Automated procedure for determination of friction grip and bearing joint strength used for space missions

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Abstract: This paper presents an algorithm that calculates the nominal torque for friction grip joints, followed by its implementation in a Microsoft Excel worksheet that is automated by a macro written in Visual Basic. The algorithm outputs not only the optimum nominal torque (the value for which the margin of safety is maximum) but also the interval of acceptable values so that the user can easily choose the torque according to his preferences.

Key Words: Excel, Visual, Basic, preload, torque, friction, grip

1. INTRODUCTION

The following chapter contains a fastener stress analysis for the design solution, according to the main requirements from [2]. This algorithm is the basis for an Excel worksheet that is used to calculate the nominal torque and associated output values and it is automated using a macro written in Visual Basic.

2. THE ALGORITHM FOR FRICTION GRIP JOINTS

The fastener analysis is made according to [1], [2], [3], [4], [5], [6], [7].

Fastener geometrical characteristics:

The following thread parameters can be calculated using the standard relations for MJ threads (ISO 5855-1):

Table 1. MJ Thread Parameters

Symbol	Unit	Description	Equation
p	-	pitch of the thread	-
H	mm	height of fundamental triangle	$0.866025 \cdot p$
d	mm	nominal fastener diameter	-
d0	mm	diameter of the smallest cross-section of fastener	-
d2	mm	pitch diameter	$d - 2 \cdot 0.375 \cdot H$
d3	mm	minor diameter of the fastener thread	$d - 2 \cdot 0.5625 \cdot H$
dS	mm	diameter used for stress calculation for MJ thread	$d_3 \sqrt{2 - \left(\dfrac{d_3}{d_2}\right)^2}$

dhead	mm	diameter of fastener head	-
Dh	mm	hole diameter of clamped parts	-
duh	mm	outside diameter at which under-head or under-nut frictional forces act	$0.5 \cdot (d_{head} + D_h)$
Anom	mm^2	nominal cross-section area	$0.25 \cdot \pi \cdot d^2$
A0	mm^2	minimum cross-section area	$0.25 \cdot \pi \cdot d_0^2$
A3	mm^2	cross-sectional area at minor diameter of fastener thread	$0.25 \cdot \pi \cdot d_3^2$
As	mm^2	stress area calculation	$0.25 \cdot \pi \cdot d_s^2$

The following thread parameters can be calculated using the standard relations for M threads taken from table 5-3 from [1]:

Table 2. M Thread Parameters

Symbol	Unit	Description	Equation
p	-	pitch of the thread	-
H	mm	height of fundamental triangle	$0.866025 \cdot p$
d	mm	nominal fastener diameter	-
d0	mm	diameter of the smallest cross-section of fastener	-
d2	mm	pitch diameter	$d - 0.64952 \cdot p$
d3	mm	minor diameter of the fastener thread	$d - 1.22687 \cdot p$
dS	mm	diameter used for stress calculation for M thread	$0.5 \cdot (d_2 + d_3)$
dhead	mm	diameter of fastener head	-
Dh	mm	hole diameter of clamped parts	-
duh	mm	outside diameter at which under-head or under-nut frictional forces act	$0.5 \cdot (d_{head} + D_h)$
Anom	mm^2	nominal cross-section area	$0.25 \cdot \pi \cdot d^2$
A0	mm^2	minimum cross-section area	$0.25 \cdot \pi \cdot d_0^2$
A3	mm^2	cross-sectional area at minor diameter of fastener thread	$0.25 \cdot \pi \cdot d_3^2$
As	mm^2	stress area calculation	$0.25 \cdot \pi \cdot d_s^2$

Required minimum clamp load:

The required minimum clamp load $F_{K,req} = 0$ – OHB suggestion.

The compliance calculation for fastener:

The joint is nut-tightened and has a cylindrical fastener head. Therefore, the appropriate substitution lengths from Table 7-1 [1] and their representation are:

- $L_{h,sub} = 0.4 \cdot d$ – substitutional extension length of the bolt cylindrical head, [mm];
- $L_{h,sub} = 0.5 \cdot d$ – for hexagonal head of the bolt, [mm];
- $L_{eng,sub} = 0.33 \cdot d$ - substitutional extension length of the engaged shank for threaded hole, [mm];
- $L_{eng,sub} = 0.4 \cdot d$ – for nut-tightened, [mm];
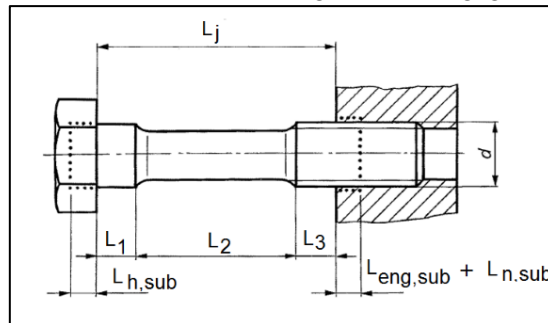- $L_{n,sub} = 0.4 \cdot d$ - substitutional extension length of the engaged nut or insert, [mm].

Figure 1. Division of a bolt into individual deformation regions [2]

Using these substitution lengths, the fastener's compliance can be calculated using the equation 7.5.5 [1]:

$$\delta_b = \frac{1}{E_b}\left(\frac{L_{h,sub}}{A_{nom}} + \frac{L_{eng,sub}}{A_3} + \sum_i \frac{L_{sha,i}}{A_{sha,i}}\right) + \frac{L_{n,sub}}{E_n A_{nom}} \tag{1}$$

Where:
- En - modulus of elasticity for the nut or the tapped thread region [MPa]
- Eb - modulus of elasticity for the fastener [MPa]

The compliance calculation for the clamped parts:

The compliance of the clamed parts is more complicated due to the 3 - dimensional state of stress state in the joint that is introduced by the preload.

The limiting diameter that serve to settle the question as to whether a deformation sleeve is present, is given by [1]:

$$D_{lim} = D_{uh,brg} + w \cdot L_c \cdot \tan\phi \tag{2}$$

Where:	
- w = 2 for a threaded hole joint and w = 1 for a nut-tightened joint	- Davail = 2·min (a1; a2) - available diameter for compression cone [mm]
- $D_{uh,brg}$ - outside diameter of the plane head bearing surface of the bolt or nut [mm]	- Lc - length of the clamped parts [mm]
- ϕ – compression cone's half angle, [1]	- Duh, brg, - outer diameter of the under-head or under-nut bearing surface [mm]
- for nut-tightened joint: $\tan\phi = 0.362 + 0.032 \ln\frac{x}{2} + 0.153 \ln y$	
- for insert joints: $\tan\phi = 1.295 - 0.246 \ln x + 0.94 \ln y$	

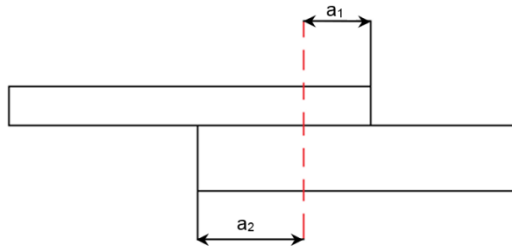| Where:	
$x = \dfrac{L_c}{D_{uh,brg}}$	$y = \dfrac{D_{avail}}{D_{uh,brg}}$

Figure 2. Minimum edge distance from central axis of bolt

The existence of the sleeve is determined as follows:

Case 1: If $D_{avail} \geq D_{lim}$, the compression zone is fully developed into a cone. The compliance of the clamped parts, δ_c, is given by:

$$\delta_c = \frac{2 \ln\left[\dfrac{(D_{uh,brg}+D_h)(D_{lim}-D_h)}{(D_{uh,brg}-D_h)(D_{lim}+D_h)}\right]}{w \cdot E_c \cdot \pi \cdot D_h \cdot \tan\phi} \tag{3}$$

Where:
- Ec - modulus of elasticity of the clamped parts [MPa]

Case 2: If $D_{uh,brg} < D_{avail} < D_{lim}$, the compression zone is formed by a partial compression cone and compression sleeve. The compliance of the clamped parts is given by:

$$\delta_c = \frac{\dfrac{2}{w \cdot D_h \cdot \tan\phi} \ln\left[\dfrac{(D_{uh,brg}+D_h)(D_{avail}-D_h)}{(D_{uh,brg}-D_h)(D_{avail}+D_h)}\right] + \dfrac{4}{D_{avail}^2-D_h^2}\left[L_c - \dfrac{D_{avail}-D_{uh,brg}}{w \cdot \tan\phi}\right]}{E_c \cdot \pi} \qquad (4)$$

Where:
- E_c - modulus of elasticity of the clamped parts [MPa]

Figure 3. Case 1

L_c

$L_c = 2*(L_{cone} + L_{sleeve})$
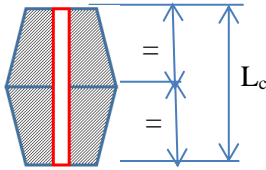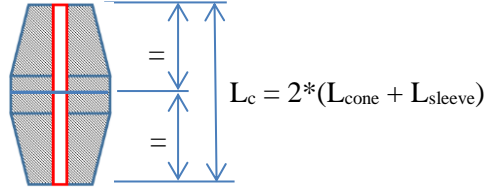
Figure 4. Case 2

The equation above includes the effect of both the compression cone and sleeve and is appropriate when the clamped parts have the same modulus of elasticity [8].

If the compliances of the cone and the sleeve have to be calculated separately (for example, when multiple materials are present in the clamped parts), than the conformity of the clamping parts compliance will be calculated by [2]:

$$\delta_c = \sum_{i=1}^{j} \delta_{cone} + \sum_{i=j+1}^{m} \delta_{sleeve} \qquad (5)$$

The *j* and *m* are determined in the following mode: the compression diagram of the clamped parts has to be overlapped onto longitudinal section joint (on the distance L_c). This way the number of *j* segments of different material in cone zone and (*m-j*) segments of different material in sleeve zone can be calculated.

The cone's compliance is [1]:

$$\delta_{cone} = \frac{\ln\left[\dfrac{(D_{uh,brg}+D_h)(D_{uh,brg}+2\cdot L_{cone}\cdot \tan(\phi)-D_h)}{(D_{uh,brg}-D_h)(D_{uh,brg}+2\cdot L_{cone}\cdot \tan(\phi)+D_h)}\right]}{E_c \cdot D_h \cdot \pi \cdot \tan(\phi)} \qquad (6)$$

and the sleeve's compliance is [1]:

$$\delta_{sleeve} = \frac{4\cdot L_{sleeve}}{E_c \cdot \pi (D_{avail}^2 - D_h^2)} \qquad (7)$$

The lengths of the cone and the sleeve are given by [1]:

$$L_{cone} = \frac{D_{avail}-D_{uh,brg}}{2\cdot \tan(\phi)} \qquad (8)$$

$$L_{sleeve} = \frac{L_C}{2} - L_{cone} \qquad (9)$$

Case 3: If $D_{avail} \leq D_{uh,brg}$, the compression zone is fully developed into a sleeve:
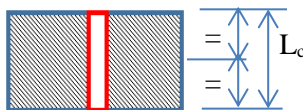

Figure 5. Case 3

For the case where only a compression sleeve exists, the last equation above is used for the sleeve compliance and L_{sleeve} is equal to the combined length of the clamped parts L_c ($L_{sleeve}=L_c$).
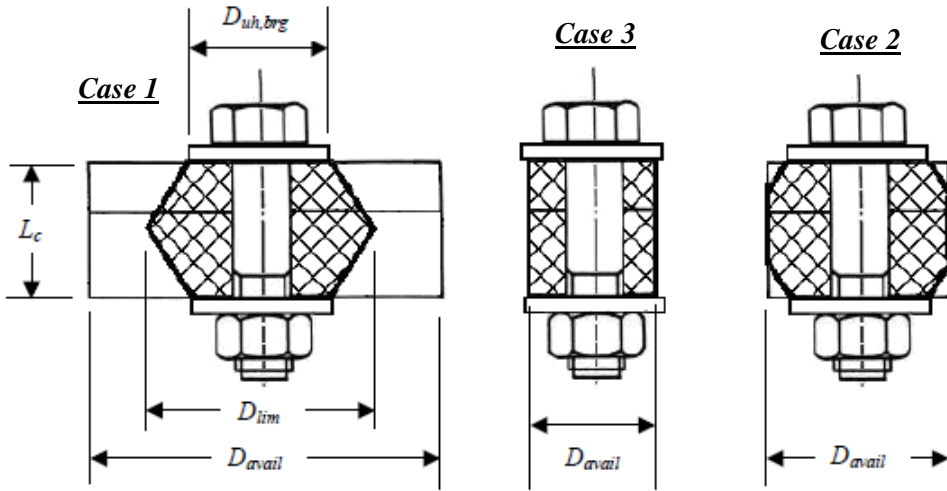

Figure 6. Compression zones in cylindrical clamped parts

The force ratio:

For the concentric axially loaded joints, with a loading plane factor $n<1$, the working loads taken from chapter 7.4 are:

$$F_A = \frac{\Delta F_{b,A}}{\Phi_n}$$

(10)

$$\Delta F_{b,A} = \Phi_n \cdot F_A \text{ and } \Delta F_{c,A} = (1-\Phi_n) \cdot F_A$$

Where:
- $\Delta F_{c,A}$ – increment in clamped parts decompression caused by the external axial load F_A [N]
- $\Delta F_{b,A}$ - increment in fastener tension caused by the external axial load F_A (taken from FEM)

The force ratio for a concentric axially loaded joint, including the loading plane factor effect is calculated with the formula below:

$$\Phi_n = n\left(\frac{\delta_c}{\delta_c+\delta_b}\right)$$

(11)

- along with the condition of equilibrium according to figure 7-6 [1] and Figure 7:

$$F_A = \Delta F_{b,A} + \Delta F_{c,A}$$

(12)

Preload loss due to embedding:

The nominal preload in the fastener F_V can be calculated using the following formula:

$$F_V = \gamma \cdot \sigma_y \cdot A_S = \sigma_V \cdot A_S \text{ [N]}$$

(13)

Where:
- σ_V – nominal pretension stress [MPa]
- σ_y – yield stress of the fastener material [MPa]
- $\gamma = 0.7$ – ratio preload stress to yield stress

The preload loss due to embedding is assumed to be 5% of the maximum preload:

$$F_z = 0.05 \cdot F_V, \text{ [N]}$$

(14)

Thermal induced loads:

The extreme values of the thermally induced force are given below, assuming that the clamped parts are made from a single material type:

$$F_{\Delta T^{\pm}} = (\alpha_c - \alpha_b) \cdot \Delta T^{\pm} \cdot E_b \cdot A_{sm} \cdot (1 - \Phi_n) \ [N] \tag{15}$$

Where:

- $A_{sm} = 0.25 \cdot \pi \cdot d_{sm}{}^2$ - the stiffness area of the fastener [mm^2]
- $d_{sm} = d_3$ [mm]
- α_b, α_c – coefficients of thermal expansion of the fastener and clamped parts, respectively [K^{-1}]

$$\Delta T^+ = T_{working,max} - T_{reference} \tag{16}$$

$$\Delta T^- = T_{working,min} - T_{reference} \tag{17}$$

Where:

- $T_{working,max}$ = 42.0 C for RCS Brackets
- $T_{working,min}$ = -34 C for RCS Brackets
- $T_{working,max}$ = 35.0 C for MPS Brackets
- $T_{working,min}$ = -36 C for MPS Brackets
- $T_{reference}$ = 21 C

If the clamped parts have multiple material types, a more general form of equation is used to account for the differing thermal expansion coefficients.

$$F_{\Delta T^{\pm}} = \frac{\left(\sum_{i=1}^{m} \alpha_{c,i} \cdot L_{c,i}\right) - \alpha_b \cdot L_b}{L_c} \cdot \Delta T^{\pm} \cdot E_b \cdot A_{sm} \cdot (1 - \Phi_n) \tag{18}$$

Maximum and minimum preload:

The maximum and minimum bound of the applied torque $M_{app,min}$ and $M_{app,max}$ are determined by:

$$M_{app,min} = (1 - \omega) \cdot M_{nom} \tag{19}$$

$$M_{app,max} = (1 + \omega) \cdot M_{nom} \tag{20}$$

where M_{nom} is the nominal applied torque (normally determined by an iterative process) first taken from annex A1 [1], and ω is the torque range accuracy chosen 10% chapter 6.3.2.2 [1].

The maximum and minimum preloads are found by the following equations:

$$F_{V,min} = \frac{M_{app,min} - M_{P,max}}{\frac{1}{2} d_2 \left(\tan\varphi + \frac{\mu_{th,max}}{\cos\theta}\right) + \frac{1}{2} d_{uh} \cdot \mu_{uh,max}} + F_{\Delta T} - F_Z \ , \ [N] \tag{21}$$

$$F_{V,max} = \frac{M_{app,max} - M_{P,min}}{\frac{1}{2} d_2 \left(\tan\varphi + \frac{\mu_{th,min}}{\cos\theta}\right) + \frac{1}{2} d_{uh} \cdot \mu_{uh,min}} + F_{\Delta T^+} \ , \ [N] \tag{22}$$

$$\tan\varphi = \frac{p}{\pi \cdot d_2} \tag{23}$$

Where:

- φ – thread helix angle
- $\theta = 30°$ - half angle of thread grooves for common case of Unified of Metric threads

- μ_{th} – coefficient of friction of the thread interface taken from Annex B [1]
- μ_{uh} – coefficient of friction between the nut (or fastener head) and the adjacent clamped part taken from Annex C [1]
- M_P - the prevailing torque of the locking device taken form Table 6-2 [1] [N·mm]

Combined stress in fastener during tightening:

The stress occurring during the tightening process is composed of tension caused by the increasing preload and a shear stress caused by the application of the torque.

These two stresses are combined by the shape deformation energy hypothesis according to Von Mises equation:

$$\sigma_{v.m.} = \sqrt{\sigma_{V,max}^2 + 3 \cdot \tau_{max}^2} \quad [\text{MPa}] \tag{24}$$

Due to the equilibrium of moments, the maximum shear stress within the fastener's shank is related to the moments due to under-head friction and the externally applied torque by:

$$\tau_{max} = \frac{M_{app,max} - M_{uh,min}}{W_p} \quad [\text{N}] \tag{25}$$

Where:

- $M_{uh,min} = \frac{d_{uh}}{2}\left(F_{V,max} - F_{\Delta T^+}\right)\mu_{uh,min}$ – minimum under-head frictional torque [N·mm]
- $W_p = \frac{\pi d_0^3}{12}$ – polar section modulus [mm³]

If the shear stress in the fastener exceeds its yield point, a state of total plastic deformation is reached, whereby the torsional stresses through the cross section are constant.

If this happens the polar section modulus formula is replaced by: $W_p = \frac{\pi d_0^3}{12}$.

The maximum pretension stress is given by:

$$\sigma_{V,max} = \frac{F_{V,max} + F_{\Delta T^+}}{A_0} \quad [\text{MPa}] \tag{26}$$

The margins of safety on tightening are calculated according to:

$$MoS_{ti,y} = \frac{\sigma_{ty}}{\sigma_{v.m.}} - 1 \tag{27}$$

$$MoS_{ti,u} = \frac{\sigma_{tu}}{\sigma_{v.m.}} - 1 \tag{28}$$

No factor of safety is used since all uncertainty sources are already included in $F_{V,\,max}$.

Joint separation (or gapping):

The MoS for joint separation is not intended to cover the case of slipping of a friction grip joint and it is given by:

$$MoS_{sep} = \frac{F_{V,min} - F_{K,req}}{(1 - \Phi_n) \cdot F_A \cdot sf_{sep}} - 1 \tag{29}$$

Where:

- $sf_{sep} = 1.4$ – safety factor for joint separation taken from Table 5-4 [1]

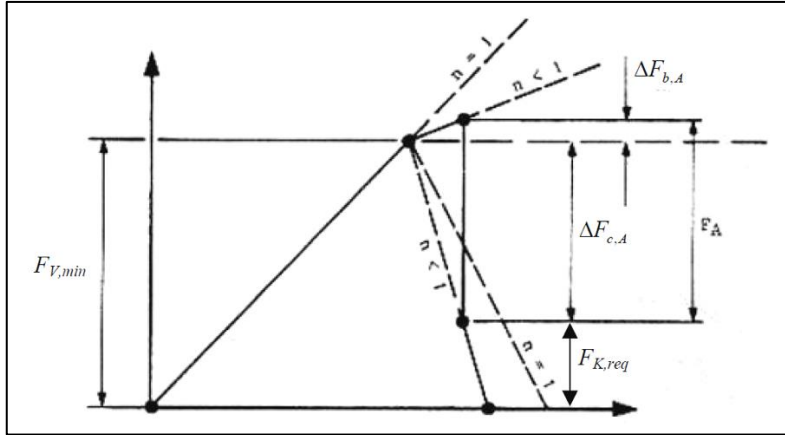The figure below shows the joint diagram at the onset of the separation failure:

Figure 7. Joint diagram at the onset of the separation failure, [1]

Fastener failure due to preload and external load:

Using the criteria for axial load analysis from Table 7-3 [1], the mode of shear load transfer is chosen to be "Friction Grip", so the failure criterion uses overall load F_{tot} in case of "no gapping" and this is given by:

$$F_{tot}=F_{V,max}+\Delta F_{b,A} \ [N] \tag{30}$$

The margins of safety for the overall load are calculated according to:

$$MoS_{tot,b,u}=\frac{A_S \cdot \sigma_u}{F_{V,max}+\Delta F_{b,A} \cdot FOSU \cdot KLD}-1 \tag{31}$$

$$MoS_{tot,b,y}=\frac{A_S \cdot \sigma_y}{F_{V,max}+\Delta F_{b,A} \cdot FOSY \cdot KLD}-1 \tag{32}$$

Where:

- FOSY=1.25 – yield design factor of safety
- FOSU=2 – ultimate design factor of safety
- KLD=1.2 – fitting factor, factor used to take into account local discontinuities and applied in series with FOSU and FOSY

Thread failure by shear pull-out:

In this chapter, the thread failure by shear pull-out is determined for the weakest material between the male and female thread.

Failure of the female thread (nut):

The critical fastener load for failure of the female thread is given by the next equation:

$$F_{ult,n} = \tau_{ult,n} \cdot A_{th,n} \cdot c_1 \cdot c_2 \ [N] \tag{33}$$

Where:

- $\tau_{ult,n}$ - ultimate tensile shear strength of the material forming the female thread [MPa]
- $A_{th,n}$ - surface area around the female thread that is assumed to fail during thread pull-out [mm²];
- c_1 - coefficient for the depth of supporting material behind the female thread
- c_2 - coefficient for the strength ratio of the two threads

The female thread's failure surface area is calculated by the following equation:

$$A_{th,n} = \pi \cdot d \cdot \left(\frac{L_{eng,eff,n}}{p}\right) \cdot \left[\frac{p}{2} + (d - D_2) \cdot \tan(\theta)\right] \tag{34}$$

Where:
- D_2- pitch diameter of the female thread (equal to d_2 of the fastener) [mm]
- $L_{eng,eff,n} = L_{eng,n} - 0.8 \cdot p$ – effective length of engaged thread [mm]
- $L_{eng,n} = L_n$- is the length of the nut (see figure below) [mm]

In nut-tightened joints, the full length of the female thread is normally engaged, therefore $L_{eng,n} = L_n$.

The coefficient c_1 is calculated below:

$$c_1 = 3.8 \cdot \left(\frac{s_w}{d}\right) - \left(\frac{s_w}{d}\right)^2 - 2.61 \tag{35}$$

Where:
- s_w - the wrench size as indicated in figure above [mm]

The equation for c_1 is valid only for metric threads with $1.4 \leq \left(\frac{s_w}{d}\right) \leq 1.9$.

The coefficient c_2 is given by:
- for $R_S \geq 1$: $c_2 = 0.897$
- for $0.4 < R_S < 1$: $c_2 = 0.728 + 1.769 \cdot R_S - 2.896 \cdot R_S^2 + 1.296 \cdot R_S^3$

where R_S is the shear strength ratio of the female and male threads according to:

$$R_S = \frac{\tau_{ult,n} \cdot A_{th,n}}{\tau_{ult,b} \cdot A_{th,b}} \tag{36}$$

Where:
- s_w - the wrench size as indicated in figure above [mm]
- $\tau_{ult,b}$ [MPa] is the ultimate shear strength of the material forming the male thread
- $A_{th,b}$ [mm^2] is the failure surface area of the male thread

$$A_{th,b} = \pi \cdot D_1 \cdot \left(\frac{L_{eng,eff,b}}{p}\right) \cdot \left[\frac{p}{2} + (d_2 - D_1) \cdot \tan(\theta)\right] \tag{37}$$

Where:
- D_1- minor diameter of the female thread (equal to that of the fastener, d_3) [mm]
- d_2 – pitch diameter of the male thread [mm]
- $L_{eng,eff,b} = L_{eng,b} - 0.8 \cdot p$ – effective length of engaged thread [mm]
- $L_{eng,b} = L_n$- is the length of the nut head (see figure above) [mm]

If both male and female threads are made of steel, R_S is calculated with the following formula:

$$R_S = \frac{\sigma_{ult,n} \cdot A_{th,n}}{\sigma_{ult,b} \cdot A_{th,b}} \tag{38}$$

Note:
$\sigma_{ult,n}$ is ultimate tensile strength of the female material [MPa].
$\sigma_{ult,b}$ is ultimate tensile strength of the male material [MPa].

Failure of the male thread (screw):
If the fastener's material is weaker than the nut or insert, the analysis is the same, and the strength of the male thread is calculated by:

$$F_{ult,b} = \tau_{ult,b} \cdot A_{th,b} \cdot c_1 \cdot c_2 \tag{39}$$

The margin of safety on shear pull-out of the thread under the external load is calculated by:

$$MoS_{th,A} = \frac{F_{ult}}{F_A \cdot FOSU \cdot KLD} - 1 \qquad \text{-for the external load} \tag{40}$$

$$MoS_{th,tot} = \frac{F_{ult}}{F_{V,max} + \Delta F_{b,A} \cdot FOSU \cdot KLD} - 1 \qquad \text{-for the overall load} \tag{41}$$

Where:
-    $F_{ult}=min(F_{ult.n}, F_{ult,b})$- thread failure load [N]

Note: No MoS calculations are necessary for yield failure of the thread.

Crushing of flanges:

The margins of safety on flange crushing are calculated using the following equations:

$$MoS_{crush,y} = \frac{\sigma_{bry}}{\sigma_{uh,max} \cdot FOSY \cdot KLD} - 1 \tag{42}$$

$$MoS_{crush,u} = \frac{\sigma_{bru}}{\sigma_{uh,max} \cdot FOSU \cdot KLD} - 1 \tag{43}$$

Where:
-    $\sigma_{uh,max}$ is the maximum compressive stress that occurs under the head/nut:
-    $\sigma_{uh,max} = \frac{F_{tot}}{k_{int} \cdot A_{uh}}$ [MPa]
-    A_{uh} - area under the head/nut or washer that can be crushed [mm^2]

The flange with the weakest material out of all the flange packages has to be chosen (k_{int} is the number of interfaces and must be maximum 2).
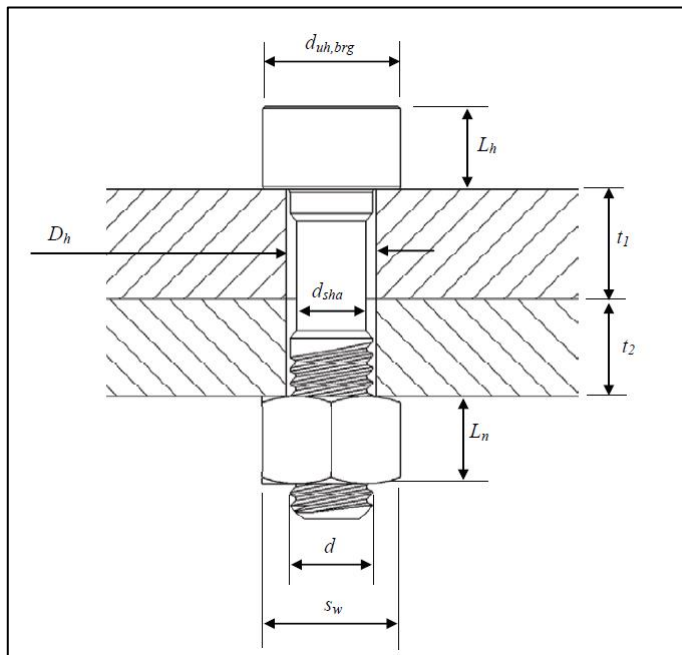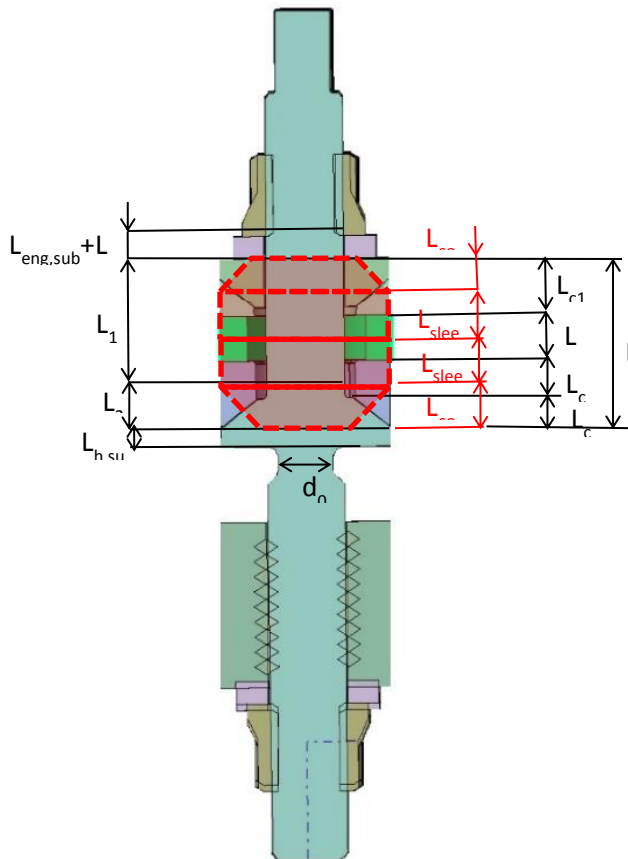

Figure 8. Fastener dimensions, [1]

Figure 9. The geometrical length for compliance calculation and compression zone

3. THE EXCEL VISUAL BASIC MACRO

The algorithm has been implemented in an Excel spreadsheet and automated using Visual Basic [9]. After the inputs have been written to the spreadsheet (material properties, bolt properties, loads, etc.) the user will access the 'MOS' sheet as seen in Figure 10.

A green box with ActiveX controls will be found, see Figure 11. From this box the user will press the 'Calculate Mnom' button. This button will start the macro that calculates the nominal torque and all the associated data. After the calculation is done the final data will be displayed as follows:
-    The interval of accepted values for the nominal torque are displayed under the 'From' and 'To' cells, with the option to use the ActiveX slider control to change the value to the one preferred by the user. The value chosen by the program is the value for which the margin of safety is maximum within that interval.
-    The value of the nominal torque and its associated variables will be displayed in the 'MOS' sheet, see Figure 12.
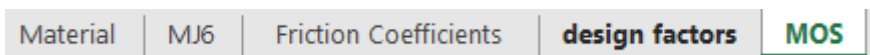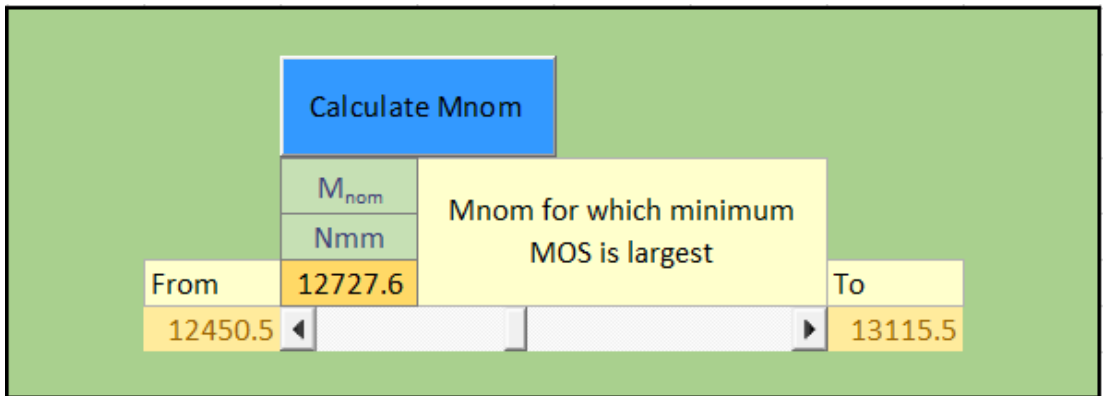

Figure 10. List of worksheets

Figure 11. User box for starting the calculation process

M_{nom}	ω	$M_{app,min}$	$M_{app,max}$	$M_{P,min}$	$M_{P,max}$	$\tan\varphi$	$\cos\theta$	$F_{V,min}$	$F_{V,max}$	F_{tot}	$M_{uh,min}$
Nmm		Nmm	Nmm	Nmm	Nmm			N	N	N	Nmm
12727.6	0.1	11455	14000	500	1000	0.05949	0.86603	3936.1	14086.7	14186.6	6972.88
	10%										
firstly taken from Anexa A1 [ECCS-E-HB-32-23A] or Anexa A1 [VDI 2230 Part 1]				prevailing torque of locking devices on unlubricated fasteners taken from Table 6-2 [ECCS-E-HB-32-23A]							

Figure 12. Detailed output data

The source code of the Excel Visual Basic macro is listed below. A brief outline of the code will be presented:

1. The program starts by finding the edges of the first interval of acceptable values. In order to minimize the execution time, a starting global interval is chosen, in this case it ranges from 0 to 50000. This interval is searched for domains where the ends have opposite signs using a starting coarse granularity of 50000. If no valid intervals have been found, then the granularity is halved, and the search starts again. This is done until a valid first interval is found.
2. A bisection scheme is used to find the solution within this first domain. This will be the lower end of the final interval of accepted values.
3. Step 1 is repeated for the next valid domain where a solution exists.
4. Step 2 is repeated for the higher end of final interval of accepted values.
5. The extremum within this interval is found using a bisection method. The derivatives are calculated numerically using a fixed step size, in this case 10.
6. The final solution is written to the 'MOS' worksheet.

```
1.  Private Sub CommandButton1_Click()
2.      Application.ScreenUpdating = False
3.
4.      granularity = 50000
5.      interval_start = 0
```

```vba
6.       interval_end = 50000
7.
8.       'Minimum value determination
9.       Range("Y4").value = -1
10.      Do Until Range("Y4").value <> -1 Or granularity < 100
11.          granularity = granularity / 2
12.          Range("Y4").value = return_interval(interval_start, interval_end, g
   ranularity)
13.      Loop
14.      interval_start = Range("Y4").value + granularity
15.      Range("AC22").value = return_solution(Range("Y4").value, Range("Y4").va
   lue + granularity, 0.1)
16.
17.      'Maximum value determination
18.      Range("Y4").value = -1
19.      Do Until Range("Y4").value <> -1 Or granularity < 100
20.          granularity = granularity / 2
21.          Range("Y4").value = return_interval(interval_start, interval_end, g
   ranularity)
22.      Loop
23.      interval_start = Range("Y4").value + granularity
24.      Range("AH22").value = return_solution(Range("Y4").value, Range("Y4").va
   lue + granularity, 0.1)
25.      Range("Y4").value = return_optimum(Range("AC22").value, Range("AH22").v
   alue, 0.1, 10)
26.
27.      ScrollBar1.Min = Range("AC22").value + 1
28.      ScrollBar1.Max = Range("AH22").value - 1
29.      ScrollBar1.value = Range("Y4").value
30.      Application.ScreenUpdating = True
31. End Sub
32.
33. Function return_solution(int_start As Double, int_end As Double, int_granul
   arity As Double) As Double
34.      While (int_end - int_start >= int_granularity)
35.          Range("Y4").value = int_start
36.          Application.Calculate
37.          val1 = Application.WorksheetFunction.Min(Range("L3:T230"))
38.
39.          Range("Y4").value = (int_start + int_end) / 2
40.          Application.Calculate
41.          val2 = Application.WorksheetFunction.Min(Range("L3:T230"))
42.
43.          If val1 * val2 < 0 Then
44.              int_end = (int_start + int_end) / 2
45.          Else
46.              int_start = (int_start + int_end) / 2
47.          End If
48.      Wend
49.      return_solution = (int_start + int_end) / 2
50. End Function
51. Function return_optimum(int_start As Double, int_end As Double, int_granula
   rity As Double, step As Double) As Double
52.      While (int_end - int_start >= int_granularity)
53.          Range("Y4").value = int_start - step
54.          Application.Calculate
55.          val1 = Application.WorksheetFunction.Min(Range("L3:T230"))
```

```
56.          Range("Y4").value = int_start + step
57.          Application.Calculate
58.          val2 = Application.WorksheetFunction.Min(Range("L3:T230"))
59.          f0 = (val2 - val1) / (2 * step)
60.
61.          Range("Y4").value = (int_start + int_end) / 2 - step
62.          Application.Calculate
63.          val1 = Application.WorksheetFunction.Min(Range("L3:T230"))
64.          Range("Y4").value = (int_start + int_end) / 2 + step
65.          Application.Calculate
66.          val2 = Application.WorksheetFunction.Min(Range("L3:T230"))
67.          f1 = (val2 - val1) / (2 * step)
68.
69.          If f0 * f1 < 0 Then
70.              int_end = (int_start + int_end) / 2
71.          Else
72.              int_start = (int_start + int_end) / 2
73.          End If
74.      Wend
75.      return_optimum = (int_start + int_end) / 2
76. End Function
```

4. CONCLUSIONS

The algorithm provides a quick overview of the acceptable values for the nominal torque, thus giving the user the possibility of changing the bolt design on-the-fly not only in the calculation phase of the project, but also on the spot in case of an emergency.

The original contributions of the authors are:
- creating and designing the excel worksheet for the algorithm presented in the previous chapters,
- automatizing the algorithm using a Visual Basic Excel macro as presented in chapter 3.

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