7.974	7.794	7.324	7.212	6.747	6.563
		7H			7.54		7.217	8g	7.974	7.694	7.324	7.144	6.747	6.495
	1.25	5H	8	7.188	7.313	6.647	6.859	4h	8	7.868	7.188	7.113	6.466	6.301
		6H			7.348		6.912	6g	7.972	7.76	7.160	7.042	6.438	6.23
		7H			7.388		6.982	8g	7.972	7.637	7.160	6.97	6.438	6.158
		6G	8.028	7.216	7.376	6.675	6.94	6e	7.937	7.725	7.125	7.007	6.403	6.195
		7G			7.416		7.01	8e	7.937	7.602	7.125	6.935	6.403	6.123
10	1	5H	10	9.35	9.468	8.917	9.107	4h	10	9.888	9.350	9.279	8.773	8.63
		6H			9.5		9.153	6g	9.974	9.794	9.324	9.212	8.747	8.563
		7H			9.54		9.217	8g	9.974	9.694	9.324	9.144	8.747	8.495
	1.25	5H	10	9.188	9.313	8.647	8.859	4h	10	9.868	9.188	9.113	8.466	8.301
		6H			9.348		8.912	6g	9.972	9.76	9.160	9.042	8.438	8.23
		7H			9.388		8.982	8g	9.972	9.637	9.160	8.97	8.438	8.158
	1.5	5H	10	9.026	9.161	8.376	8.612	4h	10	9.85	9.026	8.941	8.16	7.967
		6H			9.206		8.676	6g	9.968	9.732	8.994	8.862	8.128	7.888
		7H			9.25		8.751	8g	9.968	9.593	8.994	8.782	8.128	7.808
		6G	10.032	9.058	9.238	8.408	8.708	6e	9.933	9.697	8.959	8.827	8.093	7.853
		7G			9.282		8.783	8e	9.933	9.558	8.959	8.747	8.093	7.773
12	1.25	5H	12	11.188	11.328	10.647	10.859	4h	12	11.868	11.188	11.103	10.466	10.291
		6H			11.368		10.912	6g	11.972	11.760	11.160	11.028	10.438	10.216
		7H			11.412		10.982	8g	11.972	11.637	11.160	10.948	10.438	10.136
	1.5	5H	12	11.026	11.176	10.376	10.612	4h	12	11.850	11.026	10.936	10.160	9.962
		6H			11.216		10.676	6g	11.968	11.732	10.994	10.854	10.128	9.880
		7H			11.262		10.751	8g	11.968	11.593	10.994	10.770	10.128	9.796
	1.75	5H	12	10.836	11.023	10.106	10.371	4h	12	11.830	10.863	10.768	9.853	9.632
		6H			11.063		10.441	6g	11.966	11.701	10.829	10.679	9.819	9.543
		7H			11.113		10.531	8g	11.966	11.541	10.829	10.593	9.819	9.457
		6G	12.034	10.897	11.097	10.140	10.475	6e	11.929	11.664	10.792	10.642	9.782	9.506
		7G			11.147		10.565	8e	11.929	11.504	10.792	10.556	9.782	9.420

Thread		Nut Thread (mm)						Bolt Thread (mm)						
Nom. Dia.	Pitch	Class	Major Dia.	Pitch Dia.		Minor Dia.		Class	Major Dia.		Pitch Dia.		Minor Dia.	
				Min	Max	Min	Max		Max	Min	Max	Min	Max	Min
14	1.5	5H	14	13.026	13.176	12.376	12.612	4h	14.0	13.850	13.026	12.936	12.160	11.962
		6H			13.216		12.676	6g	13.968	13.732	12.994	12.854	12.128	11.880
		7H			13.262		12.751	8g	13.968	13.593	12.994	12.770	12.128	11.796
	2.0	5H	14	12.701	12.871	11.835	12.135	4h	14.0	13.820	12.701	12.601	11.546	11.302
		6H			12.913		12.210	6g	13.962	12.682	12.633	12.503	11.508	11.204
		7H			12.966		12.310	8g	13.962	13.512	12.633	12.413	11.508	11.114
		6G	14.038	12.739	12.951	11.873	12.248	6e	13.929	13.649	12.630	12.470	11.475	11.171
		7G			13.004		12.248	8e	13.929	13.479	12.630	12.380	11.475	11.081
16	1.5	5H	16	15.026	15.176	14.376	14.612	4h	16.0	15.850	15.026	14.936	14.160	13.962
		6H			15.216		14.676	6g	15.968	15.732	14.994	14.854	14.128	13.880
		7H			15.262		14.751	8g	15.968	15.593	14.994	14.770	14.128	13.796
	2.0	5H	16	14.701	14.871	13.835	14.135	4h	16.0	15.820	14.701	14.601	13.546	13.302
		6H			14.913		14.210	6g	15.962	15.682	14.663	14.503	13.508	13.204
		7H			14.966		14.310	8g	15.962	15.512	14.633	14.413	13.508	13.114
		6G	16.038	14.739	14.951	13.873	14.248	6e	15.929	15.649	14.630	14.470	13.475	13.171
		7G			15.004		14.248	8e	15.929	15.479	14.630	14.380	13.475	13.081
18	1.5	5H	18	17.026	17.176	16.376	16.612	4h	18.0	17.850	17.026	16.936	16.160	15.962
		6H			17.216		16.676	6g	17.968	17.732	16.994	16.854	16.128	15.880
		7H			17.262		16.751	8g	17.968	17.593	16.994	16.770	16.128	15.796
	2.0	5H	18	16.701	16.871	15.835	16.135	4h	18.0	17.820	16.701	16.601	15.546	15.302
		6H			16.913		16.210	6g	17.962	17.682	16.663	16.503	15.508	15.204
		7H			16.966		16.310	8g	17.962	17.512	16.663	16.413	15.508	15.114
	2.5	5H	18	16.376	16.556	15.294	15.649	4h	18.0	17.788	16.376	16.270	14.933	14.647
		6H			16.600		15.744	6g	17.958	17.623	16.334	16.164	14.891	14.541
		7H			16.656		15.854	8g	17.958	17.428	16.334	16.069	14.891	14.446
		6G	18.042	16.418	16.642	15.336	15.786	6e	17.920	17.585	16.296	16.126	14.853	14.503
		7G			16.698		15.896	8e	17.920	17.390	16.296	16.031	14.853	14.408

V.1.3 Resistance Area A_S

Nominal Diameter			Preferential Series						Special Series					
Column			Standard		Fine		Extra-fine		Thread Pitch					
									2.00	1.50	1.25	1.00	0.75	0.50
#1	#2	#3	Pitch	*Area* A_S	Pitch	*Area* A_S	Pitch	*Area* A_S	Bolt's Areas in mm^2					
5			0.80	14.20										16.10
		5.5												19.90
6			1.00	20.10									22.00	
		7										28.90	31.10	
8			1.25	36.60	1.00	39.20							41.80	
		9									48.00	51.00	54.10	
10			1.5	58.00	1.25	61.20	0.75	67.90				64.50		
		11								72.10		79.50	83.30	
12			1.75	84.30	1.25	92.10	1.00	96.10		88.10				
	14		2.00	115.00	1.50	125.00	1.00	134.00			129.00			
		15					1.00	155.00		145.00				
16			2.00	157.00	1.50	167.00	1.00	178.00						
		17					1.00	203.00		191.00				
	18		2.50	192.00	1.50	216.00	1.00	229.00	204.00					
20			2.50	245.00	1.50	272.00	1.00	285.00	258.00					
	22		2.50	303.00	1.50	333.00	1.00	348.00	318.00					
24			3.00	353.00	2.00	384.00	1.50	401.00						

V.1.4 Basic Dimensions of Metric Threads per DIN 13



PITCH P	H 0.86603.P	H/4 0.21651.P	r_1 0.14434.P	H/8 0.10825.P	r_2 0.0721.P	H_1 0.54127.P	h_3 0.61343.P
0.5	0.43301	0.10825	0.07217	0.05413	0.03608	0.27063	0.30672
0.6	0.51962	0.12990	0.08660	0.06495	0.04330	0.32476	0.36806
0.7	0.60622	0.15155	0.10104	0.07578	0.05052	0.37889	0.42940
0.75	0.64952	0.16238	0.10825	0.08119	0.05413	0.40595	0.46008
0.8	0.69282	0.17321	0.11547	0.08660	0.05774	0.43301	0.49075
1	0.86603	0.21651	0.14434	0.10825	0.07217	0.54127	0.61343
1.25	1.08253	0.27063	0.18042	0.13532	0.09021	0.67658	0.76679
1.5	1.29904	0.32476	0.21651	0.16238	0.10825	0.81190	0.92015
1.75	1.51554	0.37889	0.25259	0.18944	0.12630	0.94722	1.07351
2	1.73205	0.43301	0.28868	0.21651	0.14434	1.08253	1.22687
2.5	2.16506	0.54127	0.36084	0.27063	0.18042	1.35316	1.53359
3	2.59808	0.64952	0.43301	0.32476	0.21651	1.62380	1.84030

V.1.5 Ultimate Tensile Strength for Bolts (A_S x R_m) in Newtons per ISO 898/1 (Fine threads)

Diameter	Pitch	STRESS AREA (mm^2)	PROPERTY CLASS			
			8.8	9.8	10.9	12.9
M4	0.5	9.79	7832	8811	10182	11944
M5	0.5	16.1	12880	14490	16744	19642
M6	0.75	22	17600	19800	22880	26840
M8	1	39.2	31360	35280	40768	47824
M10	1.25	61.2	48960	55080	63648	74664
M12	1.5	88.1	70480	79290	91624	107482
M14	1.5	125	100000	112500	130000	152500
M16	1.5	167	133600	150300	173680	203740
M18	2	204	169320	183600	212160	248880
M20	2	258	214140	232200	268320	314760
M22	2	318	263940	286200	330720	387960
M24	2	384	318720	345600	399360	468480

V.1.6 Ultimate Tensile Strength for Bolts (A_S x R_m) in Newtons per ISO 898/1 (Std. threads)

DIAMETER	PITCH	STRESS AREA (mm^2)	PROPERTY CLASS			
			8.8	9.8	10.9	12.9
M 4	0.7	8.78	7024	7902	9131	10712
M 5	0.8	14.2	11360	12780	14768	17324
M 6	1	20.1	16080	18090	20904	24522
M 8	1.25	36.6	29280	32940	38064	44652
M 10	1.5	58.0	46400	52200	60320	70760
M 12	1.75	84.3	67440	75870	87672	102846
M 14	2.0	115	92000	103500	119600	140300
M 16	2.0	157	125600	141300	163280	191540
M 18	2.5	192	159360	172800	199680	234240
M 20	2.5	245	203350	220500	254800	298900
M 22	2.5	303	251490	272700	315120	369660
M 24	3	353	292990	317700	367120	430660

V.1.7 Under-head/nut friction coefficients μ_K for diverse lubricating conditions

μ_K	CONTACT SURFACE				BOLT HEAD									
CONTACT SURFACE	-----------	MATERIAL			STEEL									
	MATERIAL	————	FINIHING		PHOSPHATE OR BLACK OXIDE						ZINC		CADMIUM	
		FINISHING	PROCESS		COLD FORGED			MACHINED		POLISHED AND RECTIFIED	COLD FORGED			
				DRY / LUB	DRY	LUB	MoS_2	LUB	MoS_2	LUB	DRY	LUB	DRY	LUB
JOINT CONTACT SURFACE	STEEL	Blank	POLISHED AND RECTIFIED	DRY	------	0.16 TO 0.22	------	0.10 TO 0.18	------	0.16 TO 0.22	0.10 TO 0.18	0.08 TO 0.14	0.08 TO 0.16	------
			MACHINED		0.12 TO 0.18	0.10 TO 0.18	0.08 TO 0.12	0.10 TO 0.12	0.08 TO 0.12	------	0.10 TO 0.18		0.08 TO 0.16	0.08 TO 0.14
		ZINC			0.10 TO 0.16		------	0.10 TO 0.16	------	0.10 TO 0.18	0.16 TO 0.20	0.10 TO 0.18	------	------
		CADMIUM			0.08 TO 0.16						------	------	0.12 TO 0.20	0.12 TO 0.14
	GG/GTS	UNFINISHED	POLISHED AND RECTIFIED		------	0.10 TO 0.18	------	------	------	0.10 TO 0.18			0.08 TO 0.16	------
			MACHINED		------	0.14 TO 0.20	------	0.10 TO 0.18	------	0.14 TO 0.22	0.10 TO 0.18	0.10 TO 0.16	0.08 TO 0.16	------
	Al Mg				------	0.08 TO 0.20				------	------	------	------	------

V.1.8 Thread friction coefficients μ_G for diverse lubricating conditions

μ_G	THREAD				EXTERNAL THREAD (BOLT)								
	—	MATERIAL			STEEL								
	MATERIAL	—	SURFACE FINISHING		PHOSPHATE OR OXIDE				ELECTRO ZINC		CADMIUM		ADHESIVE
THREAD		FINISHING	MANUFACTURING PROCESS		ROLLED THREAD			CUTTED THREAD	CUTTED OR ROLLED				
				COPPER	DRY	LUB	MoS_2	LUB	DRY	LUB	DRY	LUB	DRY
FEMALE THREAD (NUT)	STEEL	RAW	CUTTED	DRY	0.12 TO 0.18	0.10 TO 0.16	0.08 TO 0.12	0.10 TO 0.16	—	0.10 0.18	—	0.08 TO 0.18	0.16 TO 0.25
		ZINC			0.10 TO 0.16	—	—	—	0.12 TO 0.20	0.10 TO 0.18	—	—	0.14 0.25
		CADMIUM			0.08 TO 0.14	—	—	—	—	—	0.12 a 0.16	0.12 a 0.14	—
	GG GTS	RAW			—	0.10 TO 0.18	—	0.10 TO 0.18	—	0.10 TO 0.18	—	0.08 TO 0.16	—
	Al Mg	RAW			—	0.08 TO 0.20	—	—	—	—	—	—	—

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V.1.9 Friction coefficients μ_G and μ_K for Stainless Steel Austenitic bolts

		Lubrication		Joint Elasticity	FRICTION COEFFICIENT	
		Thread	Under-head		μ_G	μ_K
A2	A2	No lub.	No lub.	Very high	0.26 to 0.50	0.35 to 0.50
		Special lubrication of Chlorinated Paraffin			0.12 to 0.23	0.08 to 0.12
		Wax			0.26 to 0.45	0.25 to 0.35
		No lub.	No lub.	Low	0.23 to 0.35	0.12 to 0.16
		Special lubrication of Chlorinated Paraffin			0.10 to 0.16	0.08 to 0.12
	Al Mg Si	No Lub	No lub	Very high	0.32 to 0.43	0.08 to 0.11
		Special lubrication of Chlorinated Paraffin			0.28 to 0.35	0.08 to 0.11

V.1.10 Recommended Thread Engagement Length for Blind Holes

	BOLT PROPERTY CLASS			
	8.8	8.8	10.9	10.9
JOINT MATERIAL	d/p Ratio			
	< 9	> 9	< 9	≥ 9
Al Cu Mg 1 F 40	1,1.d	1,4. d		---
GG 22	1,1.d	1,2. d		1,4.d
ST 37	1,0.d	1,25. d		1,4.d
ST 50	0,9.d	1,0. d		1,2.d
C 45 V	0,8.d	0,9. d		1,0.d

V.1.11 Surface Pressure p_G in N/mm^2 in diverse materials

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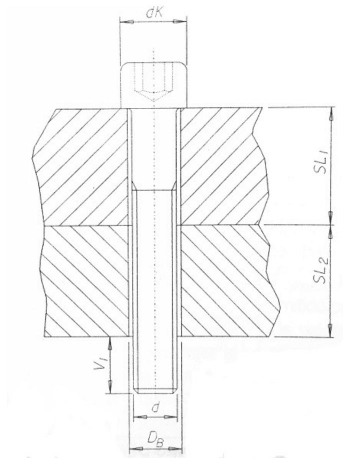
MATERIAL	STRESS LIMIT Rm	SURFACE PRESSURE STRESS LIMIT * p_G
ST 37	370	260
ST 50	500	420
C 45	800	700
42 Cr Mo 4	1000	850
30 Cr Ni Mo 8	1200	750
X 5 Cr Ni Mo 18 10	500 to 700	210
X 10 Cr Ni Mo 18 9	500 to 750	220
STAINLESS STEEL (STRUDED OR QUENCHED)	1200 to 1500	1000 to 1250
TiTAN	390 to 540	300
Ti-6 Al-4 V	1100	1000
GG 15	150	600
GG 25	250	800
GG 35	350	900
GG 40	400	1100
GG 35.5	350	900
GD Mg Al 9	300	220
GK Mg Al 9	200	140
GK Al Si 6 Cu 4	----	200
Al Zn Mg Cu 0.5	450	370
Al 99	160	140

* For engine projects, use a multiplication factor of 0.75.

V.1.12 Clearance Hole Diameters per DIN 912



Diameter	Pitch	Head Diameter	Per DIN 14			Per DIN 125		Per DIN 78
d	p	D_{Kmin}	D_B (Std.)	D_B (fine)	D_B f_{as}	d_{sch}	d_{sch}	V_{1min}
4	0.7	6.78	4.5	4.3	No Chamfer	9	0.8	4.6
5	0.8	8.28	5.5	5.3		10	1	5.6
6	1	9.78	6.6	6.4		12.5	1.6	7
8	1.25	12.73	9	8.4		17	1.6	9
10	1.5	15.73	11	10.5		21	2	11
12	1.75	17.73	14	13	16	24	2.5	13.5
14	2	20.67	16	15	18	28	2.5	15
16	2	23.67	18	17	20	30	3	17
(18)	2.5	26.67	20	19	22	34	3	20
20	2.5	29.67	22	21	24	37	3	21
(22)	2.5	32.61	24	23	26	39	3	23
24	3	35.61	26	25	28	44	4	25
(27)	3	39.61	30	---	33	50	4	28
30	3.5	44.61	33	---	36	56	4	31
(33)	3.5	49.61	36	---	39	60	5	33
36	4	53.54	39	---	42	66	5	37

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V.1.13 Tightening Factor α_A – Definition Guidelines

<table>
<tr><th>Tightening Factor α_A</th><th>Dispersion $\frac{\Delta F_M}{2.F_M}\%$</th><th>Tightening Method</th><th>Adjustment Technique</th><th colspan="2">Notes</th></tr>
<tr><td rowspan="2">1</td><td rowspan="2">± 5 to $\pm$ 12</td><td>Yield point controll (Manual or Automatic)</td><td></td><td colspan="2" rowspan="2">Clamp load dispersion is determined by the Yield stress dispersion on the assembled bolts.
In this case the bolts are designed using FMmin.</td></tr>
<tr><td>Tithening through angle control (manual or automatic)</td><td>Experimental adjustments made using torque x angle curves of the original joint.</td></tr>
<tr><td>1.2 to 1.6</td><td>± 9 to $\pm$ 23</td><td>Hydraulic bolt tensioning.</td><td>Bolt elongation or pressure monitoring</td><td colspan="2">Lower values when using long bolts and higher values when uusing short lentgh bolts.</td></tr>
<tr><td>1.4 to 1.6</td><td>± 17 to $\pm$ 23</td><td rowspan="2">Torque controll with high precision transducerized electronic torque nutrunners or electronic trasducerized torque wrenchs with dynamic torque measurement.</td><td>Experimental determination of theorical torque on original joint. I.e.: utilizing bolt elongation measurement</td><td>Lower values when values are determined through higher number of testing samples (i.e.: 20 samples). Low dispersion when using high precision torque tools.</td><td rowspan="3">Lower values for:
1. Smaller angles (hard joints)
2. Bearing surfaces with no tendency for galling
Higher values for:
1. High rotation angles (soft joints)
2. Very high surfacial hardness of the plates</td></tr>
<tr><td>1.6 to 1.8</td><td>± 23 to $\pm$ 28</td><td>Torque is determined through friction coefficients measurement</td><td>Lower values for high precision torque wrenches and electronic nutrunners. Higher values for impact, click and low precision wrentches.</td></tr>
<tr><td>1.7 to 2.5</td><td>± 26 to $\pm$ 43</td><td>Torque application using pneumatic tools</td><td>Torque tools adjusted based on estimated friction + safety factor</td><td>Lower values for clutch tools calibrated with a large sample size on the original joint</td></tr>
<tr><td>2.5 to 4.0</td><td>± 43 to $\pm$ 60</td><td>Impact wrenches</td><td>Re-tightening measurement</td><td colspan="2">Lower values when calibrated with large sample sizes, with clearance free pulsing</td></tr>
</table>

V.1.14 Fatigue – Influencing Factors

Characteristics	Fatigue Corelation	How
Reduction of bolt diameter	Increase on fatigue resistance with diameter reduction	Metalurgical properties (grain size)
Increase on residual assembling pre-load F_{Kerf}	Increase on fatigue resistance due to reduction of additional work load on the bolt. It has no direct corelation with property class of the fastener	Prevention of one-side opening of the joint interface and underneath the fastener, avoiding or reducing the cyclical additional loads on the bolt.
Material and property class	Changes on material or property class do not have significant direct influence on the fatigue resistance, unless what is caused by the pre-load increase.	The predominat notch effect suppresses any positive effect of material and property class improvements
Thread rolling after heat-treatment	Increase on fatigue resistance vary from 10 to 100% depending on the average bolt load	Residual compression stress due to manufacturing process increase the fatigue resistance specially on low pre-load assemblies. Increasing the pre-load the effect of residual compression its reduced. On bolts utilized in the plastic range, the residual compression stress is significantly reduced.
Increase on thread pitch	Limited increment on fatigue resistance	Besides a larger thread pitch has a positive effect on the radius of the thread root, this increment is also reduced due to the consequent reduction on the sectional area of the bolt. In relatively fine threads (d/p>12) and high resistance materials (≥12.9) the notch effect is dominant and in these cases the effect if increasing the thread pitch has a greater performance.

Characteristics	Fatigue Corelation	Why
Increase on thread clearance	Possible increase on fatigue resistance	With the increase on the thread clearance it is possible to obtain a higher degree of elasticity on the threads making the load distribution on the threads more evenly shared between all threads.
Increase on engagemant length	Possible increase on fatigue resistance	Increasing the engagement lentgh helps the load distribution on the threads decreasing the stress on the highly loaded 1st threads.
Decreasing the joint excentricity	Increase on fatigue resistance	Under certain conditions, it is possible to decrease the component F_{SA} with a reduction of the distance "a" and consequent reduction of the fatigue stress

VI. Tightening of Bolted Joints

VI.1 General

Assembling processes are designed to clamp two or more parts or components together which will then be used in either static or dymanic conditions, being the most comom way of assembling: the measurement of a torque value as an inderect way of "estimating" how much clamp-load is applied to the joint.

Tighetning techniques best practices will take into consideration several aspects of the joint and also the application itself (work loads) as well as variables that will affect the amount of clamp-load that is delivered and remaining in that joint after the tightening process.

Besides tighetning using torque control, there are also other techniques like: Angle control / Yield Point and Bolt Tensioning which are more complex to design and implement in production, but that will result in much higher accuracy of the achieved clamp-load. More details about each tightening techniques will be shown below

VI.1.1 Torque Control

Torque control is the method of tighetning a bolted joint with the use of a torquimeter or other pneumatic or electric tools capable of measuring or control the output torque delivered to the fastener. There are several types of tools and methods of controlling the delivered torque, and the most reliable are those which measure the torque reaction directly or indirectly in the torque tool output through the use of trasducers or strain gauges.

The total applied torque which is measured or controlled by the tool, is a resultant of the torques necessary to rotate the threads against each other and the head of the fastener being driven against the plate underneath the fastener. This relationship can be described on the following equations:

$$M_A = M_{GA} + M_K$$ *Equation 26*

$$M_A = F_M.\left[\frac{d_2}{2}.\tan(\varphi + p') + \frac{D_{Km}}{2}.\mu_K\right]$$ *Equation 27*

For 60 degree threads, we can simplify the equation as shown below as:

$$M_A = F_M.\left[0.16.p + \mu_G.0.58.d_2 + \frac{D_{Km}}{2}.\mu_K\right]$$ where: $$D_{Km} = \frac{2}{3}.\left(\frac{d_k^3 - d_B^3}{d_k^2 - d_B^2}\right)$$

Equation 28 *Equation 29*

Therefore, from these equations we can determine the assembling torques as a result of a clamp-load which must be determined considering 90% of utilization of the fastener strength's. $F_{SP} = \sigma_M.A_S$

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From equation 19 we can obtain σ_M for a bi-axial state of tensions (tensile & torsion). The clamp load F_{SP} can be determined through a torsional momemtum extremely variable depending on the property class of the fastener and also the variations of friction coefficients.
The torque / clamp-load ratio is subjected to all possible variations of thread and head friction coefficients what could lead to a large variation of the clamp-load even within bolts and nuts from the same production lots. This makes it difficult the task of building a table of correlation between torque and clamp-load using the equation 28, due to the large number of possible combinations of friction coefficients.
Moreover, in the cases of assembling processes per torque control utilizing a torque wrench for example, the total uncertainty will come from the following partial uncertainties:

a. Uncertainty of the correct friction coefficients on the threads and underneath the head;
b. Uncertainty of the tool;
c. Uncertainty of the fasteners geometry (small variation on the friction radius will affect the friction force that is reacting against the torque tool);

Testing and calibrating the tool with the original joint and with a larger number of samples will help understand and reduce some of these and other uncertainties, promoting a better (lower) tightening factor and lower dispersion of the generated clamp-loads. Some recommendations can be found on table V.1.13 of this training material.

VI.1.2 Angle Control

This method is an indirect way of using the joint resilience and achieving a desired clamp-load through the elongation of the bolt and compression of the joint. I.e. and to simplify this theory, imagine the bolt was the only element changing it's length during the assembling process and it has a known Young Modulus. Knowing the bolt pitch and assuming no variations on it, we could say the bolt will stretch by the same amount of its pitch for every 360° of revolution and the force generated could be directly calculated multiplying that value x the Young Modulus of the bolt and its sectional area. In reality, we cannot only use the bolt characteristics but we have also to use the joint's elongation which is negative and consequantly will make the bolt elongation smaller than its pitch for each 360° of revolution. Explaining this fenomena in form of equation, we would have:

$$\varphi = \frac{360}{p}.\left(\delta_S + \delta_p\right).F_M \qquad \textit{Equation 30}$$

Also, in order to use this tightening technique, we have to assume a constant deformation of the joint from start to end of the angle control process, which will only be true after some clamp-load has been already generated and we have achieved the snug torque in that joint. This of course will bring some uncertaninty to the final generated clamp-load, but much lower than those on a torque control tighthening process, in special if we use the angle control technique in the plastic zone of the joint, where the curve turns horizontal at a specific clamp-load even if you continue to rotate the fastener until before it starts to snap. Assuming the joint reaches its yield point, the tightening factor $\propto_A$ can be considered as 1 for the matter of fastener specification.

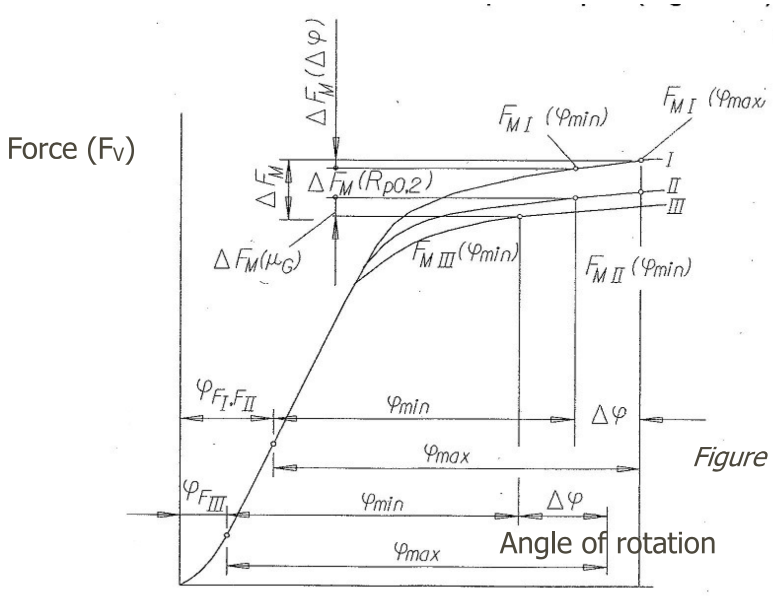


Figure 11

VI.1.3 Yield Point

On the Yield Point tightening technique, an electronic tighening system is used to measure and monitor the variations of the torque x angle during the assembling process and uses a certain window or $\Delta M_A/\Delta\varphi$ to determine when to start and to stop the tightening process. Due to variations on the $\Delta M_A/\Delta\varphi$ rationing along with the full torque x angle curve, the system must be thaught the start and end points which are respectively: after snug torque and after the gradient $\Delta M_A/\Delta\varphi$ falls to a certain percentage of the maximum gradient of the joint. It is important to determine all these gradients in order to prevent the system to start or shut-off outside the correct window.

When correctly utilizing this technique, the generated clamp-load in the application is almost independent on the friction coefficients and the tighening factor can be considered as 1.

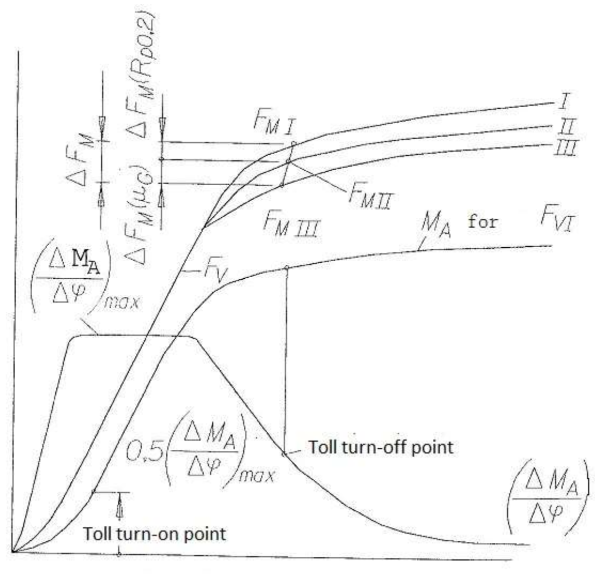


Figure 12

VI.1.4 Bolt Tensioning

The bolt tensioning technique is fairly easy to understand and it is highly precise in regards to the achievement of a specific clamp load. The only factors influencing the achieved clamp load are the small variations on the bolts resilience and the variations on the oil pressure of the device used to tension the bolt.

In this technique, there is no influence of the multi-axial stress (torsion & tension), as the bolt is tensioned without spining against the female thread. This also allows to use of the bolt on it's higher tension capacity as in here, the equation 19 is not applied due to the inexistence of thread friction during the assembling process.

The procedure is done in 4 basic steps and certain parameters must be respected i.e.: the lentgh of the exposed threads before the assemblying must be at least 1 x D.

1. The joint is assembled with a low pre-torque to allow the exposure of the threads for the insert of the tensioning to be threaded to it.
2. The tensioning device is positioned around the bolt and the threaded insert is assembled against the bolt.
3. Hydraulic pressure (pre-determined) is applied pushing the insert out of its housing which will pull and stretch the bolt to its final tension;
4. The nut is rotated to sit plane on its interface and the tension is released.

Obviously, in this tightening method, the tightening factor will be aamongst the lowest, in certain cases being close to 1, depending on the accuracy of the equipment used in the process.

VI.2 Influencing factors in the assembling process

In addition to the theorical considerations earlier discussed, the proper design of bolted joints which is strongly influenced by the joint parts and components, is also dependent on several other factors like: material, hardness, bolt profile, nut design and etc. also shown in previous sections. Other factors like: Tightening factor/ bolt & nut engagement, joint relaxation and bolt loosening will be shown here in more details.

VI.2.1 Tighthening Factor

As shown in the table V.1.13, the tightening techniques directly affect the dispersion of the generated clamp-load during the assembling process. This influence has its roots on several characteristics of the bolted joints such as:

1. Coefficients of friction between the parts in movement;
2. Geometric characteristics of the joint;
3. Tools used in the assembling process
4. Etc.

Errors in the friction coefficients estimation, dispersion of the friction coefficients within the same lot of fasteners and/or clamped parts in contact with the fastener being torqued, geometric variations of the friction radius, as well as the precision of the torque tools being used in the assembling process will lead to variations on the achieved clamp-load. These factors will lead to an over designing of the fastening elements on a way that the fastener will not fail or yield during the assembling process when the maximum pre-load is achieved as a resultant of very low friction coefficients and/or other factors.

This tightening factor is known as α_A which is the ratio F_{Mmax}/F_{Mmin}.

VI.2.2 Bolt/Nut engagement

In a bolted joint, the fastening elements are designed not to fail in the engaged region. In other words, any cracks, plastic deformation and other failure modes must happen outside of the engaged area and this is only achieved with a proper engagement lentgh of the male and female threads.

In calculating the minimum engagement lentgh, we take into consideration the nut thickness M_{FR}. The critical engagement lentgh, is the one where neither the bolt's or the nut's threads will shear and failure before the bolt fails on its cross section area. Several factors in which some interact with one another will influence the correct design of the threads engagement i.e.: thread type, pitch, thread tolerances, thread diameter, nut across flats dimension, bolt's passing hole, resilience and ductibility of the fasteners, types of stress, friction coefficients and the amount of assemblies.

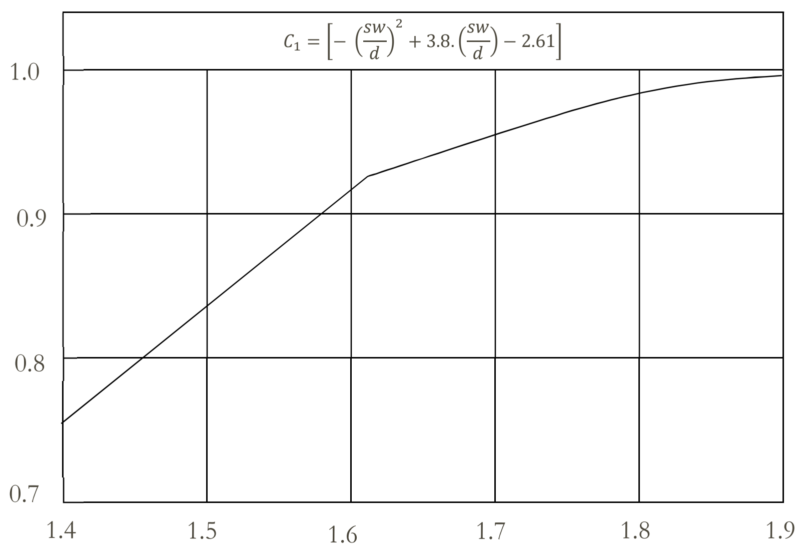
If the engagement lentgh is less than the critical value m_{kr}, either the bolt or the nut will shear on its threads.

E. A. Alexandre (Analysis and Design of Threaded Assemblies – Intl. – Automotive Congress and Exposition, Detroit 1977) developed a mathematical model to calculate the possible tension loading in bolted joints. This model predicts the failure type on a overloading situation, i.e.: shearing or the threads of the bolt or the nut.

The effective engagement lentgh m_{eff} is calculated from the difference between the total thickness of the nut m_{ges} and the internal chamfer in both surface sides of the nut. It is assumed that chamfered areas can only support 40% of the load capacity as compared to a full thread, which leads to the folowing equation to determine the effective engagement length:

$$m_{eff} = m_{ges} - (D_C - D_1).\tan 45\,.(1 - 0.4) \qquad \textit{Equation 31}$$

Where D_1 is the minor diameter of the nut and D_C the internal diameter of the surface contact of the nut.



The nut expansion caused by the radial component of the clamp-load will reduce the area of contact between the pitches of the nut and bolt and consequantly reduce the area resisting to shear. The decrease in the ability to sustain load will be determined by a factor C_1 shown in the previous figure and the explained through the following equations:

$$C_1 = \left[-\left(\frac{sw}{d}\right)^2 + 3.8.\left(\frac{sw}{d}\right) - 2.61\right] \qquad \textit{Equation 32}$$

When $1.4 \ll \frac{sw}{d} < 1.9$ the factor $R_S = R_M.A_{SG} / R_M.A_{SC}$ exemplifies the bolt nut force relationship, shearing of the bolt and nut threads and the sum of plastic deformation between the threads.
The degree of reduction on the resistance to shearing due to the expansion fenomena can be identified by the C2 and C3 factors shown on figure 14 below.

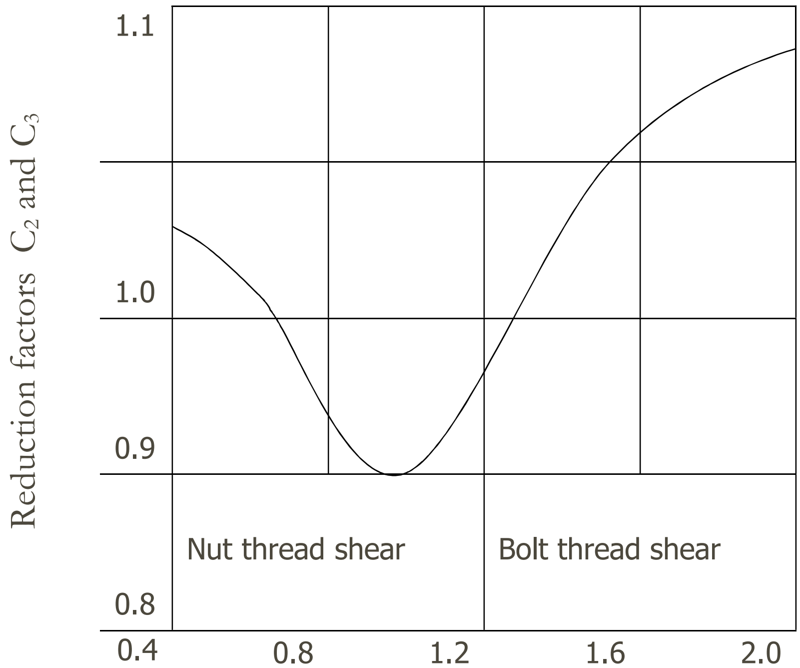


Figure 14 shows the reduction factor on the shearing resistance of the threads due to the plastic deformation on the threads

C_2 and C_3 factors are calculated by the folowing equations and conditions:

$$C_2 = 5.594 - 13.682.R_s + 14.107.R_S^2 - 6.057.R_S^3 + 0.9353.R_S^4 \qquad \text{for } 1 < R_S < 2.2$$

$$C_2 = 0.897 \qquad \text{for } R_S \leq 1$$

$$C_3 = 0.728 + 1.769.R_S - 2.896.R_S^2 + 1.296.R_S^3 \qquad \text{for } 0.4 < R_S < 1$$

$$C_3 = 0.897 \qquad \text{for } R \geq 1$$ *Equations 33 and 34*

The equantions developed to calculate the shearing forces are only valid on the axial load and cannot be used in the combined state of tensions.

The shearing resistance on the bolt's thread can be expressed per:

$$F_{max}(bolt) = R_M(bolt).A_{SG}(bolt).C_1.C_2.0.6$$ *Equation 35*

$$A_{SG}(bolt) = \frac{m_{eff} - l_B}{P}.\pi.D_1.\left[\frac{P}{2} + (d_2 - D_1).\frac{1}{\sqrt{3}}\right] + \frac{l_B}{P}.\pi.D_m.\left[\frac{P}{2} + (d_2 - D_m).\frac{1}{\sqrt{3}}\right]$$ *Equation 36*

l_B is the length of the chamfer of the nut ($\approx 0.4.m_{gers}$)

D_m is the average diameter of the nut's chamfer ($\approx 1.015.D_1$)

$$\frac{\tau_B}{R_M} = 0.6$$

The shearing resistance on the nut's threads can be expressed per:

$$F_{max}(nut) = R_M(nut).A_{SG}(nut).C_1.C_3.0.6$$ *Equation 37*

$$A_{SG}(nut) = \frac{m_{eff}}{P}.\pi.d.\left[\frac{P}{2} + (d - D_2).\frac{1}{\sqrt{3}}\right]$$ *Equation 38*

D_2 being the nut thread pitch diameter

The dinamic friction coefficient due to the relative movement between the threads in contact, reduces the thread resistance to shear under axial loads as compared to its resistance on the pure axial condition. This reduction will be on the order of 10% to 15%. Despite, the bolt's cross section resistance on the on the engaged portion of the threads is also reduced due to the multi-axial loading of the fastener but this tends to be at least 5% less than the reduction experimented in the non engaged portion of the bolt. This can shift the weaker point of the bolt during the assembling process from the threads to the portion of the bolt that is not engaged.
The minimum nut thickness necessary to shift this weaker point and avoid the threads to shear during assembling can be expressed as follows:

$$F_{max}(bolt) < F_{max}(thread, nut) < F_{max}(thread, bolt)$$

$$R_M(bolt).A_S < R_M(bolt).A_{SG}(bolt).C_1.C_2.0.6$$

$$< R_M(Nut).A_{SG}(Nut).C_1.C_3.0.6$$

Equation 39

VI.2.3 Joint Relaxation

In addition to the elastic deformation in the bolted joints, relaxation also play a role in the clamp-load loss, in special those caused by irregularity on the surfaces of the plates and planicity errors. The amount of relaxation f_Z translates into reduction of the amount of elastic elongation of the bolt and consequantly loss of pre-load.

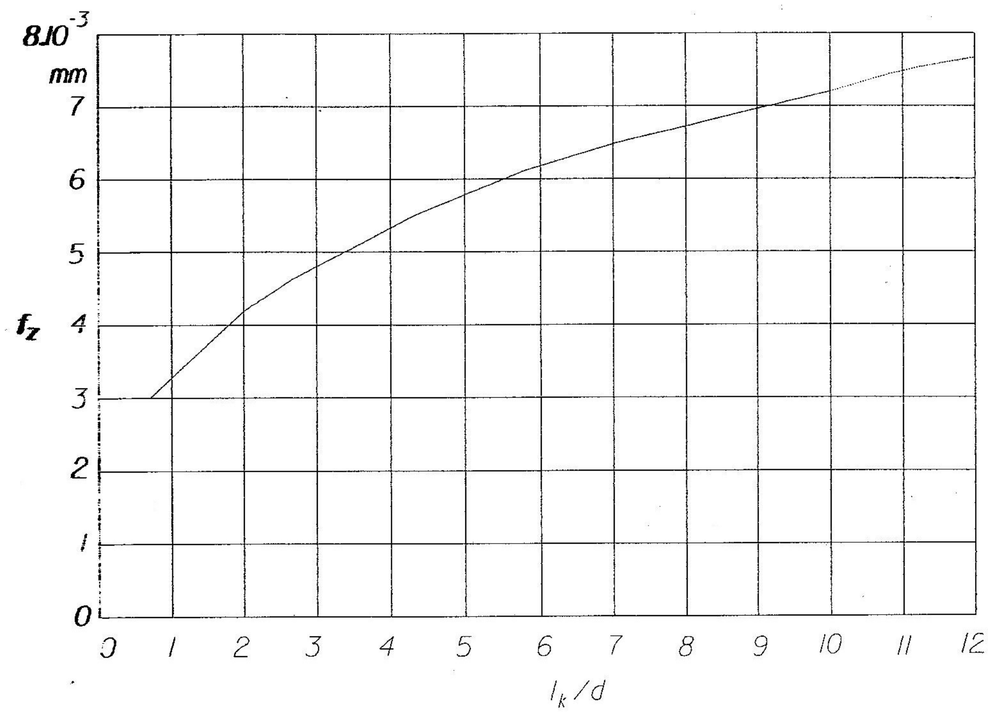


Figure 15 – Guiding values for relaxation loss in bolted joints.

As shown in figure 15, the relationship between pre-load and relaxation derives from the triangles similarities.

$$\frac{F_Z}{F_M} = \frac{f_Z}{f_{SM} + f_{PM}} = \frac{f_Z}{\delta_S . F_M + \delta_P . F_M}$$

$$F_Z = \frac{f_Z}{\delta_S + \delta_P} = \frac{f_Z}{\delta_S} . (1 - \Phi_K) \qquad \textit{Equation 40}$$

$$F_Z = \frac{f_Z . \Phi_K}{\delta_P}$$

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The sum of the relaxation f_Z is generally less than expected as a fuction of material roughness of the interface of the joint as not all the surface is sufficiently pressured to cause deformation in all peaks and valleys.

From experimental results, the amount of relaxation is not strongly dependent on the amount of interfaces and roughness, however, it is highly affected by engagement length of the joint l_K. For a hard joint (not a stack of thin plates), with the bolts in accordance with ISO 4014, we have:

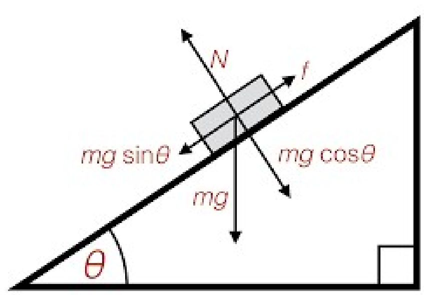
$$f_z = 3.29.\left(\frac{l_k}{d}\right)^{0.34} .10^{-3}mm$$

VI.2.4 Self-loosening of Bolted Joints

Immediately after a bolted joint its assembled to its specified torque, the preload on that joint may change relative to the achieved preload during assembling. The changes will be affected by several factors i.e.: material relaxation, rotation of the bolt, changes in temperature, overloading etc..

Although there are formulas to calculate an estimated amount of preload loss due to certain pre-determined conditions, it is always advised that the amount of preload loss is measured experimentaly and under different conditions i.e.: after assembling, after field testing, after submitting the joint to working temperatures and etc. Besides preload losses due to embedding of contact surfaces, relaxation, temperature changes and several other common known factors, one of the most concerning ones is vibration. It is a common sense that vibration can cause fasteners to come loose, but how does that work?

As previously discussed, during the design phase of each bolted joint, a certain amount of residual clamp-load is expected and it is used in the calculations to determine: bolt dimension, torque specifications, processes and etc. Assuming all these were carefuly checked, we should not experience vibration loosening.
This can be explained using the laws of motion as shown in the schematic representation below where the block will never move while the friction forces and weight are in balance.



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Translating this analogy to a bolted joint connection, the block and the ramp are the threads in contact and the friction force, directly dependent on the clamp-load in that joint. In other words, the higher the remaining clamp-load in a certain joint, the less likely that joint will self loose by vibration.

Finally, the vibration resistance of a given bolted joint, can be determined by the amount of lateral displacement necessary to break loose that union, which is can be expressed by the following formula:

$$S = \frac{F_v.\mu_K.l_K{}^3}{12.E.I} \qquad \textit{Equation 50}$$

VI.2.5 How to Prevent Self-loosening

Through an adequated design, based on the variables presented in the equation 50, we can improve the vibration resistance of bolted joints by increasing the amount of displacement necessary to promote the selft loosening phenomena.
This goal can be achieved by increasing or decreasing the absolute values of some of the variables on the right side of the equation, i.e.: Clamp-load, Head Friction, engagement length, etc..

The first and most effective way to achieve this task, without necessarily making geometric and material changes in the joint, is to guarantee the use of the fasteners on its highest level of clamp-load. Challenges with using this method, usually come from variations on the joint elements, in special on the: friction coefficients and assembling tools and strategies used in the process, and a closer assessment of each individual bolted joint as opposed to use "standard" designing methodologies is suggested in order to make this approach more effective.

Other methods to achieve a higher resistance level to vibration loosening, is the use of true locking elements in the joint, on a way that the lateral displacement of the fasteners will be reduced or avoided. I.e.: the use of chemical and/or mechanical locking elements.

Close attention must be given to the use of these methods, as the Fastener Engineer needs to be attent to the difference between holding the fastener elements in the application and holding them tight in the application. Certain locking elements will play well the function of avoiding the bolts/nuts to deatach from the joint, however, if they continue attached to the joint at a low level of clamp-load, fatigue can occur and create a much bigger problem than the self-loosening itself.

It is recommended that all new joints are tested on its ability to generate and retain clamp-load before the parts and components are released. These studies must be conducted in the design, prototype and exerimentation phases of any project and again, confirmed to be valid in the production phase when assembling methods and parts, tend to be quite different than those used in the lab environments. Testing will vary from joint to joint and will be selected in accordance with the objective of each application (sealing, pressure, hi-temperature environments etc.). The most common testing to be performed are:

1. Torque x Tension: …………………… ISO 16047 (Fastener Coefficients of Friction);
2. Junker (Vibration): …………………… DIN 25201 / 65151;
3. Torque x Angle: …………………… Analysis of joint's Torque x Angle Curves;
4. Residual Calmp-load: ……………….. Analysis of joint relaxation & other factors

VII. Symbols and Notation

δ_s……… Bolt's resilience	δ_p…..... Plate's resilience
F_A……… Workload	F_{SA}…... Additional Bolt Workload
β_s……… Bolt's bending resilience	β_p…… Plate's bending resilience
F_{KR}……. Residual Force (interface)	F_{Kerf}… Residual Clamp-load
F_Z……… Pre-load loss	α_A…… Assy clamp-load dispersion
a……… Distance to line of force	s……... Distance to joint's center of gravity
u……… Edge dist. to opening point	K_B…… Inertia radius
M_A……. Assy torque	σ_a…… Alternating tension (bolt)
M_{GA}…… Torque on the threads	M_K…… Torque on the head/nut
μ_G……... Thread friction coefficient	μ_K …… Under head friction coefficient
D_{KM}……Head friction diameter	Φ …… Force factor
η……… Yield factor	R_M…… Ultimate bolt stress

BIBLIOGRAPHY

Source: ……………………………………………………………. Description

VDI 2230 …………………………………………………………… Systematic Calculation of High Duty Bolted Joints

Rubens Cioto ……………………………………………………………………….. Bolted Joints Design and Calculation

G. H. Junker, Koblens – RFA ……………………………… Principle of the Calculation of High Duty Bolted Joints

John H. Bickford ………………………………………… An introduction to the design and behavior of bolted Joints

Oakland University ……………………………………………… A Proposed Model for Creep Relaxation of Soft Gaskets in Bolted Joints at Room Temperature

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PCL TORK TECHNOLOGIES: www.pcltork.com

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