Work flange connections of structural elements of an open profile on high-strength bolts

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Abstract: Design solutions of flange assemblies and their elements’ features for open structure rods of building structures are discussed in the article. The distribution of contact stresses both, between flanges and under bolt washers is obtained as a result of a numerical experiment. It is established that the nature of contact stress diagrams and the location (eccentricities) of lever forces depend on the ratio of flanges’ and bolts’ stiffnesses. The dependence of the bending moment in a flange on the magnitude of lever forces is revealed. High-strength connection bolts can work for eccentric tension, depending on the ratio of flanges and bolts stiffnesses.

1. Introduction
This study addresses important issues, regarding the distribution of contact stresses and leverage in a flange connection (FC). This problem was highlighted in a number of works [1, 2, 3, and 4], however, bolt failure in the flange connection of structures is not uncommon in Fig. 1. The relevance of solving this problem is obvious, because it is believed that the displacement eccentricity of tensile lever forces in bolt under load is minimal and independent of the ratio of flange and bolt stiffnesses mostly in FC calculating.

Figure 1. Bolt destruction during its eccentric tension in the flange connection of the frame unit
The goal was to study the distribution of contact stresses in the flanges under the bolt head washer, to determine the position of the resultant force from the indicated stresses relative to the bolt’s axis as the flange thickness was changing. As an object of the study, constructive decisions were made of FC open profile rods with the location of the bolt in the outer zone [2].

2. Materials and Methodology
One can propose a design scheme (Fig. 2a, b) for FC rods of an open profile which corresponds to a lever mechanism of the 2nd kind. The points of forces application in such a lever (\(P_f\), \(N_b\)) are located on one side of the support, which location is determined by the application point of \((N_f)\) resultant of force contact between the flanges. The uncertainty of such a lever operation is due to the fact that, it is possible to redistribute contact stresses under the washer and bolt head, as well as between the flanges as the external load changes for a given ratio of the bolts and flanges stiffness. In this case, the resultant \(N_b\) and \(N_f\) can shift: the first (eccentricity \(e_1\)) can shift from the axis of the bolt to the axis of the flange load, the second (eccentricity \(e_2\)) can shift in the opposite direction, to the flange free edge. The solution of the problem reduces to determining the eccentricities \(e_1\) and \(e_2\) according to this scheme, which change their values while FC loading and deforming.

![Figure 2](image-url)

Figure 2. a - Finite element model (taking into account symmetry) FC; b - calculation scheme FC with a fragment of a deformed finite element model

A numerical experiment was carried out in order to achieve this goal. Stretched T-shaped FC were considered as finite element models, representing the calculated one row section of bolts in the outer zone (Fig. 2a). In order to reduce calculations volume, the symmetry principle was applied, which allowed us to create a model of the calculated area FC in the form of its 1/4 part. A section of an open profile rod fixed to the flange by fillet welds was adopted with a thickness of 20mm. The flange thickness \(t\) was set of 5 mm pitch and took values from 15 to 35 mm. The bolts’ diameters were taken \(d_b = 24\) mm. The sizes of bolts, washers and screws were assigned according to GOST 22356, GOST 22354, GOST 22355 for high-strength bolts from 40X "Select" steel for FC.

The model was built of volumetric finite elements in the form of prisms and pyramids with triangles and convex quadrangles with a minimum scale of 2.5 mm as its base. This made it possible to vary the size of the flanges thickness during the transition from one model to another by changing the number of finite elements layers in the flange. This method allowed taking into account the "scale factor" – the influence of changes in the flange thickness on its stress-strain state. The boundary conditions between washer and flange, as well as between flanges, were determined by the parameters of unilateral contact during compression, specified by contact non-linear elements. These elements simulated unidirectional communication between adjacent nodes of the flanges’ contacting surfaces.
and washers. In this case, the effect of coupling between nodes during compression and the absence of coupling under tension were realized.

The model also took into account the physical nonlinearity of materials’ properties, which behavior was specified in the form of bilinear dependencies under load. For this purpose, the elastoplastic work diagrams of the material corresponding to the steels were adopted in the elements of FC model: plate and weld bead were C345 steel, flange were C255 steel, washer and bolt were 40X “Select” steel.

The model was loaded stepwise. The first stage of loading corresponded to the preliminary tension of the bolt with a force of $N_b = 240$ kN.

Next steps were carried out sequentially with equal fractions of the external load $P_f$ attached to the end face of the element (Fig. 2) the way, the tenth loading stage $P_f = 150$ kN, which corresponded to the nominal stresses over the element cross section of 25 kN/sm$^2$. The limiting number of loading steps was taken equal to 20, which corresponded to the tensile strength of the loaded plate.

As a result of a numerical experiment, we obtained the distribution of contact stresses depending on the external load magnitude and the flanges thickness, also determined the locations of resultant $N_f$ and $N_b$ power contacts, both between flanges and flange and washer, respectively.

The coordinates of resultant $N_f$ and $N_b$ (eccentricities $e_1$ and $e_2$) were calculated by analogy with the method of determining the gravity center in mechanics. In order to do it, we used the values of longitudinal forces $N_i$ in each contact elements (the force between nodes of contacting surfaces) and their position $x_i$ relative to the bolt’ axis. Thus, the eccentricities were calculated by the formula:

$$e_1, e_2 = \sum x_i N_i / \sum N_i.$$

The change nature in the contact stresses under the washer and the displacement $e_1$ of Nb lever force depending on the flange thickness and load can be seen in figure 3. The displacement $e_2$ from the bolt’ axis of $N_f$ lever force and the change in the corresponding contact stresses between flanges are similarly shown in figure 4. Contact stresses undergo a substantial redistribution both under the flange bottom and the bolt washer in the process of FC stretches as the external load $P_f$ increases (Fig. 3 and 4).

The most significant change in contact stresses under the washer (Fig. 3) is observed for “thin” flanges $t = 15$ mm, and vice versa, the contact stresses between flanges are more intensively shifted to the outer edge of flange for “thick” flanges at $t = 35$ mm (Fig. 4).

The experiment’s results are presented in the form of the graphs in figures 5a and 5b. The dependences in figure 5a and 5b for $e_1$ and $e_2$ out of $P_f$ according to the nature of their change can be divided into two sections, which will correspond to the elastic and elastoplastic stages of the flange operation. The first section, the elastic region of flange operation, is in the $P_f$ range from 0 to 60 kN ($t = 15$ mm), 105 kN ($t = 20$ mm), 195 kN ($t = 25$ mm), 225 kN ($t = 30$ mm), 240 kN ($t = 35$ mm). The second section is the elastoplastic work of the flange. These sections can be approximated by linear relationships between the eccentricities $e_1$ and $e_2$ and the external load $P_f$. The given absolute values of $e_1$ and $e_2$ eccentricities should be correlated with the geometric dimensions $d_b$ of the bolt and the flange (Fig. 2) in order to evaluate their contribution to FC stress-strain state.
| $P_f$, kN | Contact stress distribution under the washer, $\sigma_z$, kN/cm². | Eccentricity $e_1$, mm |
|---------|-------------------|------------------|
|         | $t = 15$ mm       | $t = 25$ mm       | $t = 35$ mm       |
| 0       | ![Diagram](image1) | ![Diagram](image2) | ![Diagram](image3) |
|         | $e_1 = 0$          | $e_1 = 0$          | $e_1 = 0$          |
| 150     | ![Diagram](image4) | ![Diagram](image5) | ![Diagram](image6) |
|         | $e_1 = 2.24$       | $e_1 = 0.46$       | $e_1 = 0.20$       |
| 225     | ![Diagram](image7) | ![Diagram](image8) | ![Diagram](image9) |
|         | $e_1 = 4.46$       | $e_1 = 1.15$       | $e_1 = 0.48$       |

**Figure 3.** Change in contact stresses under the washer and displacement $e_1$ of the lever force $N_f$ from the axis of the bolt at load steps $P_f = 0$, 150, 225 kN for flanges with a thickness of $t = 15$, 25 and 35 mm.
Figure 4. $e_2$ shift from the bolt’ axis of $N_f$ lever force of the force contact between the flanges and the change in the corresponding contact stresses at load steps $P_f = 0$, 150, 225 kN for the flanges with a thickness of $t = 15$, 25 and 35 mm.
Figure 5. Change in the eccentricities $e_1, e_2$ and the bending moment $M_f$ in the flange with increasing load for flanges different thicknesses. a) is $e_1$ eccentricity of the lever force $N_f$ under the bolt washer; b) is $e_2$ eccentricity of $N_f$ lever force of the force contact between the flanges; c) is bending moment in $M_f$ flange

Let us analyze the relative values of $e_1/d_b$ and $e_2/a$, the data, which are given in table 1. It follows that the ratio $e_1/d_b$ decreases by a factor of 9 as the thickness increases from 15 to 35 mm for $P_f=150$ and 225 kN loads. For the ratio $e_2/a$, the opposite trend is observed, i.e. for loads $P_f=150$ and 225 kN, the ratio $e_2/a$ increases by 1.4 and 2.5 times, respectively. The lever force $N_f$ is practically located at the free edge of the flanges for FC with $t=35$ mm and higher.

Table 1. Relative values of $e_1/d_b$, $e_2/a$, %

| $P_f$, kN | $t=15$ mm | $t=25$ mm | $t=35$ mm |
|-----------|------------|------------|------------|
|           | $e_1/d_b$  | $e_2/a$    | $e_1/d_b$  | $e_2/a$    | $e_1/d_b$  | $e_2/a$    |
| 150       | 9          | 37         | 2          | 46         | 1          | 51         |
| 225       | 19         | 35         | 5          | 70         | 2          | 87         |

Corresponding limiting values of the calculated load $P_f$ and $P_{fb}$ are determined for FC models, based on the bending strength of the flange and the bolts strength. Their values are given in table 2. Eccentricities $e_1$ and $e_2$, which correspond to an external load equal in magnitude to these calculated $P_f$ and $P_{fb}$ values according to the graphs in figures 5a and 5b. These eccentricities are designated as $e_{1f}$, $e_{1b}$, $e_{2f}$, $e_{2b}$ (the index ‘f’ corresponds to a load equal to $P_f$ value; ‘b’ is equal to $P_{fb}$ value). Analyzing the values of $e_{1f}$, $e_{1b}$, $e_{2f}$, $e_{2b}$ (Table 2), it can be noted that they belong to the first sections of the graphs in figures 5a and 5b, corresponding to the elastic work of the flange material.
As the thickness increases from 15 to 35 mm for an external load range from 0 to the calculated values of flexural strength $P_f$ and bolt strength $P_{fb}$, the magnitude of the bending moment in the cross section of the flange along the edge of the weld varies depending on the external load and the flange thickness. This is observed at $P_f = 90, 150, 180, 240, 242$ kN, with flange thicknesses $t = 15, 20, 25, 30, 35$ mm, respectively.

### Table 2. Design forces for model FC from the conditions of the flange flexural strength $P_f$ and bolt strength $P_{fb}$ and their corresponding eccentricities

| $t$ (mm) | $P_f / P_{fb}$ (kN) | $e_1 / e_2$ (mm) | $e_2 / e_{2b}$ (mm) |
|----------|---------------------|------------------|---------------------|
| 15       | 55/107              | 0,09/0,65        | 6,62/14,7           |
| 20       | 97/136              | 0,24/0,6         | 13,16/19,18         |
| 25       | 146/157             | 0,43/0,51        | 22,21/24,08         |
| 30       | 199/178             | 0,55/0,44        | 35/30,42            |
| 35       | 248/196             | 0,56/0,38        | 46,08/36,92         |

In addition to the parameters presented above, the bending moment in the flange section along the weld edge is calculated. A beam design scheme was used to determine the bending moment (Fig. 2b), according to this, the moment will be equal to $M_f = P_f \cdot (57 - 24 + e_2) - N_b \cdot (e_1 + e_2)$.

In this case, the corresponding values of $N_b$, $e_1$, and $e_2$ were taken for each $P_f$ value. The dependence of $M_f$ bending moment on the external load $P_f$ is shown in Fig. 5c. Between the values of $M_f$ and $P_f$ there is a dependence close to linear up $P_f = 90$ kN, independent of the flanges thickness. The dependence between $M_f$ and $P_f$ at a given external load for flanges with a thickness of $t = 15$ mm, undergoes a change due to the adopted bilinear scheme of material operation in FC model. Other models show a similar deviation from linear dependence at higher levels of external load as flange thicknesses increase. This is observed at $P_f = 90, 150, 180, 240, 242$ kN, with flange thicknesses $t = 15, 20, 25, 30, 35$ mm, respectively.

### 3. Results

1. The design scheme for FC open profile rods corresponds to a lever mechanism of the 2nd kind which points of external load and resultant of the forces from the bolt to the flange are on one side of the resultant force contact location between the flanges. In this case, the resultant ones change their values and position, depending on the external load and the flange thickness. So the resultant force from the bolt to the flange is shifted from the bolt’ axis to the load of the flange’ axis, and the resultant force contact between the flanges in the opposite direction, to the flange’ free edge.

2. The numerical values of $e_1$ and $e_2$ eccentricities from the bolt’ axis are determined, which determine the location in the flange joint of the resultant forces of the power contacts under the bolt washer and between the flanges, respectively, depending on the external load and the flange thickness.

3. Contact stresses experience a significant redistribution, both under the sole of the flange and under the bolt washer as the external load increases. The most significant change in contact stresses under the washer is observed for “thin” flanges $t = 15$ mm with respect to “thick” flanges $t = 35$ mm, and vice versa, the contact stresses between flanges are more intensively shifted to the outer edge of the flange at “thick” flanges.

4. There is a linear relationship between the external load and the above eccentricities $e_1$ and $e_2$ for flanged joints of open profile rods with an external load range from 0 to the calculated values from the conditions of the flange bending strength and tensile strength. The magnitude of the bending moment in the cross section of the flange along the edge of the weld varies depending on the external load according to a law, which does not depend on the flanges thickness.

5. It is important to correlate $e_1$ and $e_2$ eccentricities with the geometric dimensions of the bolt and the flange to assess the stress-strain state of FC. As the thickness increases from 15 to 35 mm for an external load of 150 and 225 kN, $e_1$ ratio decreases by 9 times to the diameter of the bolt, $e_2$ ratio increases by 1.4 and 2, 5 times to the size from the bolt’s axis to the free edge of the flange,
respectively and under the same loading conditions. The lever force at an external load close to the calculated value is practically located at the free edge of the flanges for FC with $t = 35$ mm and higher.

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