BAT - Bolt Analysis Tool

User Manual

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Symbols and Abbreviations

Symbols

α_A	tightening factor
α_b	coeff. of lin. thermal expansion of the bolt
α_c	coeff. of lin. thermal expansion of the clamped part (plate)
δ_b	elastic compliance of the bolt
δ_c	elastic compliance of the clamped part (plate)
λ	under-head bearing angle of bolt
μ_{th}	coeff. of friction in bolt thread
μ_{uh}	coeff. of friction under bolt head
ν	bolt utilization factor
φ	helix angle / slope of bolt thread
Φ	load factor of concentric joint (also: force ratio or relative compliance factor)
Φ_n	load factor for concentric clamping and concentric force load introduction via the clamped parts
ρ	friction angle in bolt thread
σ_n	normal stress in the bolt
σ_v	von-Mises stress in the bolt
τ	shear stress in the bolt
A_1	nominal cross section of threaded bolt
A_3	minimal thread cross section
A_p	pitch cross section of threaded bolt
A_s	stress cross section of threaded bolt
d	nominal threaded bolt diameter
d_2	pitch diameter of threaded bolt
d_3	minimal diameter of threaded bolt
d_h	minimal contact diameter under bolt head
d_s	stress diameter of threaded bolt
F_A	external, axial bolt load
F_M	preload after tightening / assembly preload
F_{PA}	additional axial plate load
F_Q	external, shear bolt load

F_{SA}	additional axial bolt load
F_V	service preload incl. embedding and thermal influence
f_Z	plastic deformation due to embeddding
F_Z	preload loss due to embedding
l_K	joint clamped length
M_p	prevailing torque of bolt locking device
n	load introduction factor
p	pitch of bolt thread

Abbreviations

BAT	Bolt Analysis Tool
CTE	Coefficient of Thermal Expansion
TBJ	Through-Bolt Joint
TTJ	Tapped Thread Joint

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1 Introduction

This document will include the BAT (Bolt Analysis Tool) User Manual [1] [2] [3].

2 Bolt and Thread Geometry

D_{Km} is the *effective diameter of under head/nut friction torque* and is defined by

$$D_{Km} = \frac{D_{hole} + d_h}{2} \tag{2.1}$$

where D_{hole} is the *through-hole diameter* in the clamped parts and d_h is the *minimum bearing surface outer diamter* of the bolt head or nut. An other input value to calculate the under-head friction torque (4.2) is the *under head bearing angle* λ seen in Figure 2.1.

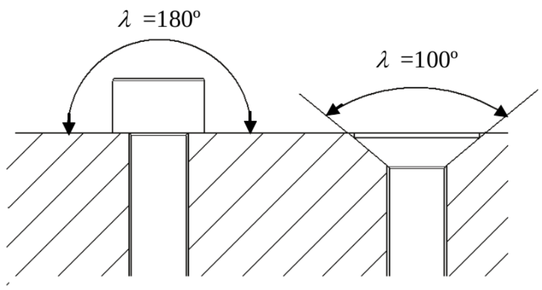

Figure 2.1: Definition of under head bearing angle [2]

3 Joint Diagram

The *joint diagram* seen in Figure 3.1 [3] visualizes the forces and displacements and helps to understand the loading conditions of a concentrically loaded bolted joint.

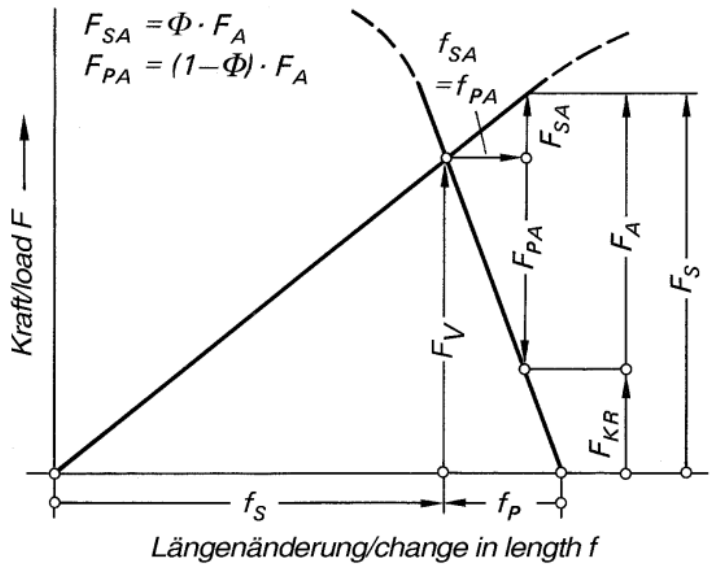

Figure 3.1: Joint diagram for the working state of a concentrically loaded bolted joint with $n = 1$ [3]

The preload force F_V (see §4.1) compresses the clamped parts f_P and elongates the bolt f_S during tightening. If an external load F_A is acting on the bolted joint the bolt is streched much further and the clamped parts are relaxed ($f_{SA} = f_{PA}$) due to the loading. The *additional axial bolt load* F_{SA} and the *additional axial plate load*[1] F_{PA} are given to

$$F_{SA} = n\Phi F_A, \qquad F_{PA} = (1 - n\Phi)F_A \tag{3.1}$$

where Φ is the load factor, which is defined as the quotient of the additional bolt load F_{SA} and the axial working load component F_A. n is the *load introduction*

[1] If the bolt is loaded in tension F_{PA} is better described as an *axial plate relaxation force*.

factor and is crucial for determining the size of the additional botl loads. As seen in Figure 3.1 F_{KR} is the residual clamping load at the interface during relief or loading by F_{PA} and after embedding in service and F_S is the maximum bolt load.

4 Method B: ECSS-E-HB-32-23A

This chapter provides a quick overview and summary of the equations used in **BAT**. A detailed description can be found in the complete ECSS-E-HB-32-23A ESA handbook [2]. Some used variables in the following equations have been changed compared to [2] by the author to increase clarity and consistency.

4.1 Preload and Torques

The torque present at the thread interface M_{th} is dependent of the *axial bolt preload* F_V and is given by

$$M_{th} = F_V \tan(\varphi + \rho) \frac{d_2}{2} \tag{4.1}$$

and the *under-head torque* M_{uh} due to friction between bolt head or nut and the adjacent clamped part (or shim) is defined by

$$M_{uh} = F_V \frac{\mu_{uh} D_{Km}}{2} \frac{1}{\sin \lambda/2} \tag{4.2}$$

where λ is the *under head bearing angle* seen in Figure 2.1. It is assumed that the friction force for M_{uh} is acting at mean bearing radius of the bolt head D_{Km} (2.1). φ is the helix angle of the thread and ρ is given by the relation

$$\tan \rho = \frac{\mu_{th}}{\cos \theta/2} \tag{4.3}$$

where θ is the half angle of the thread groves (for Unified or Metric threads $\theta = 60°$).

The *total installation torque* T_A (without torque device scatter) applied to bolt head or nut during tightening to produce the axial bolt preload F_V is

$$T_A = M_{th} + M_{uh} + M_p \tag{4.4}$$

where M_p is the *prevailing torque* of the locking device. With the approximation $\tan\varphi \tan\rho \ll 1$ the expression $\tan(\varphi+\rho)$ can be written as $\tan(\varphi+\rho) \approx \tan\varphi + \tan\rho$. Now equation (4.4) can be rewritten to

$$T_A = F_V \underbrace{\left[\frac{d_2}{2}\left(\tan\varphi + \frac{\mu_{th}}{\cos{}^{\theta}\!/_{2}}\right) + \frac{\mu_{uh} D_{Km}}{2\sin{}^{\lambda}\!/_{2}}\right]}_{K} + M_p \tag{4.5}$$

where K is the *joint coefficient*.

For calculation of the minimum and maximum axial bolt preload, **BAT**implements the *experimental coefficient method* [2] with an explicit torque scatter torque of the tightening device $T_{scatter}$. Therefore the minimum and maximum total installation torques are defined

$$T_A^{min} = T_A - T_{scatter}, \qquad T_A^{max} = T_A + T_{scatter}. \tag{4.6}$$

To calculate the minimum and maximum *axial bolt preload after tightening* $F_M^{min/max}$, (4.5) and (4.6) are combined

$$F_M^{min} = \frac{T_A^{min} - M_p^{max}}{K^{max}}, \qquad F_M^{max} = \frac{T_A^{max} - M_p^{min}}{K^{min}}. \tag{4.7}$$

If also the thermal influence and embedding is considered, this leads to the minimum and maximum *axial bolt preload at service* $F_V^{min/max}$

$$F_V^{min} = \frac{T_A^{min} - M_p^{max}}{K^{max}} + \Delta F_{Vth} - F_Z \tag{4.8a}$$

$$F_V^{min} = F_M^{min} + \Delta F_{Vth} - F_Z \tag{4.8b}$$

$$= \frac{T_A^{min} - M_p^{max}}{\frac{d_2}{2}\left(\tan\varphi + \frac{\mu_{th}^{max}}{\cos{}^{\theta}\!/_{2}}\right) + \frac{\mu_{uh}^{max} D_{Km}}{2\sin{}^{\lambda}\!/_{2}}} + \Delta F_{Vth} - F_Z \tag{4.8c}$$

$$F_V^{max} = \frac{T_A^{max} - M_p^{min}}{K^{min}} + \Delta F_{Vth} \tag{4.9a}$$

$$F_V^{max} = F_M^{max} + \Delta F_{Vth} \tag{4.9b}$$

$$= \frac{T_A^{max} - M_p^{min}}{\frac{d_2}{2}\left(\tan\varphi + \frac{\mu_{th}^{min}}{\cos{}^{\theta}\!/_{2}}\right) + \frac{\mu_{uh}^{min} D_{Km}}{2\sin{}^{\lambda}\!/_{2}}} + \Delta F_{Vth} \tag{4.9c}$$

where ΔF_{Vth} is thermal preload change (see §4.2) and F_Z is the preload loss due to embedding (see §4.3).

4.2 Thermal Influcence

If a thermal load is applied to a bolted joint, the bolt sees a change ΔF_{Vth} in the preload force due to the CTE mismatch of bolt and clamped parts seen in (4.8c) and (4.9c).

4.2.1 Linear Thermal Influence

For the following derivation it is assumed that the Young's Modulus E of bolt and clamped parts does not change with temperature (temperature independent material properties). $c_b = {}^1\!/\!{\delta_b}$ and $c_c = {}^1\!/\!{\delta_c}$ are defined as bolt stiffness and clamp-part stiffness respectively. Linear thermal elongation is defined

$$\varepsilon_{th} = \alpha \Delta T = \frac{\Delta l}{l_0}$$

The thermal elongation for bolt (index b) and clamped-parts (index c) are given to

$$\begin{aligned}
\Delta l_b &= \alpha_b \cdot \Delta T \cdot l_{0b} \\
\Delta l_c &= \sum_i \alpha_{ci} \cdot \Delta T \cdot l_{0ci}
\end{aligned}$$

where $l_K = l_{0b} = \sum_i l_{0ci}$ is the clamping length of the joint. The sign definition in **BAT**is $\Delta L = \Delta l_c - \Delta l_b$ and this leads to

$$\begin{aligned}
\alpha_c > \alpha_b &\Rightarrow +\Delta F_{Vth} \\
\alpha_c < \alpha_b &\Rightarrow -\Delta F_{Vth}
\end{aligned}$$

where $+\Delta F_{Vth}$ defines an increase and $-\Delta F_{Vth}$ a loss in bolt preload. If the standard stiffness equation $F = c \cdot \Delta x$ is used for the bolt / clamp-part joint, this leads to

$$\Delta F_{Vth} = c \cdot \Delta L \tag{4.12a}$$

$$= \frac{\Delta L}{\frac{1}{c_b} + \frac{1}{c_c}} \tag{4.12b}$$

$$= \Delta L \frac{c_b c_c}{c_b + c_c} = \Delta L \frac{1}{\delta_b + \delta_c} \tag{4.12c}$$

4.2.2 VDI Method

to be filled

4.3 Embedding

to be filled

4.4 Compliance of Bolt and Clamped Parts

4.4.1 Compliance of Bolt

The bolt consists of different sections with dedicated compliances seen in Figure 4.1.

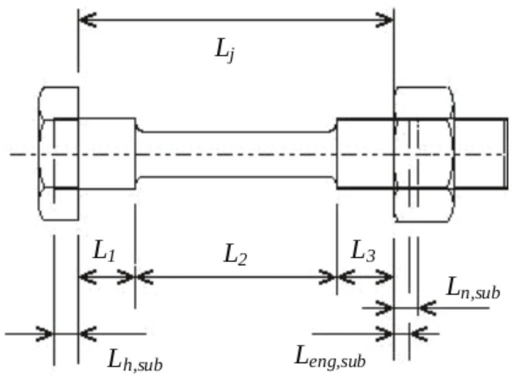

Figure 4.1: Dimensioning of the bolt for compliance calculation [2]

Applying the Hook's law to each segment of the bolt and combining the equations, the compliance of the bolt δ_b can be written

$$\delta_b = \frac{1}{c_b} = \frac{\Delta L_b}{F_V} = \frac{1}{E_b} \sum \frac{L_i}{A_i} \tag{4.13}$$

where $c_b = 1/\delta_b$ is the bolt stiffness, $\Delta L_b = \sum \Delta L_i$ is the overall length for bolt compliance calculation with segment length L_i. E_b is the bolt Young's Modulus and A_i are the cross sections for each bolt segement. Expanding (4.13) and introducing substitution lengths for deformation in the bolt head and engaged region in thread / nut the equation leads to

$$\delta_b = \frac{1}{E_b} \left[\frac{L_{h,sub}}{A_1} + \frac{L_{eng,sub}}{A_3} + \frac{0.4d}{A_1} + \frac{L_{eng,sub}}{A_3} \right] \tag{4.14}$$

$$\begin{aligned} \delta_b &= \frac{1}{E_b} \left(\frac{0.4d}{A_1} + \frac{l_K}{A_3} + \frac{0.4d}{A_1} + \frac{L_{eng,sub}}{A_3} \right) && \text{(4.15a)} \\ &= && \text{(4.15b)} \end{aligned}$$

BAT Info:
The current BAT implementation does not include *shank bolts* with different diameter sections; complete shaft length is threaded ($L_1 = L_2 = 0$). Also for the bolt head compliance only cylindrical head is considered $L_{h,sub} = 0.4d$ for simplicity.

4.5 Stesses in Bolt and Clamped-Parts

4.5.1 Stress in bolt after tightening

Based on §4.1 the stresses in the bolt after tightening can be derived. The minimum and maximum shear stress in the bolt $\tau^{min/max}$ after tightening can be calculated as follows

$$\tau^{min} = \frac{T_A^{min} - M_{uh}^{max}}{W_p}, \qquad \tau^{max} = \frac{T_A^{max} - M_{uh}^{min}}{W_p} \tag{4.16}$$

where T_A is defined in (4.6), M_{uh} in (4.2) with $F_V = F_M$ and $W_p = \pi d_s^3/16$ is the polar section modulus. The minimum and maximum normal stress $\sigma_n^{min/max}$ in the bolt caused by the preload is defined

$$\sigma_n^{min/max} = \frac{F_M^{min/max}}{A_s} \tag{4.17}$$

The minimum and maximum von-Mises equivalent stress $\sigma_v^{min/max}$ in the bolt after tightening (with 100% shear stress contribution / 0% shear stress relaxation) is defined

$$\sigma_v^{min/max} = \sqrt{\left(\sigma_n^{min/max}\right)^2 + 3\left(\tau^{min/max}\right)^2} \tag{4.18}$$

Based on the von-Mises equivalent stress in the bolt the minimum and maximum *bolt utilization* $\nu^{min/max}$ is defined

$$\nu^{min/max} = \frac{\sigma_v^{min/max}}{\sigma_y} \tag{4.19}$$

where σ_y is the bolt material yield strength and this defines the tightening status of the joint.

4.5.2 Stress and margins of safety in bolt at service

The bolted joint can be loaded with an axial force F_A and a shear force F_Q. For the axial loading the additional bolt load is defined in (3.1) and for the shar loading the required clamping force for friction grip per bolt is defined

$$F_{Kreq} = \frac{F_Q}{q_F \mu_T} \tag{4.20}$$

where q_F is the number of force transmitting and μ_T the coefficient of friction in the clamped part interfaces.

Local Slippage Margin MOS_{slip}^{local}

The *local slippage margin* is defined for a minimum preload F_V^{min}

$$MOS_{slip}^{local} = \frac{F_V^{min} - F_{PA}}{F_{Kreq} FOS_{slip}} - 1 \tag{4.21}$$

where FOS_{slip} is the factor of safety against slippage. It is also possible that the local slippage margin is defined with the mean preload $F_V^{mean} = 0.5(F_V^{min} + F_V^{max})$, based on engineering judgedment.

Local Gapping Margin MOS_{gap}

The *local gapping margin* is defined for a minimum preload F_V^{min}

$$MOS_{gap} = \frac{F_V^{min}}{F_{PA} FOS_{gap}} - 1 \tag{4.22}$$

where FOS_{gap} is the factor of safety against gapping.

Yield and Ultimate Bolt Margin $MOS_{y/u}$

The *yield and ultimate margins* are defined for a maximum preload F_V^{max}

$$MOS_y = \frac{\sigma_y}{\sqrt{\left(\frac{F_V^{max}+F_{SA}FOS_y}{A_s}\right)^2 + 3\left(0.5\tau^{max}\right)^2}} - 1 \tag{4.23a}$$

$$MOS_u = \frac{\sigma_u}{\sqrt{\left(\frac{F_V^{max}+F_{SA}FOS_u}{A_s}\right)^2 + 3\left(0.5\tau^{max}\right)^2}} - 1 \tag{4.23b}$$

where $FOS_{y/u}$ is the factor of safety against yield and ultimate. For the $MOS_{y/u}$ calculation a 50% shear load reduction $\tau = 0.5\tau^{max}$ is considered[2] .

[2] After tightening the shear stress in the bolt relaxes significantly after a few minutes and a conservative 50% relaxation is assumed for margin evaluation [2, 3].

5 References

[1] Guidelines for threaded fasteners. ESA Guideline ESA PSS-03-208 Issue 1, Structures and Mechanism Division ESTEC, December 1989.

[2] Space engineering - threaded fasteners handbook. ECSS Handbook ECSS-E-HB-32-23A, ECSS European Cooperation for Space Standardization, 16 April 2010.

[3] Systematic calculation of highly stressed bolted joints - joints with one cylindrical bolt. VDI Guideline VDI2230 Part 1, VDI - Verein Deutscher Ingenieure, November 2015.