Bolt stressed by bending

The additional bend in the connecting screws must be avoided as the resulting bending stress can be several times greater than the tensile stresses of prestressing and operating forces. The causes of the additional bending moment can be divided into two groups. One group of causes is production inaccuracy. For example, a head or a nut rests on an uneven surface; the bearing surface of the head or nut is not perpendicular to the axis of the thread; the thread axis is not identical to the screw axis or the nut (diagonally threaded); bolt holes are inclined to the flange contact surface, etc. If the bolts or surfaces are plastic deformed, the bolt is not bent more flexibly. The second group consists of the causes caused by the deformation of the connected parts (e.g. the flanges in Fig. 1, where the deformation of the flanges and the bolt is heavily excessive) due to prestressing and operation forces.

The amount of additional bending stress, for example, for the flange bolt shown in Fig. 1 and the given deviation ψ and the load axial force F_A can be determined from the calculation model shown in Fig. 2. Here the screw is considered as a cylindrical rod of diameter d_B, length l and at one end firmly connected. Force ratios are shown in Fig. 2.

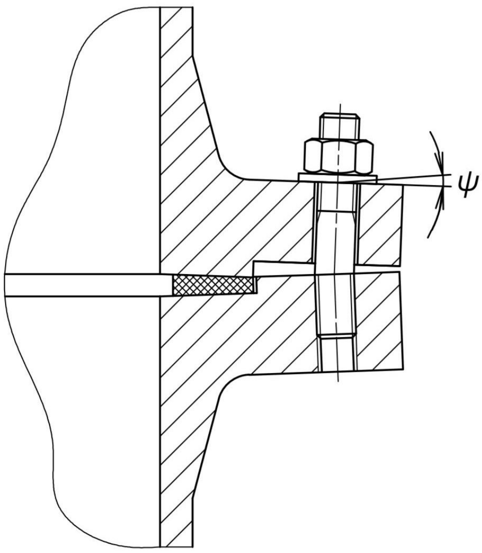

Fig.1 Emergence bending of the bolt

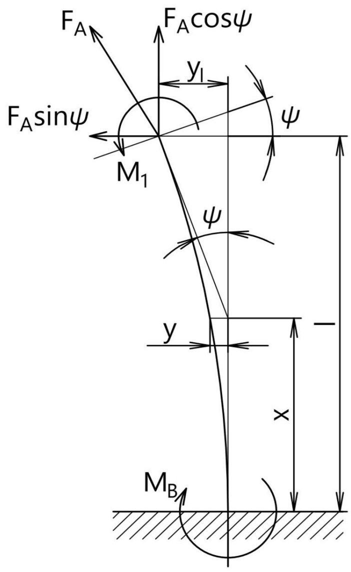

Fig.2 Computational Bolt Model

Assuming that the nut of the connecting bolt abuts the entire surface of Fig. 1, the bending moment is at x (Fig. 2)

$$M_B = M_1 + F_A * \sin\psi(1 - x) - F_A \cos\psi\,(y_l - y) \qquad (1)$$

Differential equations of the deflection line are

$$y'' = \frac{M}{EJ} \qquad y'' = \frac{1}{Q} \qquad (2)$$

where $J = \frac{\pi d_B^4}{64}$ is the axial moment of inertia of the diameter cross section d_B

E – the tensile modulus of the material

Q – radius of curvature

By combining equations (1) and (2), basic differential equations are formed. Assuming that the angle ψ is small, i.e. sin ψ ~ arc ψ, we obtain after integrating and considering the boundary conditions (Figure 2)

$$M_B = \frac{F_A * arc\psi}{\lambda * \tanh(\lambda * l)} \quad (3)$$

$$M_1 = \frac{F_A * arc\psi}{\lambda * \sinh(\lambda * l)} \quad (4)$$

$$\lambda = \sqrt{\frac{F_A * \cos\psi}{EJ}} = \sqrt{\frac{F_A}{EJ}} \quad (5)$$

$$M_B = \frac{d_B^2 * F_A * arc\psi * \sqrt{E * \pi}}{8\sqrt{F_A} * \tanh\left(\frac{8l}{d_B^2} * \sqrt{\frac{F_A}{E * \pi}}\right)} \quad (6)$$

Because the stress is not dependent on the threads or the influence of the plastic deformations in the rear faces, the stresses determined according to the given relations are somewhat higher than the real one.

For values $\lambda * l \leq 0{,}13 \div 0{,}14$ approximately $M_B = M_1$ applies. For larger values $\lambda * l$ is $M_B \geq M_1$, the longer screws are broken in the threaded section, as is known from experience.

The additional bending stresses must be respected for the strength calculation or screw inspection.

Using spherical bearing surfaces (head screw, nut, washers) can prevent additional bending from manufacturing inaccuracies, but not from deformation of the connected parts. The friction is so large that it does not allow the shift of the relative shift. It is therefore important that the connected parts are rigid enough to make their deformations negligible.

Example:

We need to determine the additional bending stress in the M16 bolt from manufacturing inaccuracies. d_B = 13,546mm; F_A = 50kN; E=2*10^5MPa; l=60mm; ψ=1*10^{-3}rad.

$$M_B = \frac{d_B^2 * F_A * arc\psi * \sqrt{E * \pi}}{8\sqrt{F_A} * \tanh\left(\frac{8l}{d_B^2} * \sqrt{\frac{F_A}{E * \pi}}\right)} =$$

$$= \frac{13{,}546^2 * 50000 * 3 * 10^{-4} * \sqrt{2 * 10^5 * \pi}}{8\sqrt{50000} * tanh\left(\frac{8 * 60}{13{,}546^2} * \sqrt{\frac{50000}{2 * 10^5 * \pi}}\right)} = 6474{,}7Nmm$$

$$\sigma = \frac{M_B}{\frac{\pi d_B^3}{32}} = \frac{6474{,}7}{\frac{\pi * 13{,}546^3}{32}} = 26{,}5 MPa$$

Literature:

František Pospíšil: Závitová a šroubová spojení. SNTL 1968
František Boháček: Části a mechanismy strojů I. 1984