Bolt Calculation according to VDI 2230

Theory of VDI2230 Part 1 (partially Part 2)

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Schedule (online)

Day 1	
14:00	Welcome to the training
14:15 – 15:45	Introduction, Scope, Calculation Model, Distortion Triangles
15:45 – 16:00	Break
16:00 – 18:00	Tightening Factor, Resiliiences, Dividing the Working Load
18:00	End of Live Stream Day 1
After Training	Independent Work on the Exercises (approx. 2 h)

Day 2	
14:00 – 15:00	Discussion of the Exercises, Questions & Answers
15:00 – 16:15	Eccentric Clamping and Loading
16:15 – 16:30	Break
16:30 – 18:00	Stress/Strength, VDI Part 2, Calculation with FE Results
18:00	End of Training

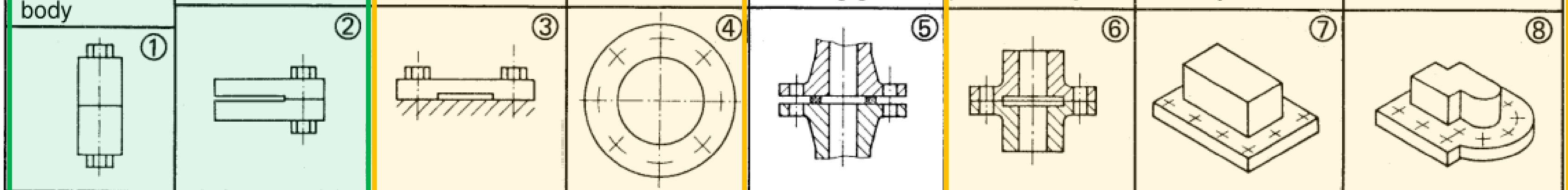
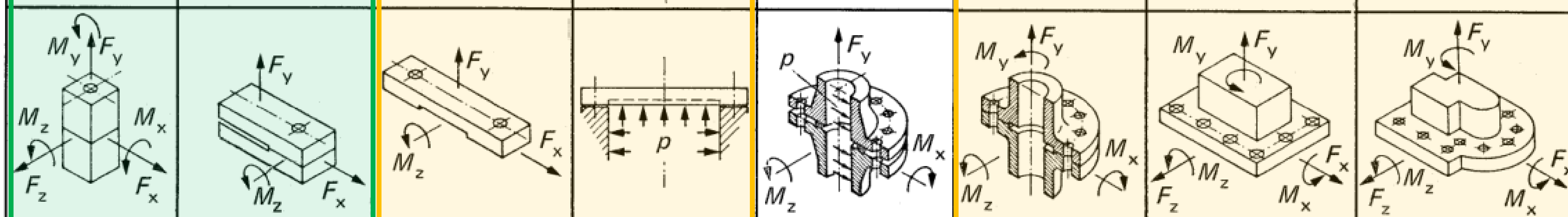
Overview

1. Scope of VDI2230 Part 1
2. Basic concepts of VDI2230 Part 1
3. Calculation steps of VDI2230 Part 1
4. Short introduction to VDI2230 Part 2

- Included:
  - Steel, 60° flank angle
  - strength grades 8.8 to 12.9
  - frictional transmission of working load (through clamped parts)
  - static or dynamic axial force, bending moments and transverse forces
  - limited size of contact areas at inner interfaces (G)

- Not included:
  - Extreme stresses (e.g. corrosion)
  - sudden and stochastic loads
  - determination of external loading

Scope VDI2230

Single-bolted joints		Multi-bolted joints						Bolted joints
concentric or eccentric		in a plane		axial symmetry		symmetrical	asymmetrical	bolt axes
cylinder or prismatic body	beam	beam	circular plate	flange with sealing gasket	flange with plane bearing face	rectangular multi-bolted joint	multi-bolted joint	
① 	②	③	④	⑤	⑥	⑦	⑧	joint geometry
								relevant loads
axial force F_A transverse force F_Q working moment M_B	axial force F_A transverse force F_Q moment in the plane of the beam M_Z	axial force F_A transverse force F_Q moment in the plane of the beam M_Z	internal ressure p	axial force F_A (pipe force) working moment M_B internal pressure p	axial force F_A torsional moment M_T working moment M_B	axial force F_A transverse force F_Q torsional moment M_T working moment M_B	axial force F_A transverse force F_Q torsional moment M_T working moment M_B	forces and moments
VDI 2230		limited treatment by VDI 2230		DIN EN 1591 AD 2000 Note B7	limited treatment by VDI 2230			calculation procedure
bending beam theory with additional conditions			plate theory		limited treatment using simplified models			
finite element method (FEM)								

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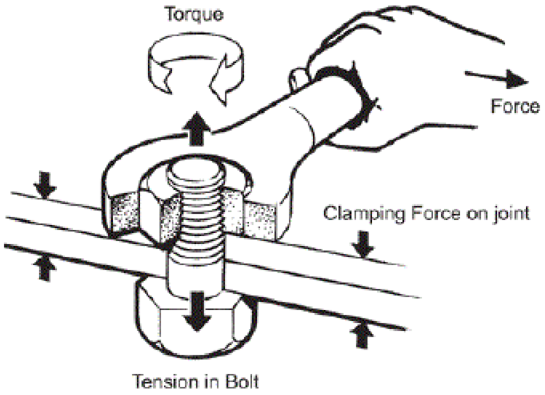
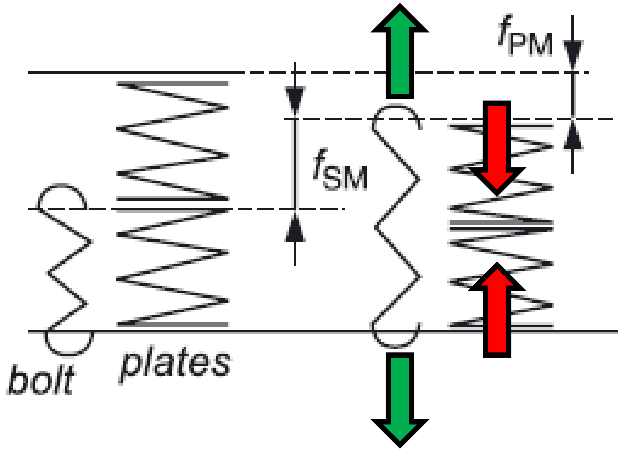
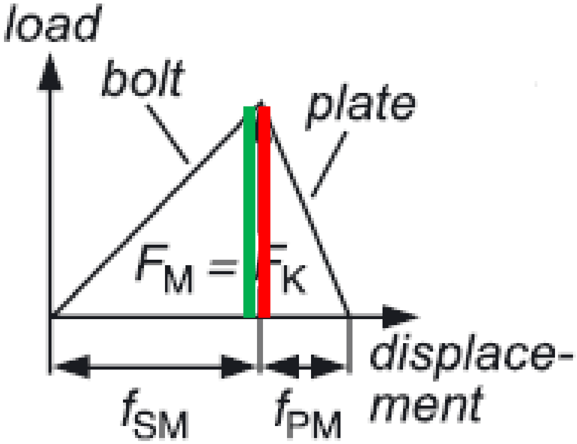
Assembly state: „**internal load**“

During assembly an axial preload force $\mathbf{F_M}$ is generated. This force is present as a **tensile force in the bolt (Fs)** and as a **compressive force in the joint ($\mathbf{F_K}$)**.







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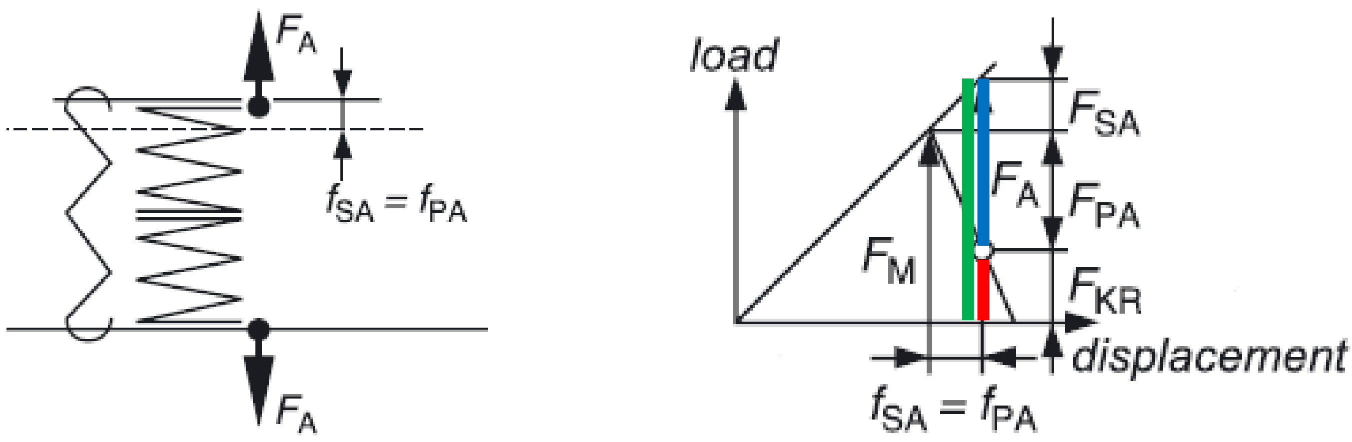
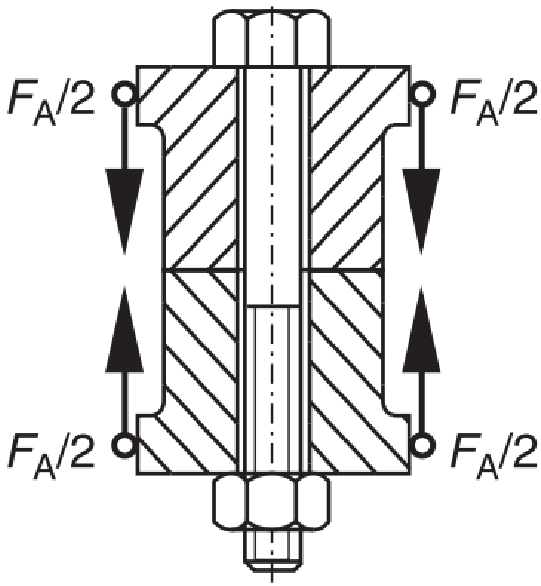
Working state: **„external load, F_A >0“**

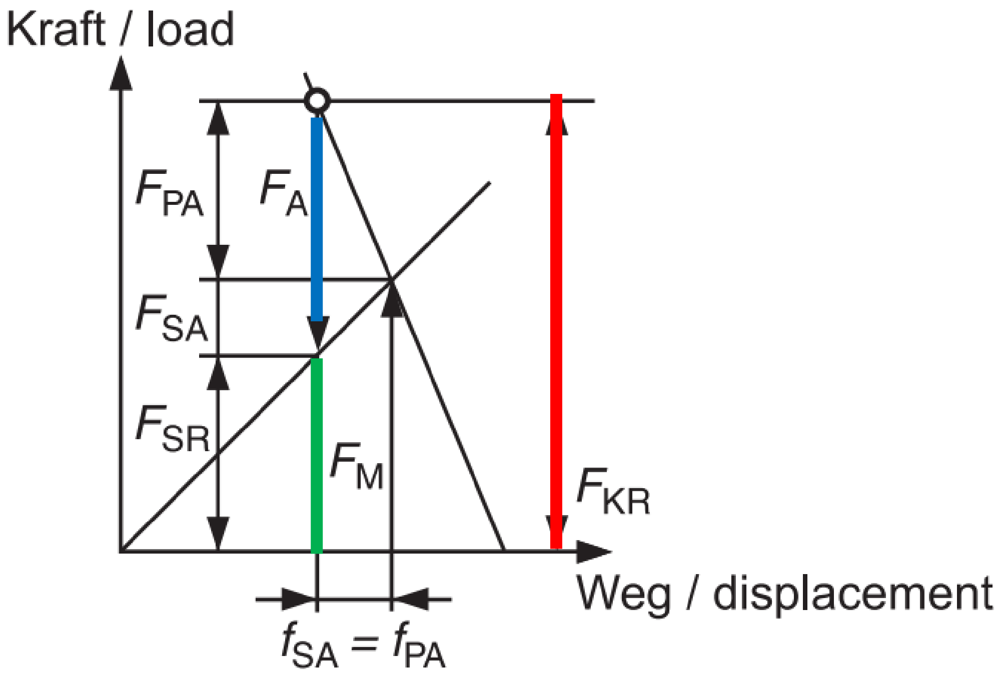
Introducing the axial working load **F_A** the bolt is additionally loaded with the additional bolt load **F_{SA}** which results in the total **bolt load F_S**. The plates are relieved by the amount **F_{PA}** which results in a **residual clamp load F_{KR}**.



Special working state: **„external load, F_A >0“**

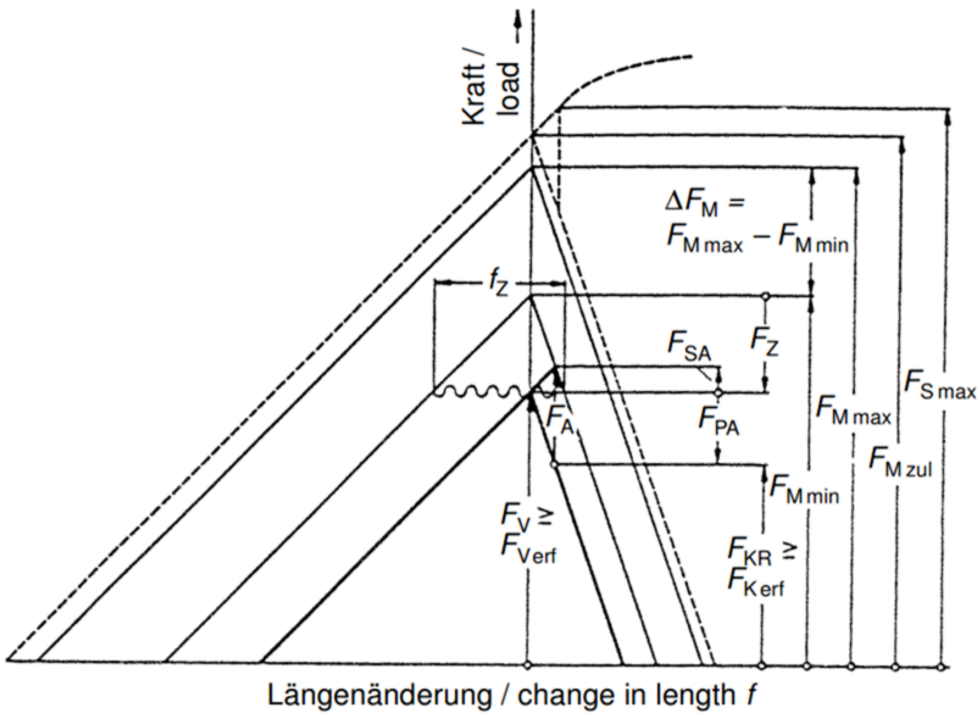
The clamping force increases from F_M to **F_{KR}**. Relieving the bolt with **F_{SA}** results in a reduction of the **bolt force** to **F_{SR}**.



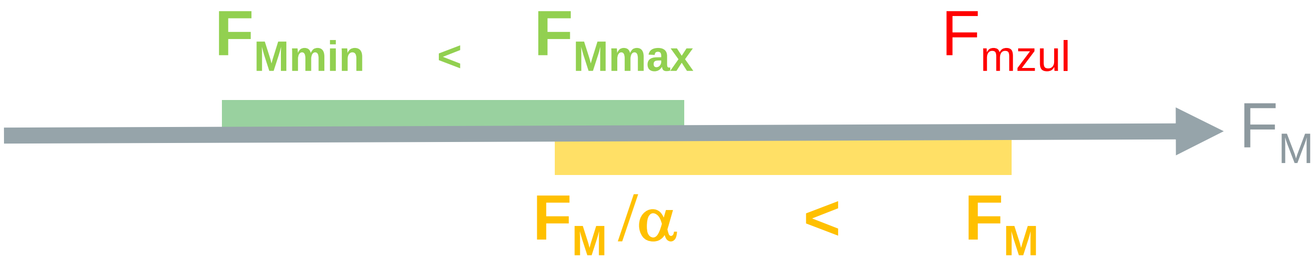


$$F_{\mathrm{M\,max}} = \alpha_{\mathrm{A}} \cdot F_{\mathrm{M\,min}} \qquad (16)$$
$$= \alpha_{\mathrm{A}} \left[F_{\mathrm{Kerf}} + (1 - \Phi)\, F_{\mathrm{A}} + F_{\mathrm{Z}} + \Delta F_{\mathrm{Vth}}\right]$$

F_Z Loss of preload due to embedding
ΔF_{Vth} thermal. loss of preload



The mounting force F_M needs to be
bigger than the maximum required assembly preload F_{Mmax}
and smaller than the **permissible assembly preload** F_{mzul}



F_{Mmax} inlcudes:

- Required clamp load
- Loss of preload
- Working load and variation in assembling (α)

The **VDI 2230** names this force “max. assembly preload”

F_{Mzul} includes:

- Yield strength ← Your choice of bolt
- Utilization ← Your choice of tightening
- Friction

This load is also named “permitted pretension force”, meaning the (max.) achieved preload using the tabular value of tightening torque

In KISSsoft, the calculation is run with F_M = F_{Mzul}

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Basic concepts of VDI 2230

Inputs

- R0 nominal diameter, limiting measurements
- R1 tightening factor
- R2 minimum clamp load

Distortion triangle

- R3 dividing the working load, load factor
- R4 preload changes
- R5 min. assembly preload
- R6 max. assembly preload

Stress cases & strength verifications

- R7 assembly stress
- R8 working stress
- R9 alternating stress
- R10 surface pressure
- R11 minimum length of engagement
- R12 slipping, shearing
- R13 tightening torque

R0 nominal diameter and limiting size G

The bolt nominal diameter is roughly determined according to Table A7.

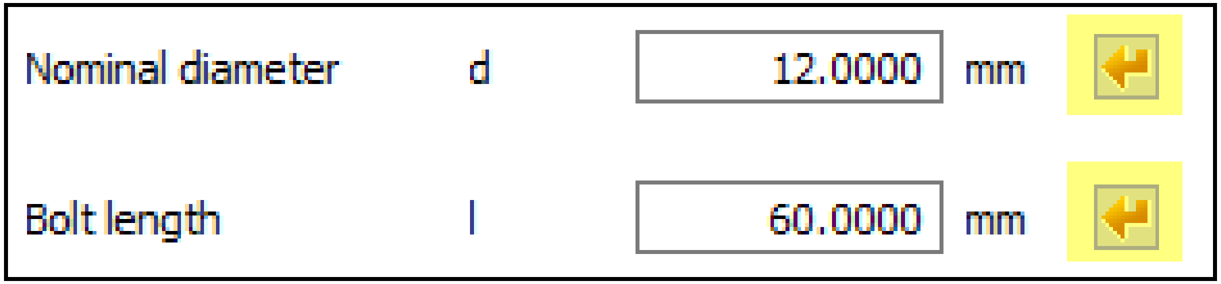
In the first step (step A) the next **highest load** to the load acting on the bolt joint is selected.

Depending on the **type of load** (static, dynamic, transverse load..) increase the number of load steps (step B).

Finally, the load is increased depending on the **tightening method** (step C).

Columns 2 to 4 give the required bolt dimensions in mm for the selected **strength grade** of the bolt (step D).

1	2	3	4
Load in N	**Nominal diameter** in mm		
	Strength grade		
	12.9	10.9	8.8
250			
400			
630			
1 000	3	3	3
1 600	3	3	3
2 500	3	3	4
4 000	4	4	5
6 300	4	5	6
10 000	5	6	8
16 000	6	8	10
25 000	8	10	12
40 000	10	12	14
63 000	12	14	16
100 000	16	18	20
160 000	20	22	24
250 000	24	27	30
400 000	30	33	36
630 000	36	39	

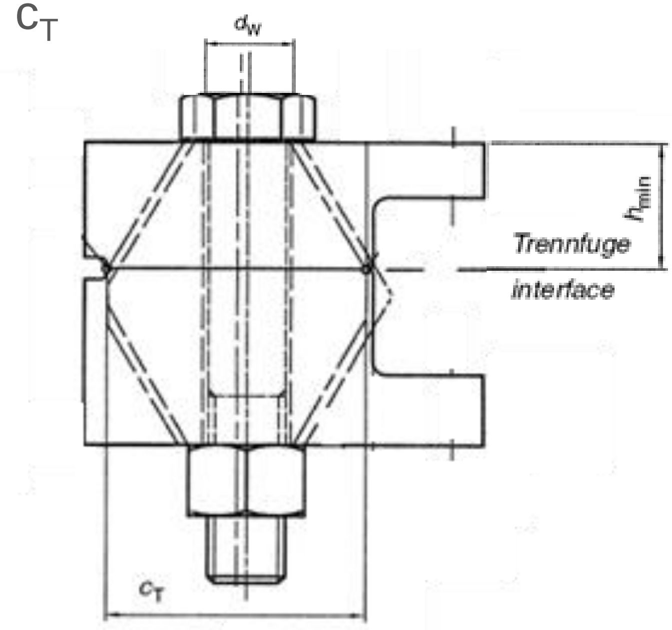


Sizing of bolt diameter
Based on VDI 2230, table A7, and the working load, the bolt diameter is proposed. The next higher diameter from database is used.

Sizing of bolt length
Based on the thickness of clamped parts and the bolt diameter the bolt length from KISSsoft database is proposed. If there is no suitable length, no proposition is given.

R0 nominal diameter and limiting size G

The calculation procedure of VDI 2230 for eccentrically clamped and eccentrically loaded joints is only valid within the limiting size G. The interface c_T has to be smaller than G:



TTJ: $G' = (1.5..2)d_w$

TBJ: $G = d_w + h_{min}$

$$c_T < G$$

d_w	outside diameter of the plane head bearing surface
h_{min}	thickness of the smaller plate of two clamped plates
c_T	size of the interface

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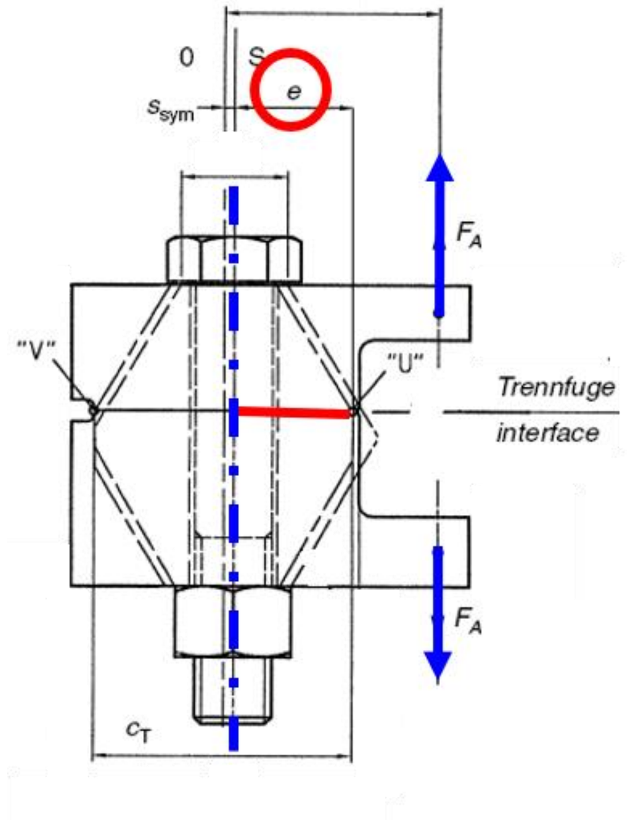
The distance to the point which is in danger of opening must nut be further than ½ G away from the bolt.

$$\frac{G_{\min}}{2} > e$$



e Distance from bolt axis to point U

"U" Point which is at risk of opening

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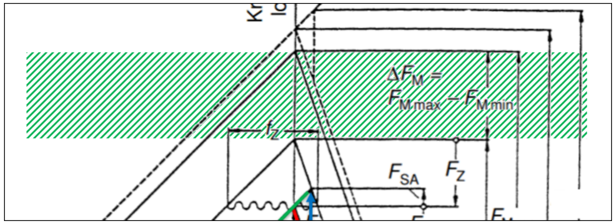
The required minimum assembly preload has to be achieved despite of the uncertainities in tightening (scatter of friction coefficient in the thread μ_G and in the head bearing area μ_K as well as tightening technique)

$$\boxed{F_{M\max} = \alpha_A \cdot F_{M\min}} = \alpha_A \cdot [F_{Kerf} + (1-\Phi_{en})F_A + F_Z + \Delta F_{Vth}]$$

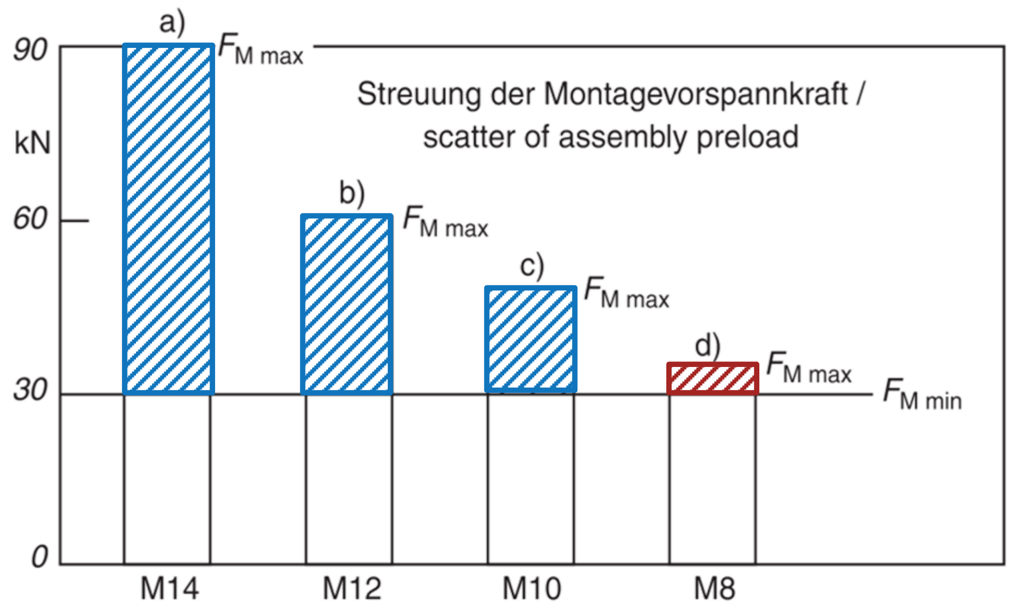
F_{Mmax}	required maximum assembly preload
α_A	tightening factor
F_{Mmin}	required minimum assembly preload

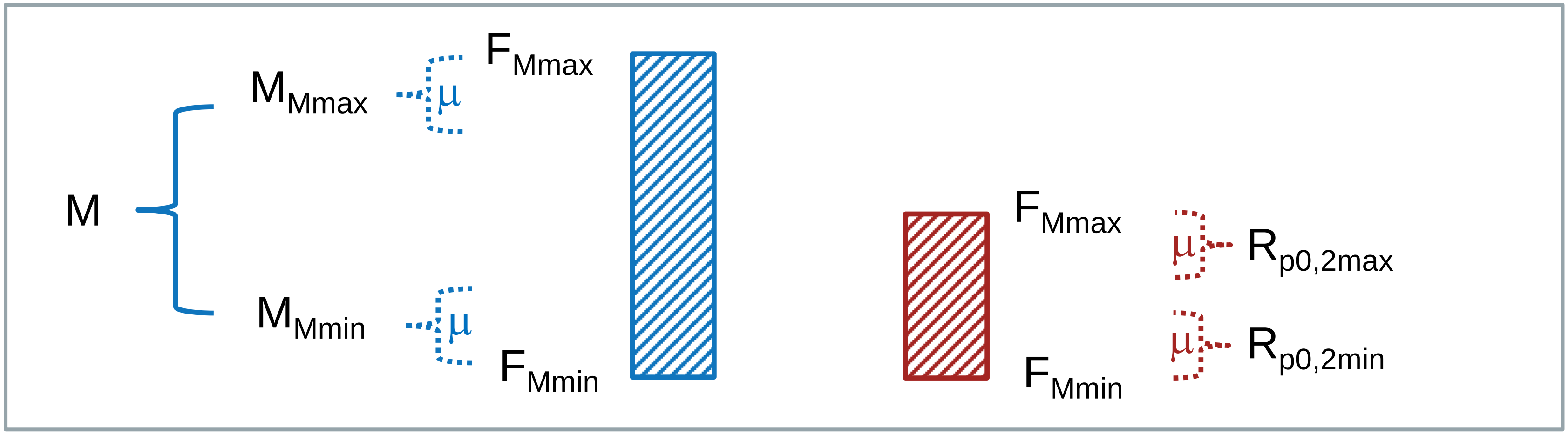


The tightening factor α is listed in table A8. The coefficients of friction are listed in table A5. Alternatively the tightening factor can be calculated based on scatter of tightening torque and friction.

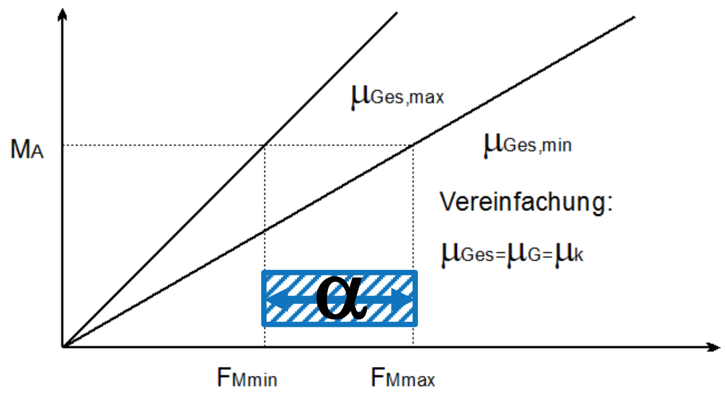
$$F_{M\max} = \alpha_A \cdot F_{M\min}$$

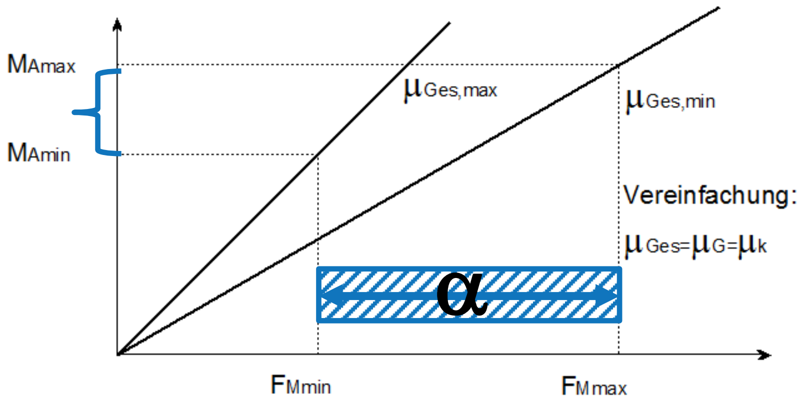
a) Impact wrench
b) Turning wrench
c) Torque wrench
d) Yield-controlled turning wrench





Torque-controlled tightening





Case 1: scatter of friction coefficient, no torque scatter

In case of an exact tightening torque, the tightening factor is only detemined by the scatter of friction coefficient.

Case 2: scatter of friction coefficient and tightening torque

In most cases both values vary. The tightening factor includes both effects.

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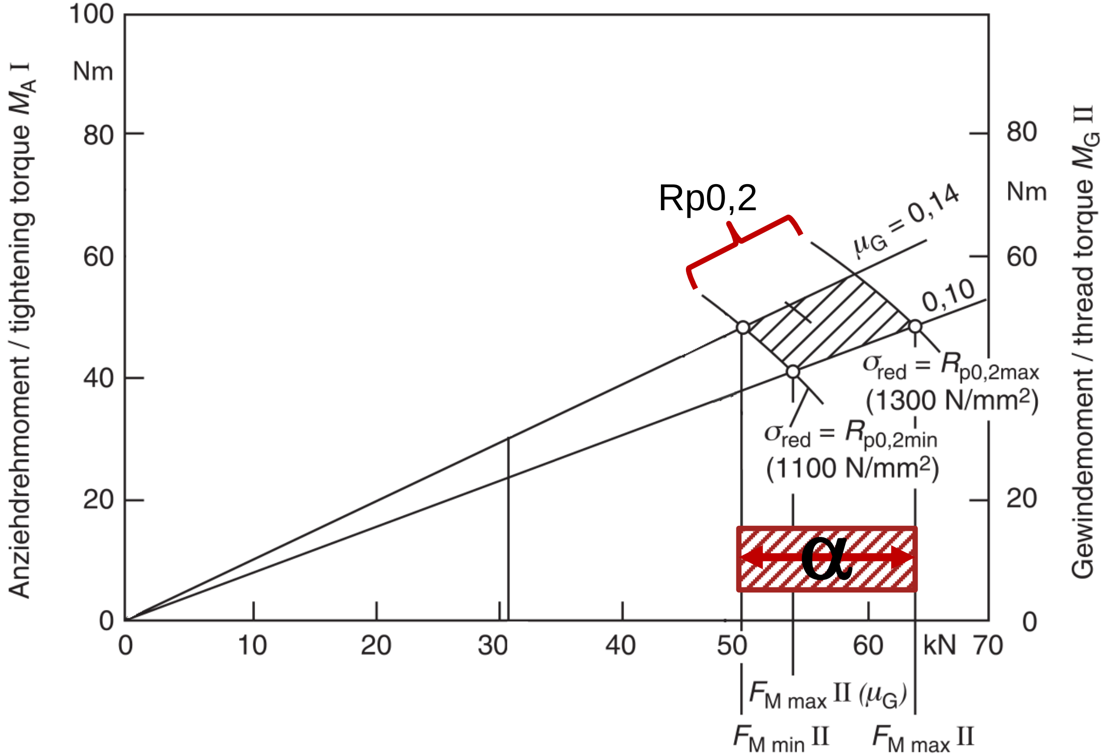
Yield-controlled tightening

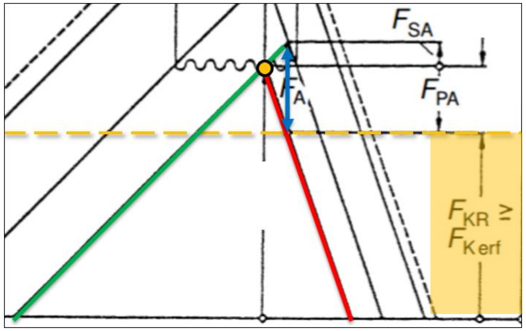


With yield-controlled tightening the yield point is controlled. The tightening is influenced by the scatter of yield value $R_{p0.2min/max}$ and friction of thread μ_G , but not by the friction of bolt head bearing area μ_K

The minimum clamp load is the minimum required clamp load to fulfill the functionality of bolt joint:

$$F_{M\max} = \alpha_A \cdot F_{M\min} = \alpha_A \cdot [F_{Kerf} + (1-\Phi_{en})F_A + F_Z + \Delta F_{Vth}]$$

$$F_{Kerf} \geq \max(F_{KQ}\,;F_{KP} + F_{KA})$$

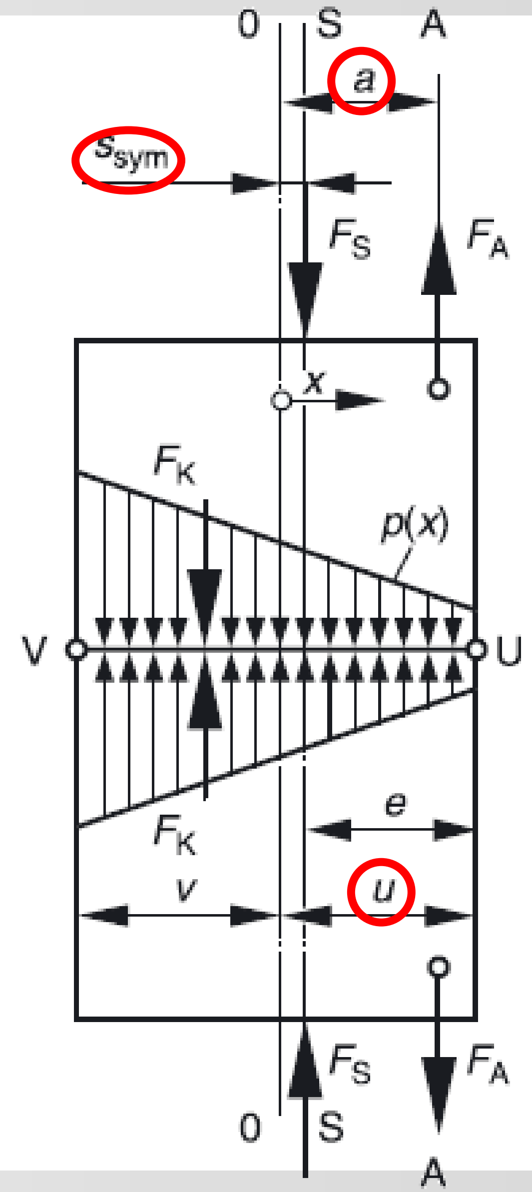


- Friction grip to transmit a transverse load F_{KQ} or torque
  the transverse load is calculated into an axial clamp load using the friction coefficient of the plates
- Sealing against a medium F_{KP}
- Prevention of opening F_{KA} (see next slide)

Opening can occure with eccentrically loaded or eccentrically clamped bolt joints.

$$F_{KA} = F_{Kab} = F_A(f(u,a,s_{sym})) + M_B(f(u,s_{sym}))$$

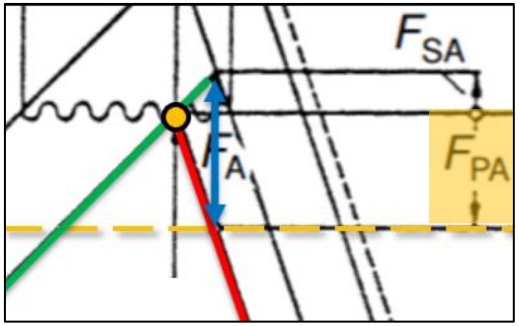
The clamp load at opening limit F_{Kab} depends on the distance of the loading a, the point of opening u and the eccentricity of the clamping s_{sym}



The axial working load F_A is divided into an additional bolt load F_{SA} and a plate relieving load F_{PA}

$$F_{M\max} = \alpha_A \cdot F_{M\min} = \alpha_A \cdot [F_{Kerf} + (1-\Phi_{en})F_A + F_Z + \Delta F_{Vth}]$$

$$F_{PA} = (1-\Phi) \cdot F_A$$



F_A	axial working load
F_{PA}	plate relieving load
Φ_{en}	load factor, describes the resilience of the plate δ_P in relation to the total resilience $\delta_{P,S}$

➔ In case of **no axial working load**, as i.e. only torque, these calculations aren‘t necessary

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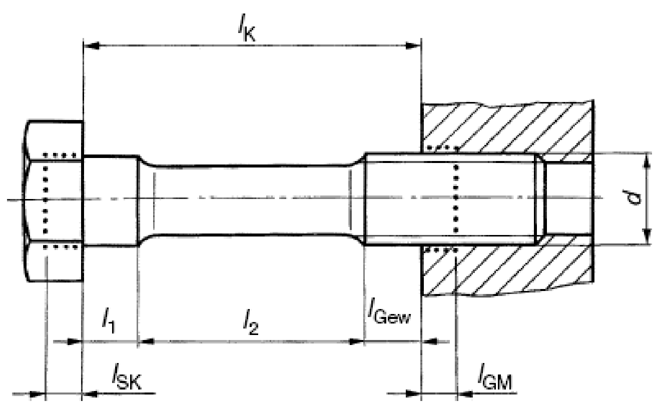
The elastic resilience of the bolt is calculated as follows:

$$\delta_S = \delta_{SK} + \delta_{S1} + \ldots + \delta_{Gew} + \delta_{GM}$$

δ_{SK}	Bolt head
$\delta_{S1\ ,2\ldots}$	Shafts, with the individual length dimensions
δ_{Gew}	Thread, unengaged part
δ_{GM}	compressive deformation of the nut



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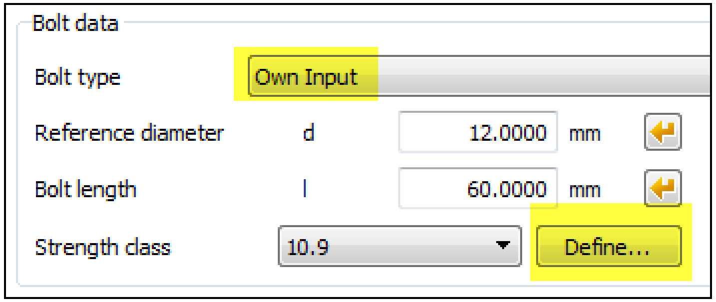

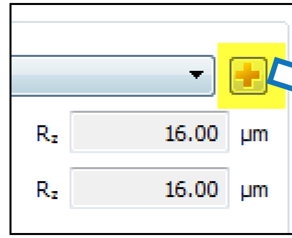

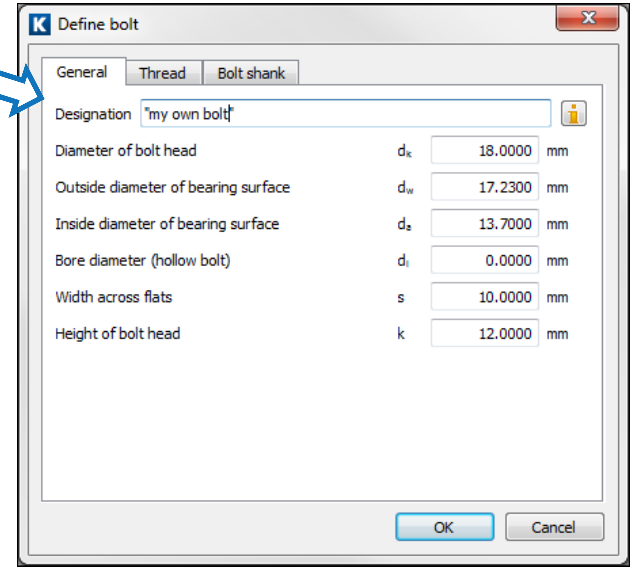

Add a bolt

A bolt can directly be defined with «own input» then with «Define…» and «+».

This way you can enter anti-fatigue bolts or hollow bolts etc.

If you preselect a standardized bolt the values are shown as default inputs after switching to «own input».

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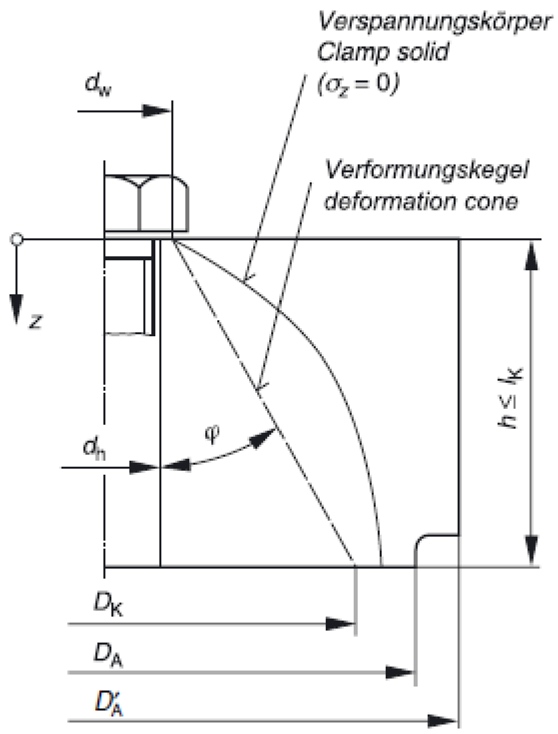
The elastic resilience of the plates is calculated as follows:

$$\delta_P = \delta_{P1} + \delta_{P2} + \dots$$

For the calculation of the resilience, the clamp solid body is substituted by a virtual deformation cone with the same resilience.

The cone angle is calculated individually based on the actual geometrical and clamping conditions.



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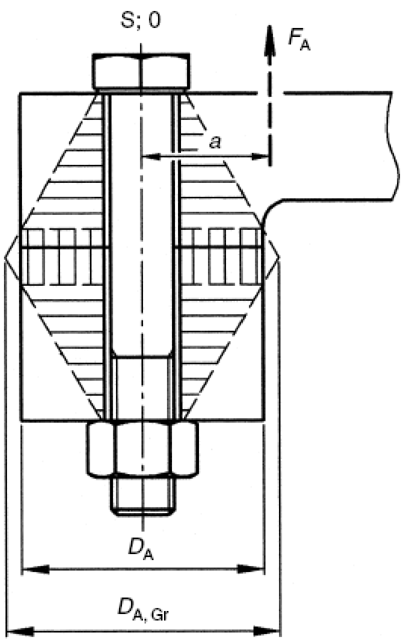
To determine the resilience of the (substitutional) deformation body, it has to be checked, whether the deformation cones reach the outer edge of the plates.

As a result of i.e. large clamp length or small plate dimensions, the cones are disturbed by the outer egde, and the middle part of the deformation body is changed to a sleeve.

The limiting diameter $D_{A,Gr}$ serves to settle the question as to whether a deformation sleeve (cylindrical ring) is present between the cones:

$$D_{A,Gr} = d_W + w \cdot l_K \cdot \tan\varphi$$

TBJ: w = 1
TTJ: w = 2



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The load factor describes, how much the bolt load F_S is increased by the additoinal bolt load F_{SA} , when the axial working load F_A is applied:

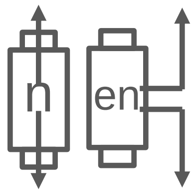
$$\Phi_K = \frac{\delta_P}{\delta_S + \delta_P} = \frac{F_{SA}}{F_A}$$

δ_S	resilience of the bolt
δ_P	resilience of the plates
Φ_K	load factor

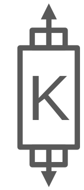
The load factor Φ_K presumes the (theoretical case of) load introduction directly under the bolt head and nut bearing areas (n=1).

For eccentrically loaded joints or load which is not introduced directly under the head/nut, the **load introduction factors a and n** have to be considered. For eccentrically clamped joints the factor $\mathbf{s_{sym}}$.

R3 load factor (extended)



$$\Phi_n = \Phi_{en} = n \frac{\delta_P + \delta_{Pzu}}{\delta_S + \delta_P + \delta_{PH}} n\Phi_K$$



if TTJ

if extension sleeve

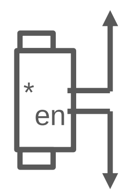
e: Eccentric loading

n: load introduction inside clamped part

$$\Phi_n^* = n \frac{\delta_P + \delta_{Pzu}}{\delta_S + \delta_P^* + \delta_{PH}}$$

* = eccentricly clamped

δ_P^* considers s_{sym}



$$\Phi_{en}^* = n \frac{\delta_P^{**} + \delta_{Pzu}}{\delta_S + \delta_P^* + \delta_{PH}}$$

δ_P^{**} considers s_{sym} and a

eccentricly clamped (*)
and eccentricly loaded (e)

δ_P^* considers s_{sym}

$$\delta_P^* = \delta_P + s_{sym}^2 \cdot \beta_P$$
$$\delta_P^{**} = \delta_P + a \cdot s_{sym} \cdot \beta_P$$

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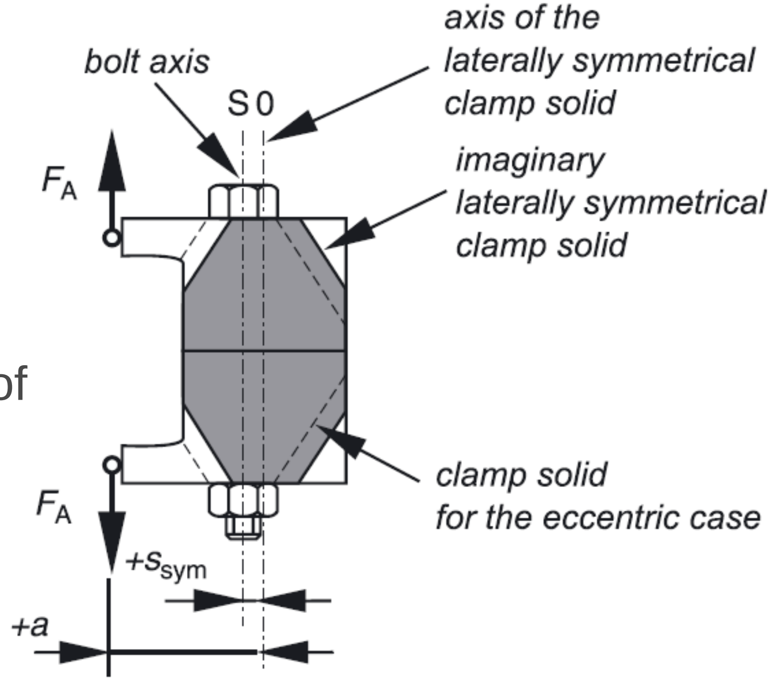
For eccentricly clamped and loaded parts, the influence on the plate resilience is taken into account by the factors s_{sym} and a.

s_{sym} distance from the bolt axis to the imaginary laterally symmetrical clamp solid

a the distance of the substitutional line of action of the axial working force F_A to the axis of the imaginary laterally symmetrical clamp solid



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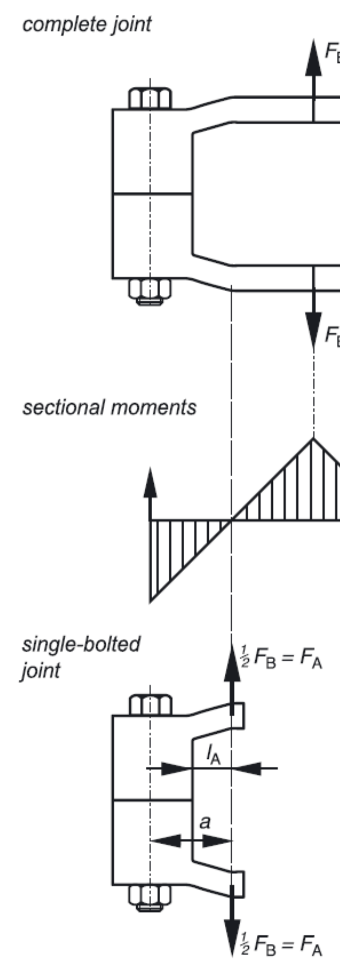
The single-bolted joint should be released (cut away) from ist surroundings on the load side in such a way, that the sectional planes are free of moments.

Hence the distance of load introduction **a** is determined in such a way, that the moment is = 0.

For the bolt calculation only the axial working load $\mathbf{F_A}$ and the distance **a** are used.



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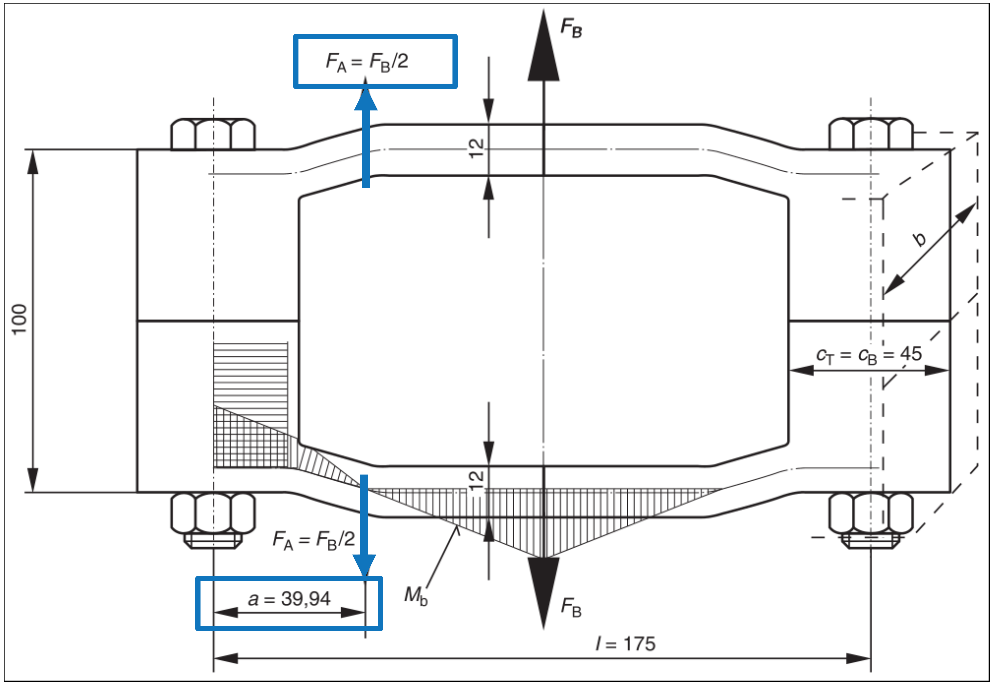
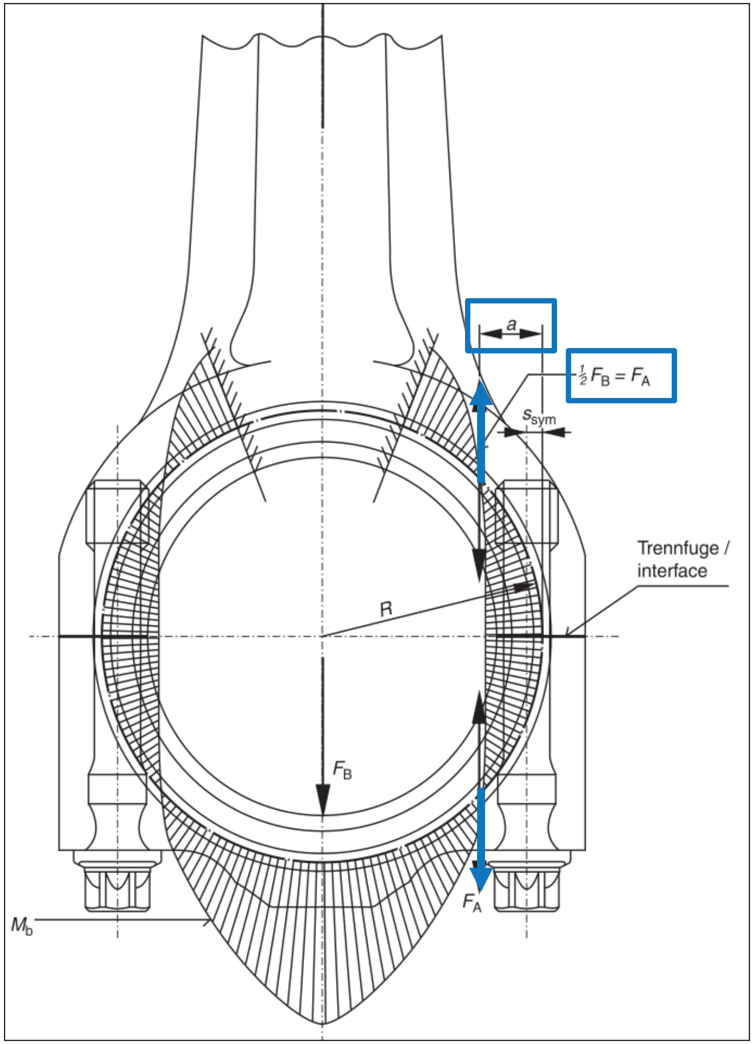
R3 Eccentrial loading





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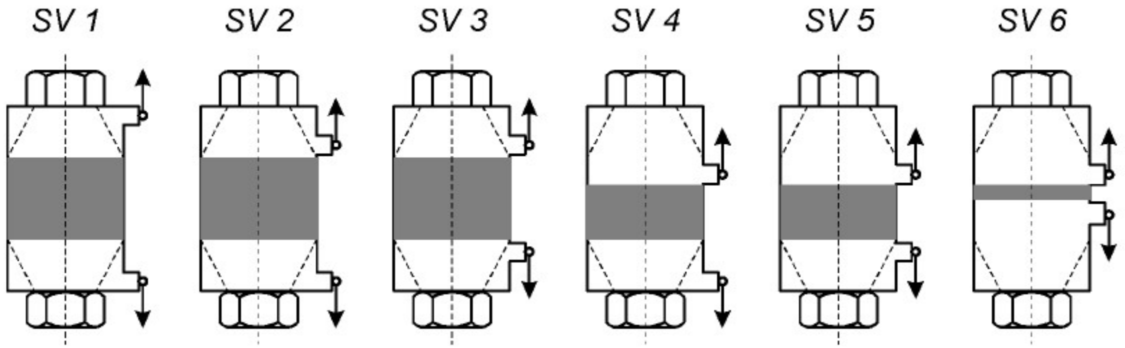


n= 0.7 … … … … n=0.15 (values are for centrical loading)

The load introduction factor **n** considers the axial height and lateral distance of the load application.

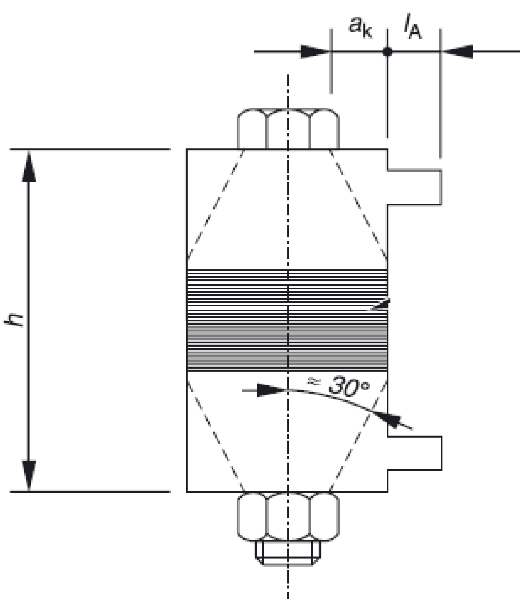
With SV1, the force is close to the bolt head, which results in higher additional bolt load than with SV6. For tapped thread joints (TTJ) the types SV1, SV2 and SV4 are to be applied.

The bigger the eccentricity, the smaller the load introduction factor n.

The eccentricity is divided in a part from basic solid a_K and a connecting solid l_A .

h	height of bolt joint
a_K	distance between the edge of preloading area and the force introduction point of the basic solid
l_A	length between basic solid and load introduction point

A load introduction factor n<1 decreases the additional bolt load F_{SA}



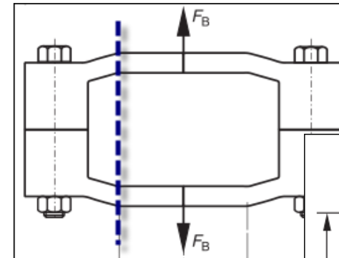
Simplified Procedure:

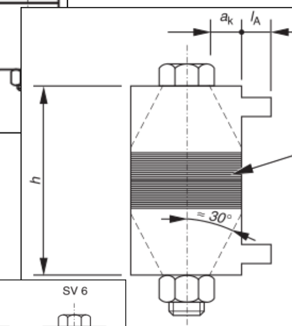
1) Release single-bolted joint

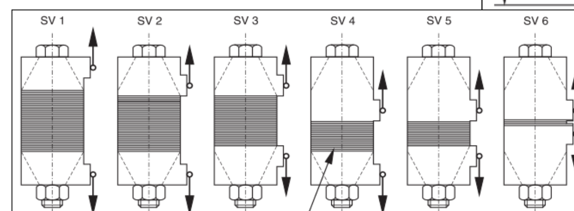
2) Divide into basic and connecting solid

3) Establish joint type

4) Determine n from table







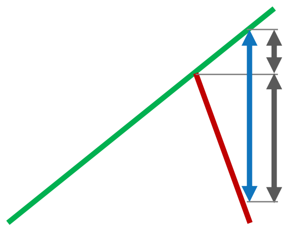
														≥ 0,30			
												≥ 0,50	0,00	0,10	0,30	≥ 0,50
SV 1	0,70	0,55	0,30	0,13	0,52	0,41	0,22	0,10	0,34	0,28	0,16	0,07	0,16	0,14	0,12	0,04
SV 2	0,57	0,46	0,30	0,13	0,44	0,36	0,21	0,10	0,30	0,25	0,16	0,07	0,16	0,14	0,12	0,04
SV 3	0,44	0,37	0,26	0,12	0,35	0,30	0,20	0,09	0,26	0,23	0,15	0,07	0,16	0,14	0,12	0,04
SV 4	0,42	0,34	0,25	0,12	0,33	0,27	0,16	0,08	0,23	0,19	0,12	0,06	0,14	0,13	0,10	0,03
SV 5	0,30	0,25	0,22	0,10	0,24	0,21	0,15	0,07	0,19	0,17	0,12	0,06	0,14	0,13	0,10	0,03
SV 6	0,15	0,14	0,14	0,07	0,13	0,12	0,10	0,06	0,11	0,11	0,09	0,06	0,10	0,10	0,08	0,03

Full Procedure:

-> See VDI 2230 Annex C (pages 171-175)

R3 load factor

$$\Phi_{n} = n \cdot \Phi_{K}$$



$$F_{SA} = \Phi_n \cdot F_A$$

$$F_{PA} = (1 - \Phi_n) \cdot F_A$$

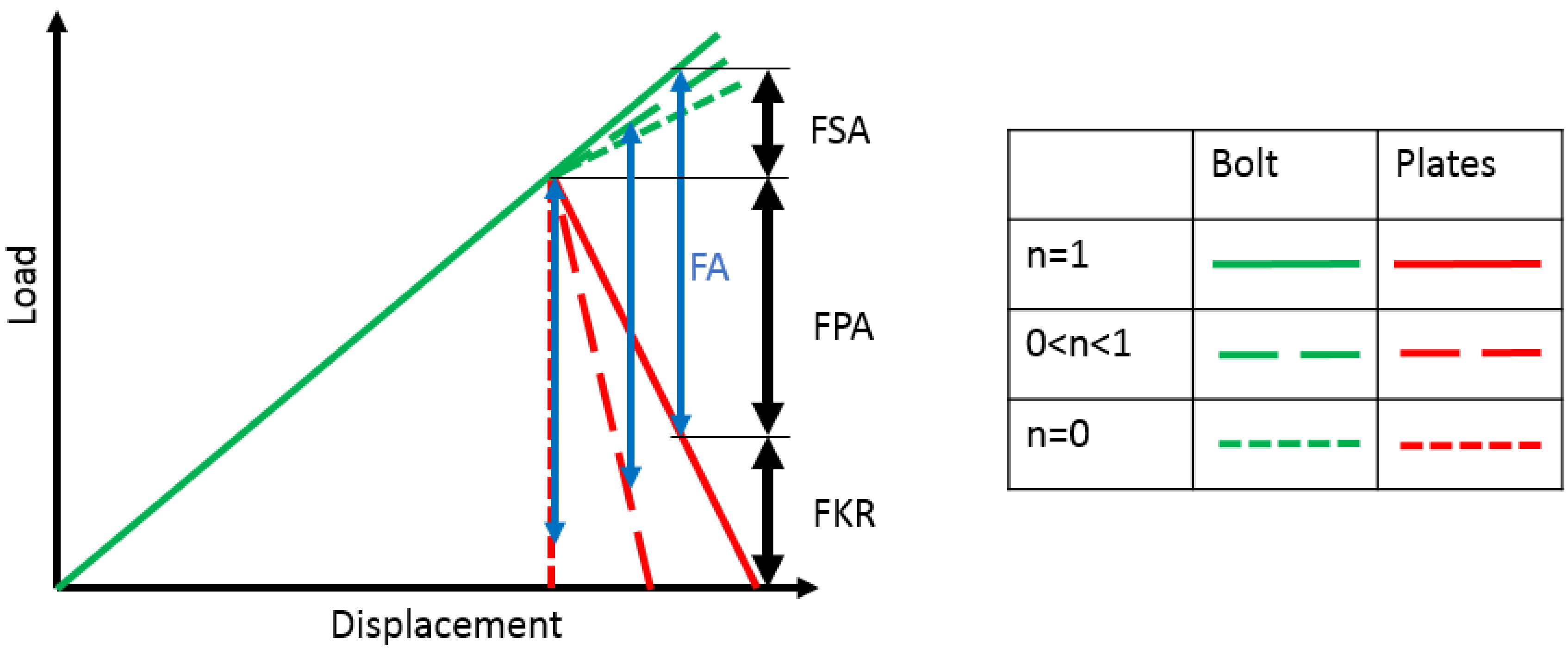
Φ_n	load factor, considering the load introduction factor n
n	load introduction factor
$F_{SA/PA}$	additional bold load/plate load (plate relieving load)

With **n=1**: The **additional bolt load** F_{SA} is highest

With **n=0**: The **additional plate load (relieving load)** F_{PA} is highest**,** the interface is relieved and the minimum clamp load is decreased

The joint diagram looks different depending on the load introduction factor.

The inclination of the lines for F_{PA} and F_{SA} change based on the value of n.



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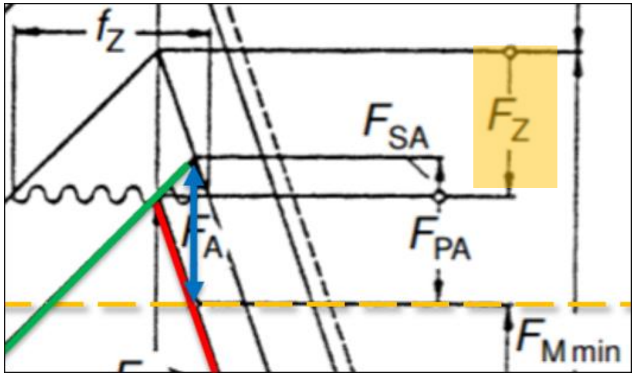
$$F_{M\max} = \alpha_A \cdot F_{M\min} = \alpha_A \cdot [F_{Kerf} + (1-\Phi_{en})F_A + F_Z + \Delta F_{Vth}]$$



Preload changes occur mainly due to two effects:

- Embedding due to flatening of surface roughness or relaxation

  The guide values for the amount of embedding f_z can be taken from tables in VDI 2230.

- Different thermal expansion due to different materials and coefficients or different temperatures

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„Main dimensioning formula“

$$F_{M\max} = \alpha_A \cdot F_{M\min} = \alpha_A \cdot [F_{Kerf} + (1 - \Phi_{en})F_A + F_Z + \Delta F_{Vth}]$$

F_{Mmax}	maximum required assembly preload
α_A	tightening factor
F_{Mmin}	minimum required assembly preload
F_{kerf}	required clamp load in the interface
Φ_{en}	load factor for eccentrical clamping and loading
F_A	axial working load
F_Z	preload changes due to embedding
ΔF_{Vth}	preload changes due to temperature influences

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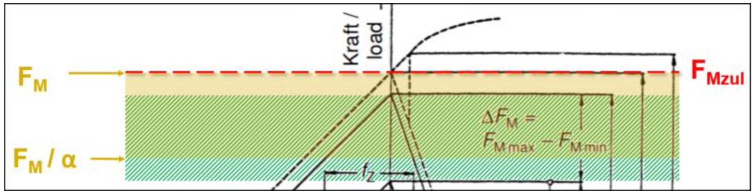
$$\sigma_{red,Mzul} = \sqrt{\sigma_M^2 + 3\tau_M^2} = R_{p0.2} \cdot \nu$$

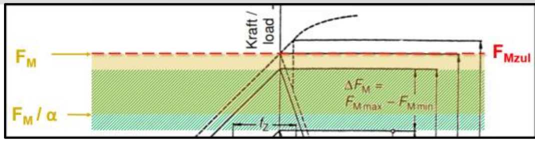


σ_M	tension stress
τ_M	torsional stress
$\sigma_{red,Mzul}$	uniaxial comparative stress in the assembly state
ν	utilization factor, mostly 90%
$R_{p0,2min}$	minimum yield point

During assembly – with the usual tightening techniques with torsional stress – tension stress σ_M and torsional stress τ_M are present.

These stresses are converted by means of the deformation energy theory (GEH) to an equivalent uniaxial stress $\sigma_{red,Mzul}$.

R7 Assembly stress



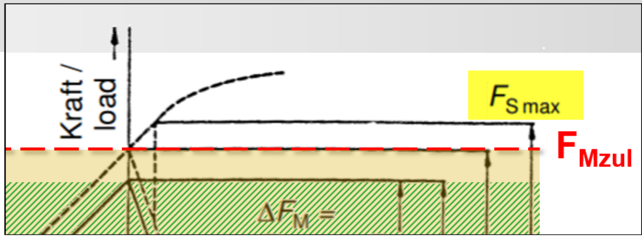
$$\sigma_{Mzul} = \frac{\sigma_{red,Mzul}}{\sqrt{1+3\cdot\left[\frac{3}{2}\frac{d_2}{d_0}\left(\frac{P}{\pi\cdot d_2}+1.155\cdot\mu_{G\min}\right)\right]^2}}$$

$$\boxed{F_{Mzul}} = A_0\cdot\sigma_{Mzul}$$

$\sigma_{red,Mzul}$	uniaxial comparative stress in the assembly state
σ_{Mzul}	permissible tension, determined from comparative stress
F_{Mzul}	permissible mounting force, based on bolt strength
A_0	minimum cross-sectional area of the bolt (usually with $0.5(d_2+d_3)$)

➔ The permissible mounting force F_{Mzul} must be larger than the maximum required mounting force F_{Mmax}

R8 Working stress



$$\sigma_{z\max} = \frac{F_{S\max}}{A_S} = \frac{F_{Mzul} + \Phi_n \cdot F_{A\max} - \Delta F_{Vth}}{A_S}$$

$$\tau_{\max} = \frac{M_G}{W_P}$$

σ_{zmax} maximum stress in working state
τ_{max} torsional stress

Using the axial working load, the stresses in working state are higher because of the operating load than in the assembly state

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$$\sigma_{red,B} = \sqrt{\sigma_{z\max}^2 + \left(k_\tau \cdot \tau_{\max}\right)^2}$$

$$\underline{\underline{S_F}} = \frac{R_{p0,2\min}}{\sigma_{red,B}} \geq 1$$

$\sigma_{red,B}$ comparative stress in working state
k_τ factor to consider the lower torsion (compared with assembly state, recommended is 0.5

S_F Safety against deformation
$R_{p0,2min}$ minimum yield point

➔ **safety against deformation**
The calculation is not applicable, if there is no axial working load.
The safety is then calculated in assembly state.

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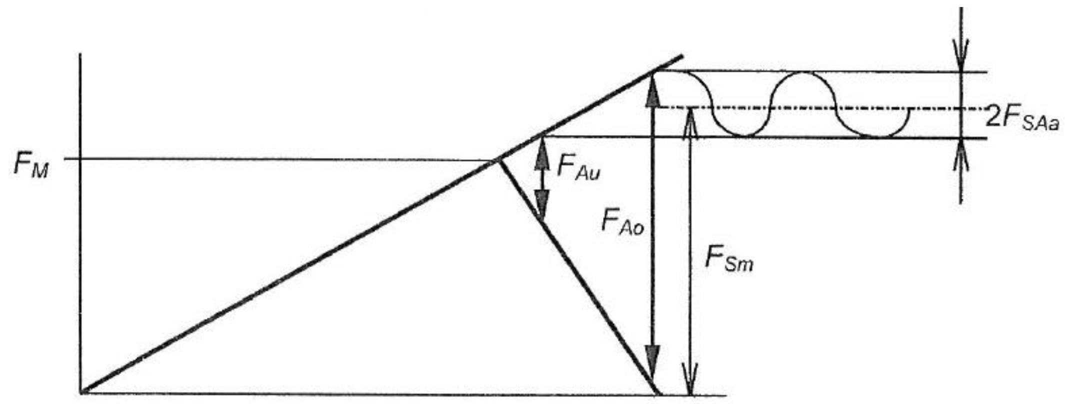
Hydraulic friction and torsion free mounting can also be considered.

$$R_{p0,2\min} \cdot A_0 \geq F_{S\max}$$

$$\underline{\underline{S_F}} = \frac{R_{p0,2\min}}{\sigma_{z\max}} \geq 1$$

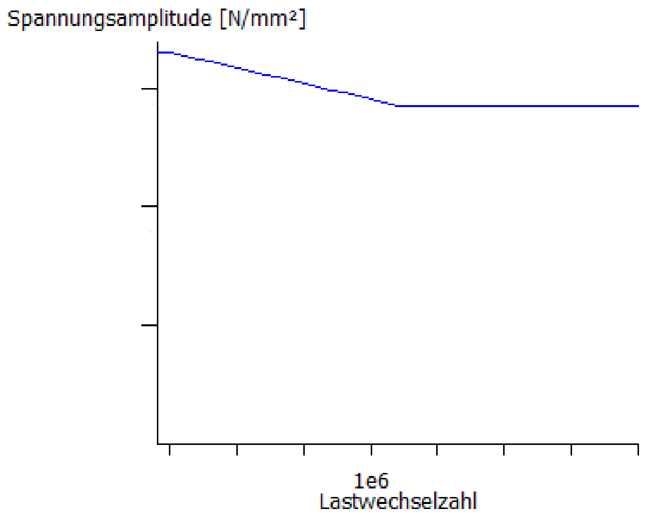
$R_{p0,2min}$ minimum yield point
σ_{zmax} maximum stress in working state
S_F safety against deformation

$$\sigma_a = \frac{\Phi_K \cdot \left(F_{Ao} - F_{Au}\right)}{2 \cdot A_S}$$



F_{Ao}	upper load
F_{Au}	lower load
Φ_K	load factor
A_S	stress cross section
$\sigma_{a/ab}$	continuous alternating stress acting on the bolt

The **acting stress amplitude** is calculated with the difference between upper and lower additional bolt load.



$$\sigma_{ASV} = 0.85\left(\frac{150}{d} + 45\right)$$

$$\sigma_{AZSV} = \sigma_{ASV}\left(\frac{N_D}{N_Z}\right)^{1/3}$$

$\sigma_{ASV,G}$ stress amplitude of the endurance limit of bolts rolled before or after the heat treatment, for load cycles $N_D>2*10^6$

$\sigma_{AZSV,G}$ stress amplitude of the fatigue strength of bolts rolled before or after the heat treatment, for load cycles $10^4 < Nz < 2*10^6$

The **permissible stress amplitude** for the endurance limit and fatigue strength (with failure probability of 1%) are listed in VDI 2230.

$$\underline{\underline{S_D}} = \frac{\sigma_{AS}}{\sigma_{a/ab}} \geq 1.2$$

S_D sagety against fatigue
σ_{AS} permissible stress amplitude
$\sigma_{a/ab}$ acting stress amplitude

➔ **safety against fatigue**
The calculation is not applicable, if there is no axial working load.

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$$\underline{\underline{S_P}} = \frac{p_G}{p_{M/B\max}} \geq 1$$

S_P	Safety against pressure
p_G	limiting surface pressure
$p_{M/Bmax}$	maximum acting surface pressure in assembly or working state

The maximum surface pressure is obtained with highest load (assembly or working state) and smallest area.

For the area the exact geometry is used, considering chamfers etc.

$$m_{erf} \leq m_{vorh,eff}$$

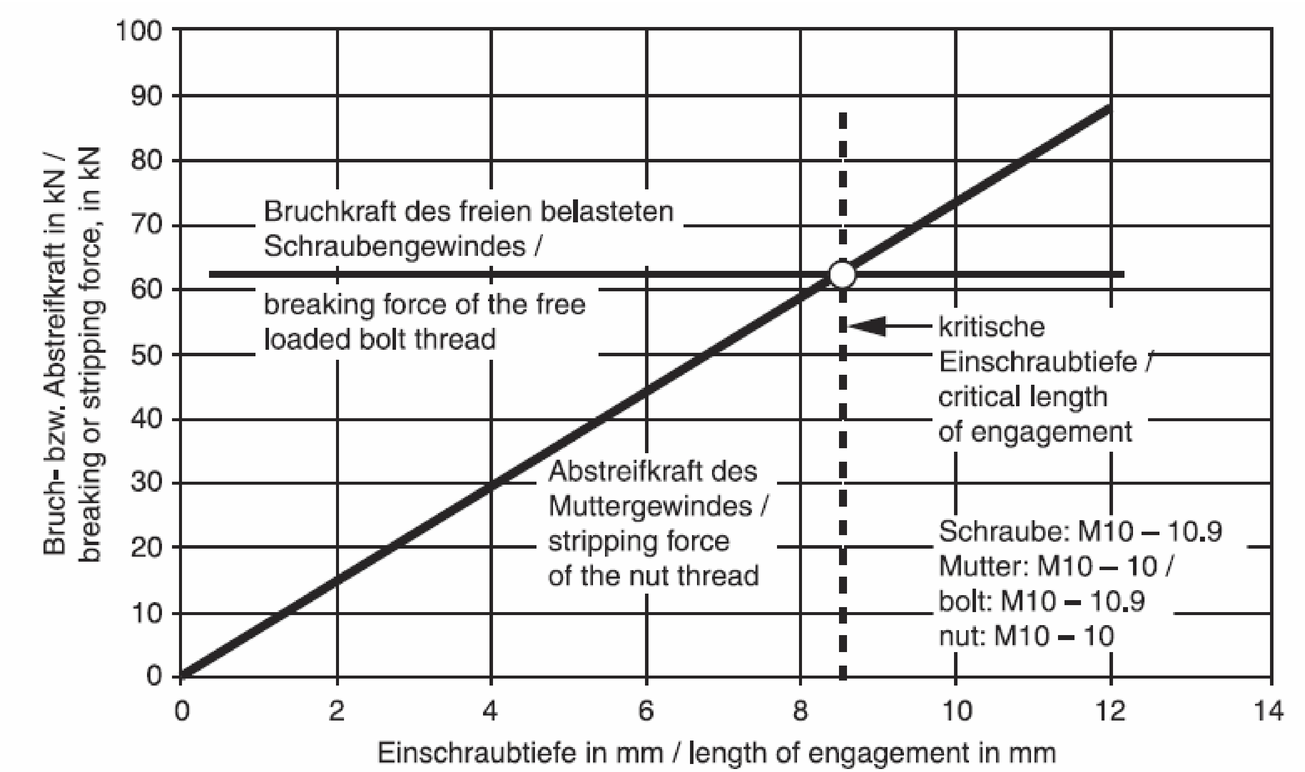
$m_{eff,min}$ required length of engagement
$m_{vorh,\ eff}$ actual length of engagement

The maximum tensile force of the bolt must be less than the critical stripping force of the internal or bolt thread.

$$F_{mS} \leq \min\left(F_{mGM} ; F_{mGS}\right)$$

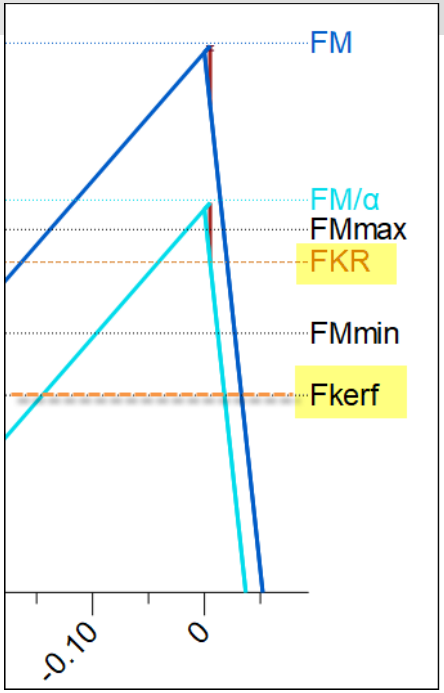
F_{mS} Breaking force of the free loaded bolt thread
F_{mGM} Stripping force of nut or internal thread
F_{mGS} Stripping force of bolt thread

R11 minimum length of engagement



The effective length of engagement is determined by the bolt length minus clamping length and thread chamfer.

$$\underline{\underline{S_G}} = \frac{F_{KR\min}}{F_{KQerf}} \geq 1$$



S_G safety against slipping

F_{KRmin} residual clamp load at the interface considering relief by F_{PA} and after embedding

F_{KQerf} minimum required clamp load at the interface for transmitting a transverse load and / or a torque by friction grip

The calculation is not applicable, if there is no transverse working load.

$$\underline{\underline{S_A}} = \frac{\tau_B}{\tau_{Q\mathrm{max}}} \geq 1.1$$

S_A safety against shearing
τ_B shearing strength, is determined with the „shearing strength ratio" in table 5.5/2 (values for ratio of shearing strength / tensile strength)
τ_{Qmax} shearing stress in the bolt cross section at the interface

Overloading resp. overcoming the static friction at the interface may lead to shearing or bolt-bearing.

For the determination of the shearing stress the full transverse load F_{Qmax} is used.

$$M_A = M_G + M_K$$

$$M_A = \boxed{F_{Mzul}}[0.16 \cdot P + 0.58 \cdot d_2 \cdot \mu_{G\min} + \frac{D_{Km}}{2} \cdot \mu_{K\min}]$$

M_A	tightening torque
M_G	thread friction moment
M_K	head friction moment
P	pitch of the thread
μ_{Gmin}	(minimum) friction coefficient in the thread
μ_{Kmin}	(minimum) friction coefficient in the bolt head area

The tightening torque required for torque-controlled tightening can be taken from tables in VDI 2230 (with utilization 90%).

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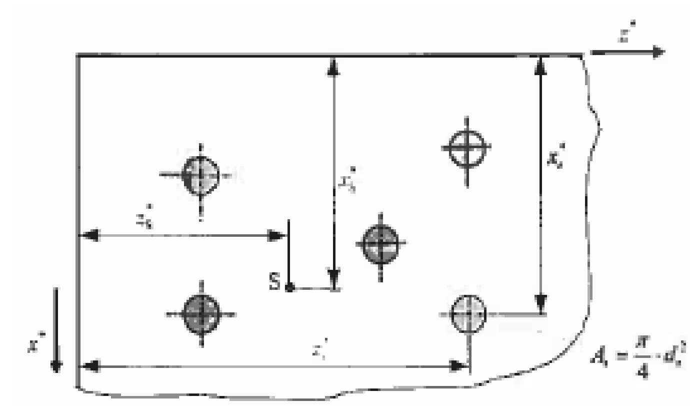
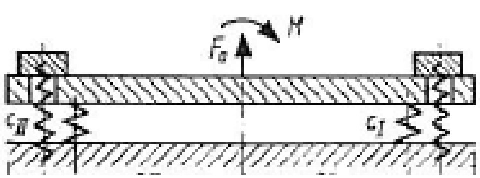
Part 2 considers multi bolt joints which is applicable for general geometries and bolt positions.



Rigid body mechanics

The plates are rigid, the bolts are not preloaded



Elasto mechanics

The plates are pliable, the bolts are not preloaded

FE method

The plates and bolts are considered with their real properties.

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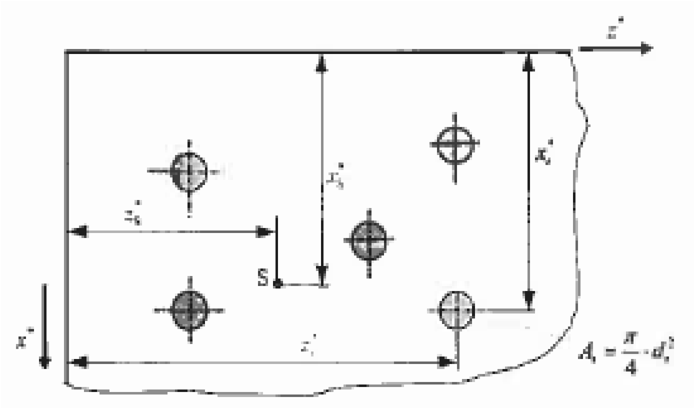
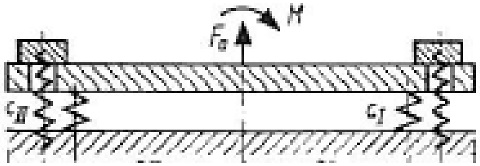
The guideline VDI 2230, part 2 (draft status) provides a calculation procedure for load distribution (axial forces, bending, torque) in multi-bolted joints, consideriung also the individual bolt positions and bolt sizes.



For the load distibution the plates are rigid (inelastic), but the bolts are flexible.



Rechengang bei Axialkräften

Eine Axialkraft angreifend im Schwerpunkt:

$$F_{A\max} = \frac{F_B}{n_S}$$

Axialkraft angreifend ausserhalb des Schwerpunktes erzeugt ein zusätzliches Biegemoment:

$$F_{A\max} = F_{A(Moment)} + \frac{F_B}{n_S}$$

Eine schief angreifende Kraft wird in die Komponenten Axialkraft und Querkraft aufgeteilt:

$$F_A = F_B \cdot \sin\alpha$$

$$F_Q = F_B \cdot \cos\alpha$$

Rechengang bei Biegemomenten

Berechnung des Schwerpunktes der Schraubenverbindung, mit Berücksichtigung der Positionen und Schraubenflächen

Berechnung der neutralen Faser (rote Linien, durch Schwerpunkt der Linien) und Flächenträgheitsmomente I_{xx}, etc. bezüglich der Schwerpunktachse, also mit dem Satz von Steiner umgerechnet

Berechnung der resultierenden Schraubenkraft F_{Amax} über:

$$F_{A\max} = \frac{M_x \cdot A \cdot z_i}{I_{xx}} + \frac{M_z \cdot A \cdot z_i}{I_{zz}}$$

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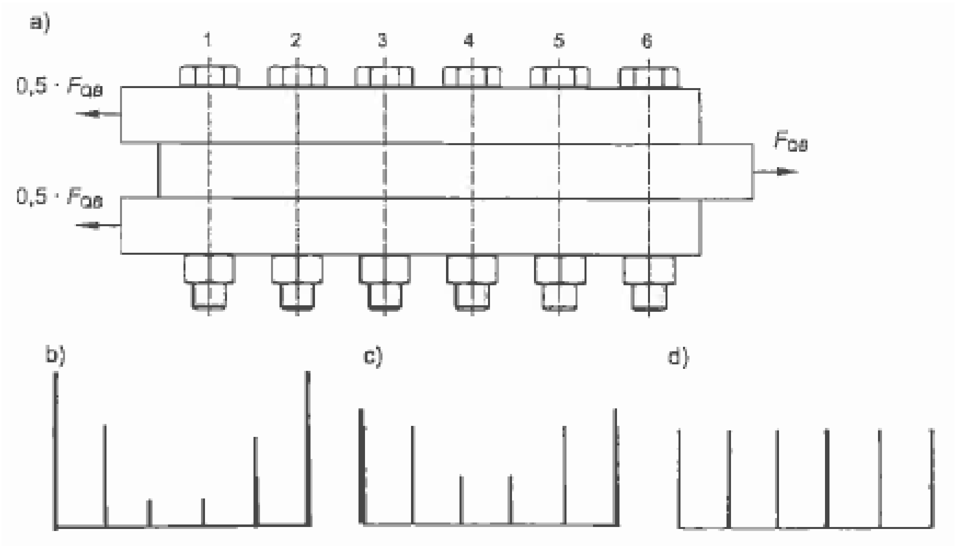
Rechengang bei Querkräften

Mit dem Starrkörperansatz ist die Verteilung der Querkraft pro Schraube gleichmässig:

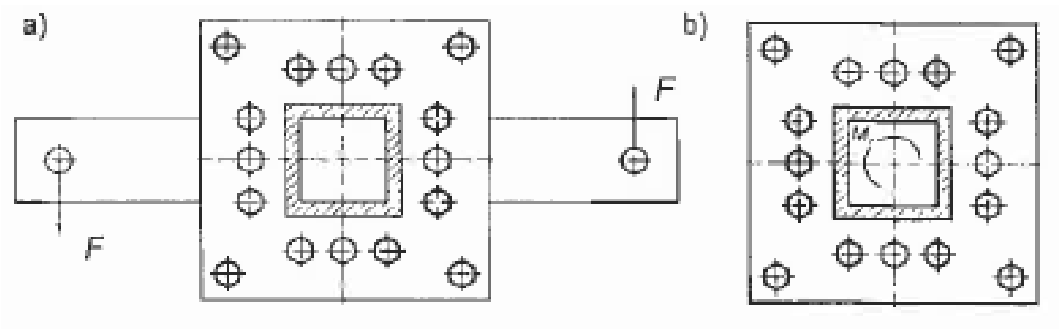
$$F_{qB\max} = \frac{F_{QB}}{n_S}$$



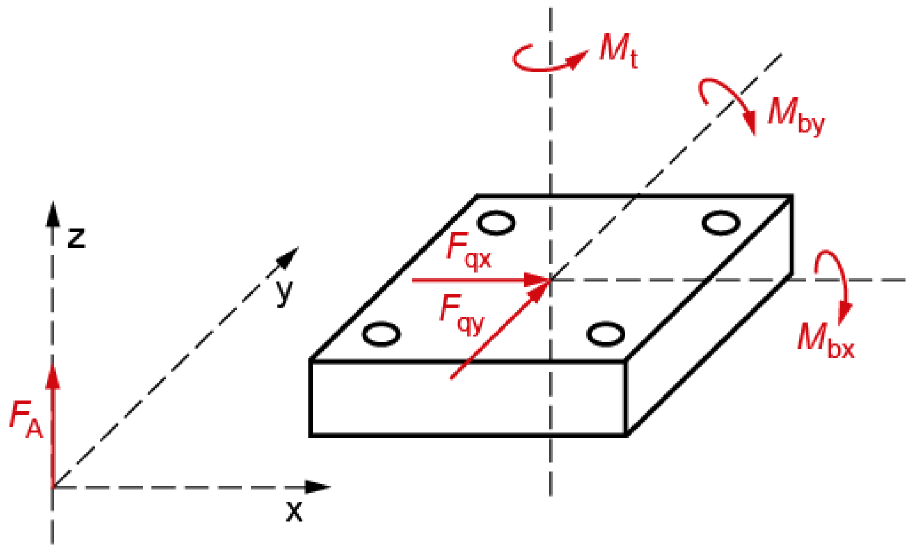
Rechengang bei Torsionsmoment

Für die äussersten Schrauben gilt bei einer proportionalen Aufteilung der Reaktionskräfte:

$$F_{qM\,\max} = \frac{M_B \cdot \left(\sqrt{x^2 + z^2}\right)_{\max}}{\sum_{i=1}^{n}\left(x^2 + z^2\right)}$$



Definition of Moments and Forces in KISSsoft
(Right hand rule)



Schematic

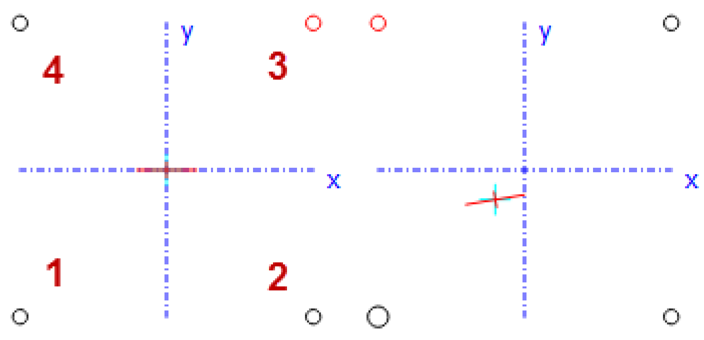
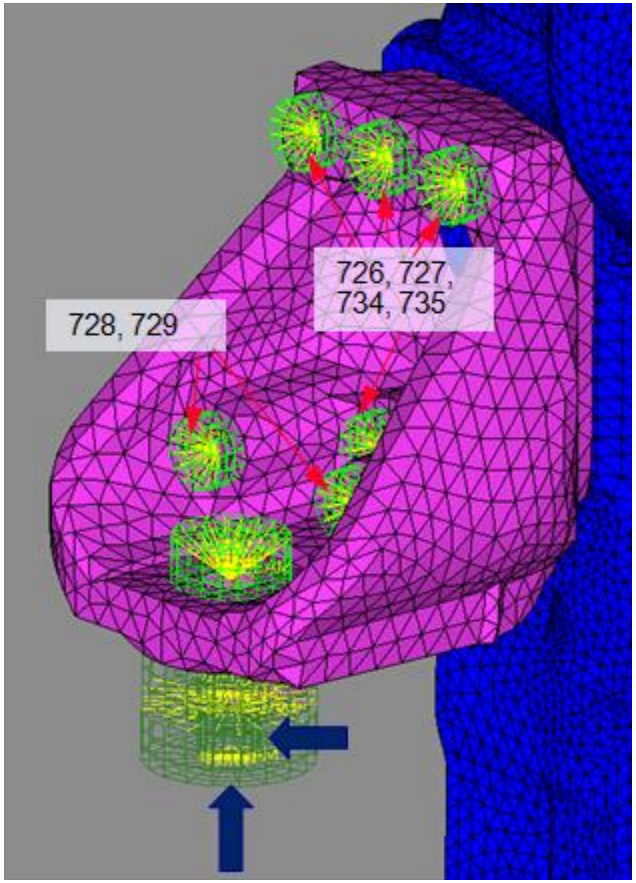


Diagram
Center of area: cyan lines
Neutral axis: red lines

With FE analysis the real plate geometries and loads can be considered.

This is a big advantage compared to the VDI guideline and becomes more and more applied in the industry.

The guideline VDI 2230 part 2 provides 4 classes of FE modelling procedures and explains how to integrate the results into the calculation of VDI 2230 part 1.



Integration of FEM results into VDI 2230 (part 2)

The classes I – IV have an increasing accuracy regarding FE modelling. Hence, also the effort of modelling increases.

Modelling class I is very simplified, i.e. no modelling of bolt.
Classes II & III show a good ratio of effort and benefit.
Class IV is very detailed, i.e. also the thread is modelled.

Modelling class	**I**	**II**	**III**	**IV**
effort of modelling	low	medium	medium	high
modelling of bolt	none	1-dim	3-dim (simplified)	3-dim (detailed)
contact in interface	none	possible	modelled	modelled
pretension	none	possible	modellled	modelled

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Integration of FEM results into VDI 2230 (part 2)

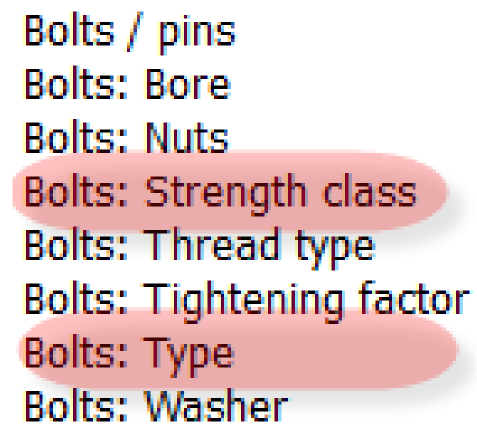
According to the modelling accuracy the factors can be used from FE instead of VDI 2230, part 1.

Red: the parameters are taken from VDI 2230 part 1, tests, ..
Yellow: the parameters are from FE modelling

Parameter	I	II	III	IV
δ_s				
δ_p (δ_p^* δ_p^{**})				
n				
α_K				
f_Z				
$\Delta F_{therm.}$				
μ				

Add bolt types
In the database „bolts:Type“ additional bolts (length, diameter, ..) can be added. The data are saved in a Ascii file.

Add strength classes
In the database „bolts: Strength class“ additional materials (Aluminium, casting,..) can be added.



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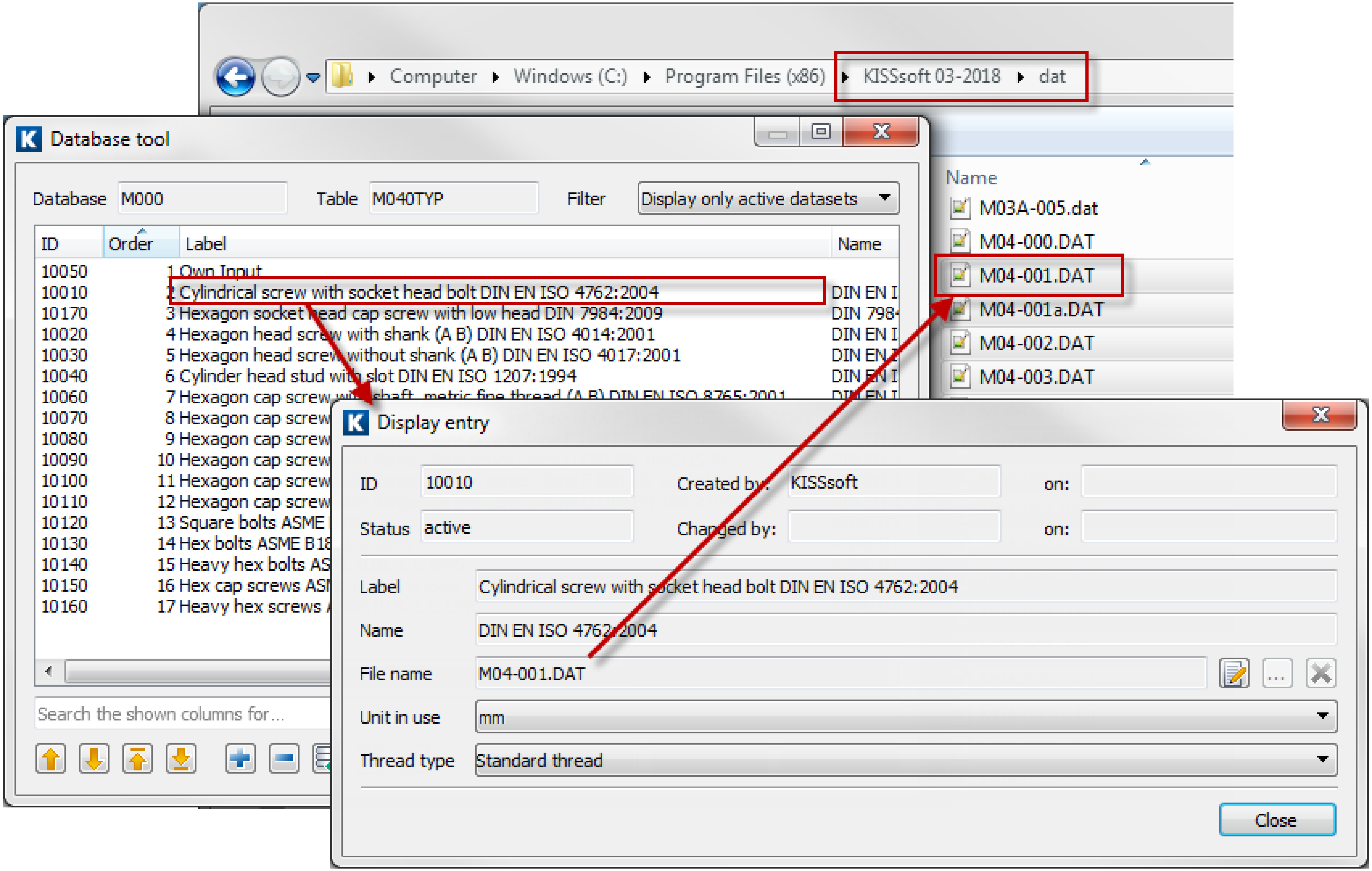
Bolt geometries in database



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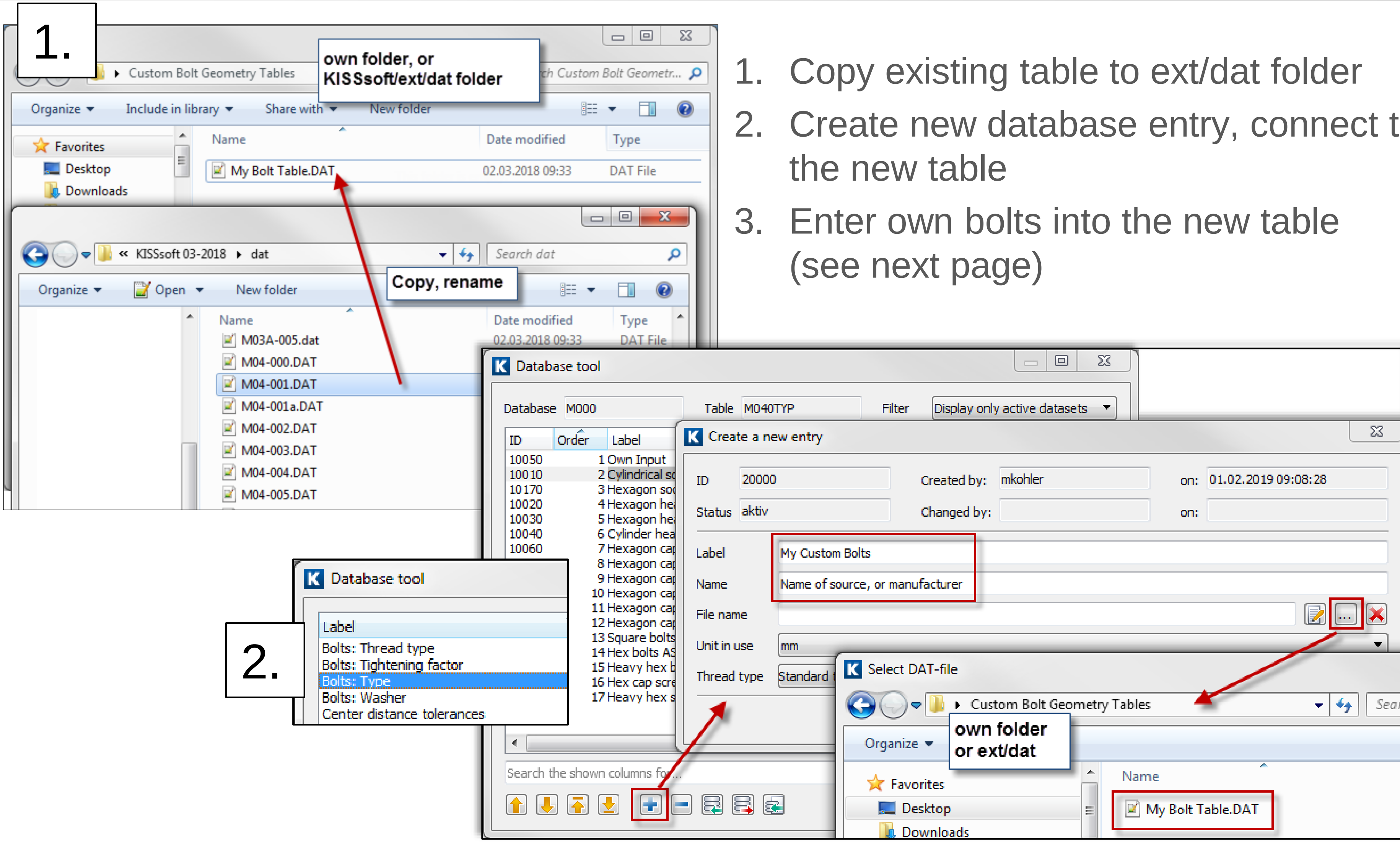
Bolt geometries in database – new entry (1/2)



1. Copy existing table to ext/dat folder
2. Create new database entry, connect to the new table
3. Enter own bolts into the new table (see next page)

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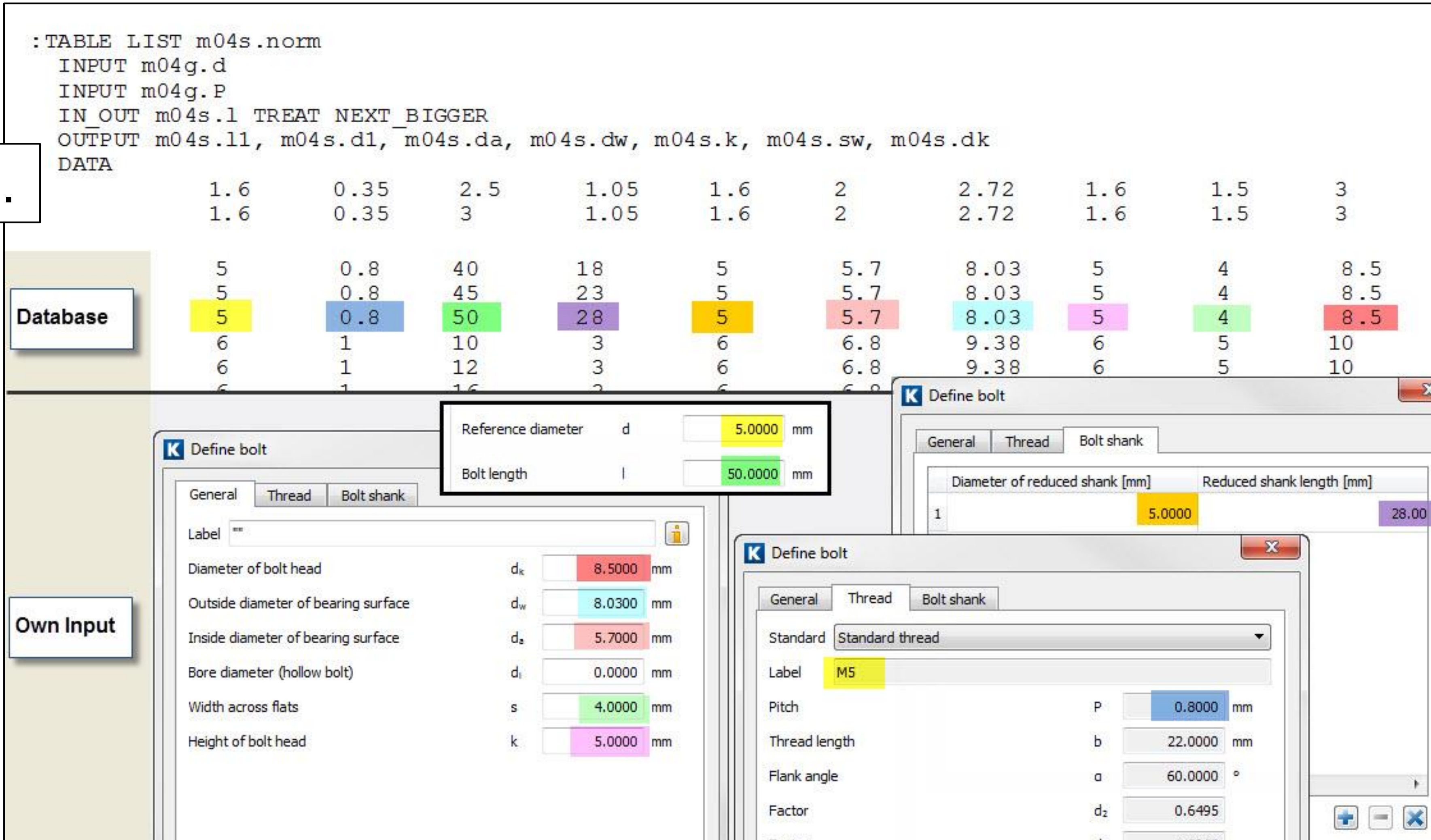
Bolt geometries in database – new entry (2/2)

3.



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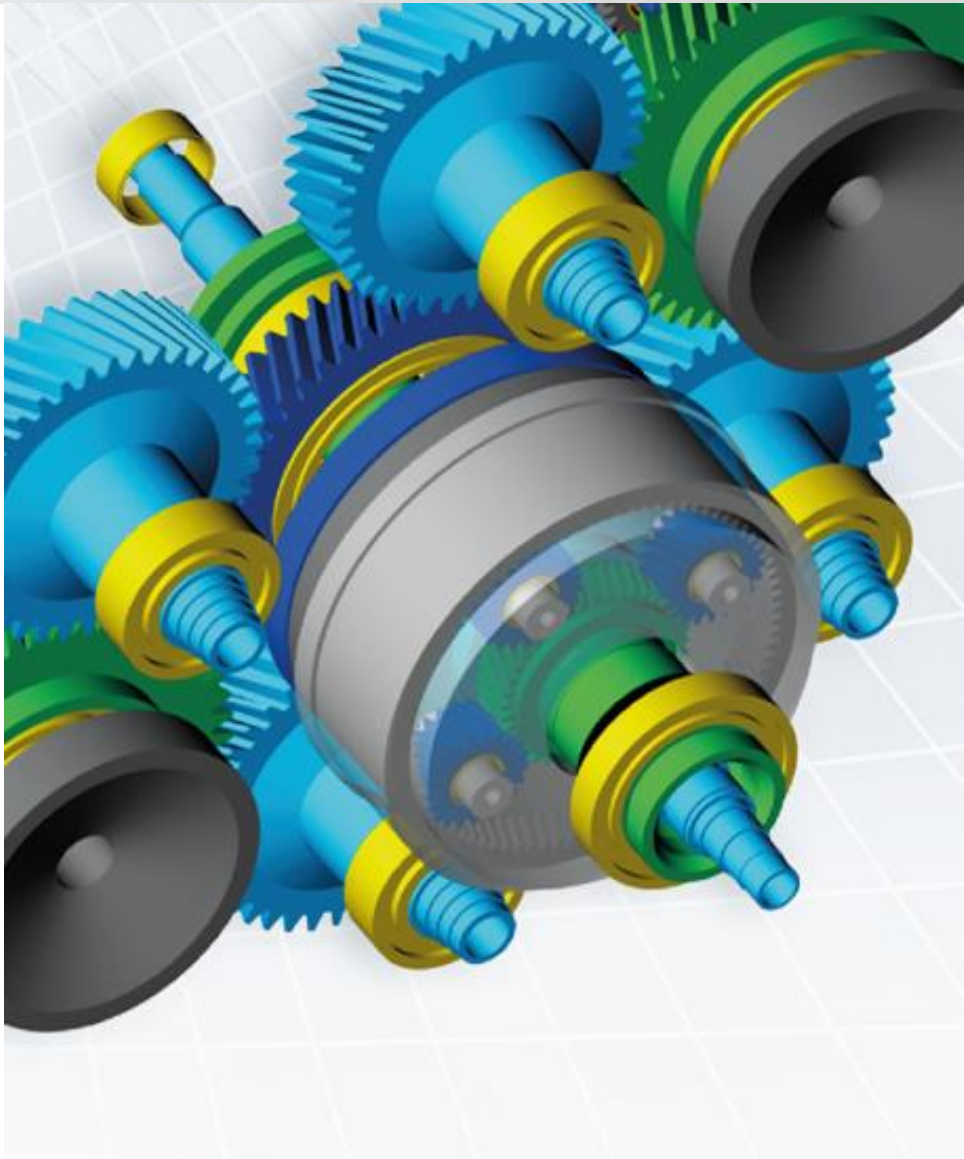
Thank you for your attention!



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