STRESSES IN METAL ROCK BOLTS
AND THE CHOICE OF THEIR DIAMETER

(UDC 622.281.5)

B. I. Strygin
Skochinskii Mining Institute, Moscow
Translated from Fiziko-Tekhnicheskie Problemy Razrabotki Poleznykh Iskopaemykh, No. 4, pp. 30-37, July-August, 1965
Original article submitted December 19, 1964

In recent years bolts* have found wide use in the mining industry: many operations are now using expansion bolts. Experience has shown that expansion bolts operate more reliably than slot-and-wedge bolts; they can be withdrawn and used again, saving metal.

While the diameter of slot-and-wedge bolts is usually 25-30 mm, that of expansion bolts is usually below 20 mm, or 16 mm with alloy steels [1]. Thanks to their smaller diameters, expansion bolts use 2.5-3 times less metal than slot-and-wedge bolts. The shank diameter of slot-and-wedge bolts is chosen so that they will not yield while being driven in, will not lose stability, and can transmit dynamic loads from the projecting ends so as to tighten the yokes. In expansion bolts this is not required, as the expansion yokes are fixed into the boreholes by rotating the shanks (which screw into wedge-shaped or cone-shaped nuts in the yokes, which press the bearing elements against the sides of the boreholes), or by screwing nuts on to the lower ends of the projecting shanks. The wedge-shaped element, which is attached to the shank or integral with it, moves upwards and fixes the yoke into the borehole.

For efficient operation of the support, a tension of 3-4 t must be set up in each bolt, so that the shank becomes deformed by stretching. Since the tension is set up by applying a turning moment to the tensioning or wedging nuts of the yokes, friction occurs, and the shanks undergo deformation by twisting as well as by stretching. The shanks are thus in a complex state of stress and are subjected to simultaneous stretching and twisting. By means of mathematical analysis and a scientifically-based approach to the choice of shank diameter for each particular case, we can ensure reliability of the bolts in use with minimum expenditure of materials, thus further increasing their economic efficiency.

Investigations in mines have shown that, in addition to the initial stresses due to shotfiring, the shanks suffer additional dynamic loadings which decrease as the working face gets further from the point of installation of the supports. This decrease is more rapid at first, and as the face retreats the loads approach static values asymptotically.

As the dynamic loads in the shanks due to shotfiring are only effective in the first rows from the face, the optimum shank diameter must be calculated from the yield point, thus giving a better estimate of their bearing capacity. This allows us to plan more equally-strong and economic methods of construction, as the shanks will suffer lower loads as their distance from the face increases, and will thus be operating with greater reserves of strength.

As the length of a shank is many times greater than its diameter, we can assume that the stress is the same for all cross sections of the shank (see figure), while the deformation (axial extension) will be the same for all points in the shank. We shall therefore take $\varepsilon_x = \text{const}$.

Since the deformations are small, we can assume that there is no deformation of the cross-section's shape in its own plane, i.e., that it deforms like a rigid disk: this is equivalent to the condition that

---

*This type of support is often called a shank. The term "roof bolts" will be used in this article when we are referring to the support as a whole. The term "shank" will be reserved to mean that element of the support which acts as a link between the yoke and the tension-bearing part.
Let us write down all the equations of the theory of plasticity which are applicable to our problem. From the three differential equations of equilibrium, only one remains (the others reducing to identities) [2]:

\[ \frac{\partial \tau_{xx}}{\partial x} + \frac{\partial \tau_{yz}}{\partial y} + \frac{\partial \sigma_z}{\partial z} = 0. \] (1)

Of the constitutive relations there remain:

\[ \epsilon_z = \frac{1}{3G'} \sigma_z; \] (2)

\[ \tau_{xz} = G' \gamma_{xz}; \quad \tau_{yz} = G' \gamma_{yz}, \] (3)

where \( \epsilon \) is the relative elongation, \( G' \) is the generalized modulus of deformation, and \( \gamma \) is the angular displacement. The plasticity condition will be

\[ \sigma_z^2 + K_0 (\tau_{xz}^2 + \tau_{yz}^2) = \sigma_T^2, \] (4)

where \( K_0 = 3 \) when the Huber-Mises fluidity condition applies, \( K_0 = 4 \) when the Saint-Venant condition applies; \( \sigma_T \) is the yield stress of the material of the shank.

The relation for the strain, in the absence of curvature of the cross section in a plane, is written in the same way as for the case of pure twisting:

\[ \gamma_{xz} = \alpha y; \quad \gamma_{yz} = \alpha x, \] (5)

where \( \alpha \) is the relative angle of torsion of the cross section.

Let us now introduce our condition, given above,

\[ \epsilon_z = \text{const.} \] (6)

By (5) and (3) we have

\[ \tau_{xz} = G' \alpha y; \quad \tau_{yz} = G' \alpha x. \] (7)

By (2) and (7),

\[ \frac{\tau_{xz}}{\sigma_z} = \frac{\alpha}{3\epsilon_z} y; \quad \frac{\tau_{yz}}{\sigma_z} = \frac{\alpha}{3\epsilon_z} x, \] (8)

whence

\[ \tau_{xz}^2 + \tau_{yz}^2 = \left( \frac{\alpha}{\epsilon_z} \right)^2 \frac{\sigma_z^2}{9} \rho^2, \] (9)

where \( \rho \) is the distance from the center of the shank cross section to the point under consideration.

We introduce the symbol

\[ \frac{\alpha}{\epsilon_z} = \phi. \] (10)

*Only slightly different results are obtained from calculations with the Saint-Venant and Huber-Mises conditions. Of these two equations, it is therefore usual to choose that which leads to the simplest solution.