# BAT - Bolt Analysis Tool

## User Manual


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# Symbols and Abbreviations

## Symbols

| | |
|---|---|
| $\alpha_A$ | tightening factor |
| $\alpha_b$ | coeff. of lin. thermal expansion of the bolt |
| $\alpha_c$ | coeff. of lin. thermal expansion of the clamped part (plate) |
| $\delta_b$ | elastic compliance of the bolt |
| $\delta_c$ | elastic compliance of the clamped part (plate) |
| $\lambda$ | under-head bearing angle of bolt |
| $\mu_{th}$ | coeff. of friction in bolt thread |
| $\mu_{uh}$ | coeff. of friction under bolt head |
| $\nu$ | bolt utilization factor |
| $\varphi$ | helix angle / slope of bolt thread |
| $\Phi$ | load factor of concentric joint<br>(also: force ratio or relative compliance factor) |
| $\Phi_n$ | load factor for concentric clamping and concentric<br>force load introduction via the clamped parts |
| $\rho$ | friction angle in bolt thread |
| $\sigma_n$ | normal stress in the bolt |
| $\sigma_v$ | von-Mises stress in the bolt |
| $\tau$ | 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].

$$p(\boldsymbol{\Theta}|\boldsymbol{y}) = \frac{p(\boldsymbol{y}|\boldsymbol{\Theta})\ p(\boldsymbol{\Theta})}{p(\boldsymbol{y})}\ , \tag{1.1}$$

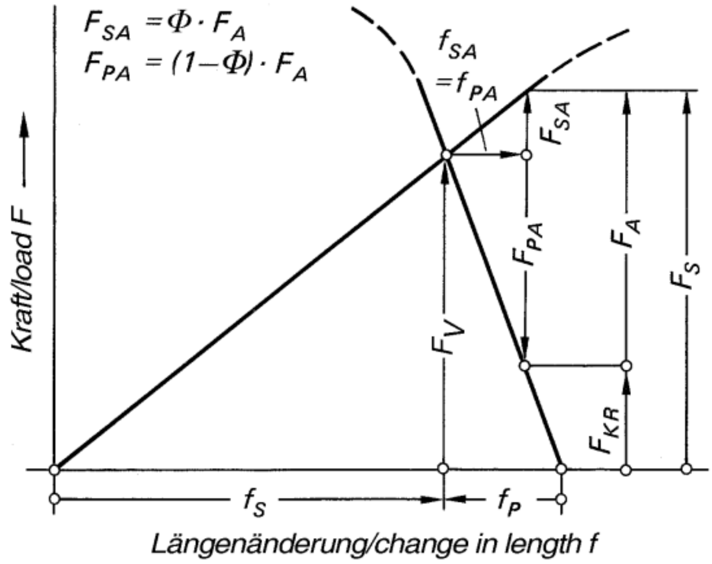


**Figure 1.1:** Joint diagram for the working state of a concentrically loaded bolted joint with $n = 1$ [3]

# 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.

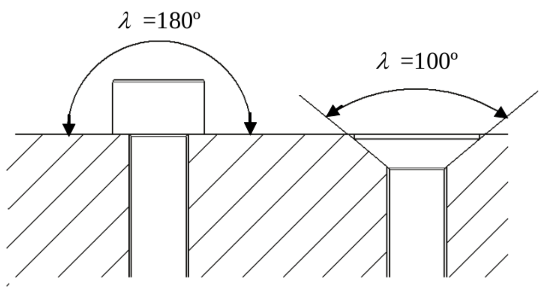


**Figure 2.1:** Definition of under head bearing angle [2]

# 3 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.

## 3.1 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

$$M_{th} = F_V \tan(\varphi + \rho) \frac{d_2}{2} \tag{3.1}$$

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

$$M_{uh} = F_V \frac{\mu_{uh} D_{Km}}{2} \frac{1}{\sin {}^{\lambda}\!/_{2}} \tag{3.2}$$

where $\lambda$ 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). $\varphi$ is the helix angle of the thread and $\rho$ is given by the relation

$$\tan \rho = \frac{\mu_{th}}{\cos {}^{\theta}\!/_{2}} \tag{3.3}$$

where $\theta$ 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

$$T_A = M_{th} + M_{uh} + M_p \tag{3.4}$$

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 (3.4) can be rewritten to

$$T_A = F_V \underbrace{\left[\frac{d_2}{2}\left(\tan\varphi + \frac{\mu_{th}}{\cos{}^{\theta}\!/_{2}}\right) + \frac{\mu_{uh} D_{Km}}{2\sin{}^{\lambda}\!/_{2}}\right]}_{K} + M_p \tag{3.5}$$

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

$$T_A^{min} = T_A - T_{scatter}, \qquad T_A^{max} = T_A + T_{scatter}. \tag{3.6}$$

To calculate the minimum and maximum *axial bolt preload after tightening* $F_M^{min/max}$, (3.5) and (3.13) are combined

$$F_M^{min} = \frac{T_A^{min} - M_p^{max}}{K^{max}}, \qquad F_M^{max} = \frac{T_A^{max} - M_p^{min}}{K^{min}}. \tag{3.7}$$

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

$$F_V^{min} = \frac{T_A^{min} - M_p^{max}}{K^{max}} + \Delta F_{Vth} - F_Z \tag{3.8a}$$

$$F_V^{min} = F_M^{min} + \Delta F_{Vth} - F_Z \tag{3.8b}$$

$$= \frac{T_A^{min} - M_p^{max}}{\frac{d_2}{2}\left(\tan\varphi + \frac{\mu_{th}^{max}}{\cos{}^{\theta}\!/_{2}}\right) + \frac{\mu_{uh}^{max} D_{Km}}{2\sin{}^{\lambda}\!/_{2}}} + \Delta F_{Vth} - F_Z \tag{3.8c}$$

$$F_V^{max} = \frac{T_A^{max} - M_p^{min}}{K^{min}} + \Delta F_{Vth} \tag{3.9a}$$

$$F_V^{max} = F_M^{max} + \Delta F_{Vth} \tag{3.9b}$$

$$= \frac{T_A^{max} - M_p^{min}}{\frac{d_2}{2}\left(\tan\varphi + \frac{\mu_{th}^{min}}{\cos{}^{\theta}\!/_{2}}\right) + \frac{\mu_{uh}^{min} D_{Km}}{2\sin{}^{\lambda}\!/_{2}}} + \Delta F_{Vth} \tag{3.9c}$$

where $\Delta F_{Vth}$ is thermal preload change (see (3.12c)) and $F_Z$ is the preload loss due to embedding (see equXX).

## 3.2 Thermal Influcence

If a thermal load is applied to a bolted joint, the bolt sees a change $\Delta F_{Vth}$ in the preload force due to the CTE mismatch of bolt and clamped parts seen in (3.8c) and (3.9c).

### 3.2.1 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

$$\Delta F_{Vth} = c \cdot \Delta L \tag{3.12a}$$

$$= \frac{\Delta L}{\frac{1}{c_b} + \frac{1}{c_c}} \tag{3.12b}$$

$$= \frac{\Delta L c_b c_c}{c_b + c_c} = \frac{\Delta L}{\delta_b + \delta_c} \tag{3.12c}$$

### 3.2.2 VDI Method

to be filled

## 3.3 Embedding

to be filled

## 3.4 Stesses in Bolt and Clamped-Parts

### 3.4.1 Stress in bolt after tightening

Based on §3.1 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

$$\tau^{min} = \frac{T_A^{min} - M_{uh}^{max}}{W_p}, \qquad \tau^{max} = \frac{T_A^{max} - M_{uh}^{min}}{W_p} \tag{3.13}$$

where $T_A$ is defined in (3.13), $M_{uh}$ in (3.2) 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

$$\sigma_n^{min/max} = \frac{F_M^{min/max}}{A_s} \tag{3.14}$$

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

$$\sigma_v^{min/max} = \sqrt{\left(\sigma_n^{min/max}\right)^2 + 3\left(\tau^{min/max}\right)^2} \tag{3.15}$$

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

$$\nu^{min/max} = \frac{\sigma_v^{min/max}}{\sigma_y} \tag{3.16}$$

where $\sigma_y$ is the bolt material yield strength and this defines the tightening status of the joint.

### 3.4.2 Stress in bolt at service

to be filled

# 4 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.