Development of a flange connection model to study the effect of uneven bolt tightening forces on the stress-strain state

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Abstract. It is shown that during the operation of technological equipment and pipelines very often there is an uneven tightening (UT) of fasteners for flange connections (FC), which can lead to an emergency. On the model of flange connection developed in the computer-aided design system “KOMPAS-3D”, the influence of UT on the stress-strain state of FC elements from the point of view of ensuring their strength was studied. The adequacy of the model was proved by comparing the stresses calculated in the PASSAT software package in accordance with GOST 34233.4-2017 with the stresses obtained by modeling the flange and bolt with a uniform tightening force. A methodology has been developed to create an uneven tightening force (TF) on the FC model. The calculation results showed that when the bolt load per bolt increases even by a factor of 1.25, the circumferential and meridional membrane stresses in the FC increase, respectively, by 2.5 and 3.3 times, which can cause FC depressurization or impair its strength. It is necessary to continue research with other parameters of loading and structural design of the FC in order to determine the most dangerous combinations and recommend continuous monitoring of TF during operation.

A huge amount of technological equipment and pipelines are used at oil and gas refineries, to ensure the tightness of which in most cases flange connections (FC) of various designs are used. These connections are subject to various loads: internal or external pressure, bending, torques, etc. One of the requirements for such connections is to ensure uniform tightening of bolts (studs) [1]. Unequal tightening of threaded connections (TC) can lead to misalignment of parts, uneven compression of the gasket and subsequent skipping of the product, distortion of the parts to be joined, plastic deformation of fasteners, rupture of bolts or studs, and emergency situations. It is known that uneven tightening of threaded connections can occur both as a result of installation errors and during operation due to various factors, for example, vibration, cyclic changes in loads, temperature effects, etc. [2,7].

This work is devoted to the study of the influence of inhomogeneous tightening forces on the stress-strain state and to finding out how dangerous this situation is from the point of view of ensuring the strength of the main elements of the FC.

The study is based on the FC modeling method. To select a software product, an analysis of various CAD systems was carried out. As a result, the products of ASCON - CAD KOMPAS-3D were selected. For modeling and strength analysis of flange connections, the APM FEM: Strength Analysis module was used.

As the object of modeling, a flat welded flange (DN 250, PN 1 from steel 20) and a bolt (M20, L75 from steel 35), which was under the influence of internal overpressure P, were selected. Previously,
the impact of pressure changes on the PASSAT software package was studied bolt load \( P_b \) and stresses arising in various elements of the FC in the design sections. A complex dependence of \( P_b \) and stresses on \( P \) was established both during tightening and in working conditions. For an example, figures 1 and 2 show graphs of changes of these parameters.

![Figure 1. Dependence of bolt load on internal pressure.](image)

![Figure 2. The dependence of stresses in the flange on the internal pressure.](image)

Based on the analysis of the obtained dependences at the first stage, modeling was carried out at an internal pressure of \( P = 0.5 \) MPa.

When developing any model, first of all, it is necessary to check its adequacy. To this end, in this work, models of the flange and bolt were built, to which uniform bolt load and internal pressure were initially applied. The adequacy was checked by comparing the stresses obtained in the calculation in
the APM FEM: Strength Analysis module with similar values calculated in the PASSAT software package.

For this purpose, the values of loads and stresses at a pressure of $P = 0.5$ MPa in the sleeve and plate of flanged connections in the calculated sections, as well as in the bolt [3], which are presented in summary table 1, were summarized and analyzed.

| Parameter                                      | Designation in PC                  | Designation according to GOST R | Value |
|------------------------------------------------|-----------------------------------|---------------------------------|-------|
| Operating bolt load, H                         | $P_{\theta}^P$                    | $P_{\theta}^P$                  | 154600|
| The meridional bending stress in the sleeve, MPa| $\sigma_{\theta}^P$               | $\sigma_{\theta}^P$             | 92.5  |
| The meridional membrane stress in the sleeve,  | $\sigma_{\theta_{mm}}^p$          | $\sigma_{\theta_{mm}}^p$        | 2.8   |
| MPa                                             |                                   |                                 |       |
| The peripheral membrane stress in the sleeve,  | $\sigma_{\theta_{m67}}^P$         | $\sigma_{\theta_{m67}}^P$       | 5.1   |
| MPa                                             |                                   |                                 |       |
| Radial stress in the plate, MPa                 | $\sigma_{R}^P$                    | $\sigma_{R}^P$                  | 50.6  |
| Circumferential stress in a plate, MPa          | $\sigma_{T}^P$                    | $\sigma_{T}^P$                  | 14.2  |
| Equivalent stresses in a bolt, MPa              | $\sigma_{E2}$                     | $\sigma_{E2}$                   | 57.25 |

Based on the initial data and the calculations given in table 1, we measured the stresses on the models in different sections.

The methodology for constructing a model and performing calculations usually consists in choosing a design scheme, introducing boundary conditions, in particular, fixing constraints, applying loads, and splitting finite elements into a grid. Then a static calculation is performed.

When the bolts were evenly tightened, the flange was fixed in the form of a hard seal in the section connecting the flange sleeve, for example, with the body (to exclude the effect of the fastening on the stress-strain state of the FC) and in the section along the inner edge of the flange hole (to bring the design scheme to the regulatory scheme flange calculation). As a load on the flange, an internal pressure of $P = 0.5$ MPa and a bolt load in the form of a reduced specific force of 8.11 H/mm$^2$ to the face of the holes at the locations of the fasteners were applied.

For the bolt, fastening was carried out at the free end of the bolt rod, and the bolt load was applied to the back side of the bolt head in the form of reduced tensile specific force $F = 31.2$ H/mm$^2$. Then they were divided into finite element mesh and carried out a static calculation.

The fixing conditions and load application schemes for the flange connection model and bolt are shown in figure 3.
For the flange connection model, stress measurements were performed along the X, Y, Z axes, while along the Z axis – meridional stresses $S_Z$, the X axis – radial $S_X$, and the Y axis – circumferential $S_Y$. In this calculation model, it is possible to measure stresses at any point of the sleeve and plate. We were interested in membrane meridional stresses only in the sleeve of the flange connection far from the interface unit with the plate. We also studied annular, radial, and bending plates at the interface between the plate and the sleeve. For the bolt model, equivalent stresses were measured.

The results of measuring the meridional and circumferential stresses in the plate and the sleeve of the flange connection are shown in figure 2, the radial stresses in the plate - in figure 5, the equivalent stresses for the bolt - in figure 6.

![Figure 4](image1.png)

**Figure 4.** The values of the meridional and circumferential stresses in the plate and sleeve of the flange connection.

![Figure 5](image2.png)

**Figure 5.** Radial stress values in flange plate.

![Figure 6](image3.png)

**Figure 6.** Values of equivalent stresses in a bolt.

Comparison of the results obtained in the PC “PASSAT” and the model built in the PC “KOMPAS-3D” are presented in table 2.

**Table 2.** Comparison of voltage values calculated in the PASSAT and KOMPAS-3D software systems.

| FC element | Parameter | Designation in the PC “PASSAT” | Designation in the PC “KOMPAS-3D” | Discrepancy, % |
|------------|-----------|--------------------------------|-----------------------------------|----------------|
| Sleeve     | The meridional bending stress in the sleeve, $\sigma_\text{b}$, MPa | 92,5 | 77,5 | -16,2 |
|            | The meridional membrane stress in the sleeve, $\sigma_{\text{mem}}$, MPa | 2,8 | 2,7 | -3,6 |
|            | Circumferential membrane stress due to sleeve pressure, $\sigma_{\text{mem}}$, MPa | 5,1 | 5,1 | 0 |
Plate radial stress, $\sigma_{\text{r}}$, MPa

|          |        |        |
|----------|--------|--------|
| Plate    | 50.6   | 50.5   |
|          | 50.8   | +0.4   |
|          | 14.2   | 14.4   |
|          | 14.6   | +2.8   |
|          | 56.1   | -1.9   |
|          | 56.2   | -1.7   |
|          | 56.7   | -0.9   |
|          |        |        |
| Circular stress in the plate, $\sigma_{\text{c}}$, MPa |
|          | 14.2   | +1.4   |
|          | 14.6   | +2.8   |
|          | 56.1   | -1.9   |
|          | 56.2   | -1.7   |
|          | 56.7   | -0.9   |
|          |        |        |
| Equivalent stresses in a bolt, $\sigma_{\text{b}}$, MPa |
|          | 57.2   |        |
|          | 56.2   | -1.7   |
|          | 56.7   | -0.9   |

The discrepancy between the values showed the adequacy of the developed calculation models to the picture of the stress distribution in the flange connection.

Next, the effect of uneven tightening of fasteners on the stress-strain state of the flange connection and bolt was investigated. As practice has shown, the deviation of the torque of the bolts of the flange connection from the average value can reach 25% or more. At the first stage, the bolt load was changed only for one bolt, and the range of change was 125% and 200% of the calculated value of $P_B$, i.e. bolt loads, respectively, were equal to $P_{B1} = 1.25*P_B = 16104 \text{ H}$; $P_{B2} = 2*P = 25766 \text{ H}$, where $P_B$ is the bolt load under operating conditions.

When the bolts were not evenly tightened, the fastening of the flange and bolt were the same as when the bolts were evenly tightened.

An internal pressure equal to $P = 0.5 \text{ MPa}$ was also applied as a load on the flange. The difference was that a bolt load in the form of reduced force was applied to one of the holes, respectively 1.25 and 2.0 times greater than with uniform tightening, i.e. $F_1 = 10.14 \text{ H/mm}^2$ and $F_2 = 16.22 \text{ H/mm}^2$. A specific force equal to $F = 8.11 \text{ H/mm}^2$ (as with uniform tightening) was applied to the other holes.

Then, a partition into a finite element mesh was also carried out and a static calculation was carried out.

The results of these studies are presented in figures 7-10 and in tables 3-4.

**Figure 7.** The values of the meridional and circumferential stresses in the plate and sleeve of the flange connection at $R_{B1}$.

**Figure 8.** Values of radial stresses in a plate of a flange connection at $R_{B1}$ and $R_{B2}$. 
Figure 9. Values of equivalent stresses in a bolt at \( R_{B1} \) and \( R_{B2} \).

Figure 10. Values of meridional and circumferential stresses in a plate and a sleeve of a flange connection at \( R_{B2} \).

Table 3. The results of studies of the stress-strain state of the flange connection with \( R_{B1} \).

| FC element | Parameter | Bolt stress values | Discrepancy, % |
|------------|-----------|--------------------|----------------|
|           |           | \( P_{B1} = 12883 \text{ H} \) | \( P_{B1} = 16104 \text{ H} \) |
| Sleeve    | The meridional bending stress in the sleeve, \( \sigma_{\rho}^{p} \), MPa | 77,5 | 84,7 | 109,3 |
|           | The meridional membrane stress in the sleeve, \( \sigma_{\rho}^{oms} \), MPa | 2,7 | 4,2 | 155,6 |
|           | Circumferential membrane stress due to sleeve pressure, \( \sigma_{\rho}^{oms} \), MPa | 2,7 | 3,8 | 140,7 |
|           | Plate radial stress, \( \sigma_{p}^{R} \), MPa | 5,1 | 12,9 | 252,9 |
| Plate     | Circular stress in the plate, \( \sigma_{p}^{T} \), MPa | 50,5 | 56,0 | 110,9 |
|           | 50,8 | 56,9 | 112,0 |
|           | 14,4 | 16,0 | 111,1 |
|           | 14,6 | 16,6 | 113,7 |
|           | 56,1 | 70,2 | 125,1 |
|           | 56,2 | 70,3 | 125,1 |
|           | 56,7 | 70,9 | 125,0 |
| Bolt      | Equivalent stresses in a bolt, \( \sigma_{B2} \), MPa | 14,4 | 16,0 | 111,1 |
|           | 14,6 | 16,6 | 113,7 |
|           | 56,1 | 70,2 | 125,1 |
|           | 56,2 | 70,3 | 125,1 |
|           | 56,7 | 70,9 | 125,0 |

Table 4. The results of studies of the stress-strain state of the flange connection with \( R_{B2} \).

| FC element | Parameter | Bolt stress values | Discrepancy, % |
|------------|-----------|--------------------|----------------|
|           |           | \( P_{B2} = 12883 \text{ H} \) | \( P_{B2} = 25766 \text{ H} \) |
| Sleeve    | The meridional bending stress in the sleeve, \( \sigma_{\rho}^{p} \), MPa | 77,5 | 121,3 | 156,5 |
|           | The meridional membrane stress in the sleeve, \( \sigma_{\rho}^{oms} \), MPa | 2,7 | 9,1 | 337,0 |
|                      |     |     |     |
|----------------------|-----|-----|-----|
| \( \sigma_{\text{r}, \text{m}} \), MPa | 2,7 | 5,7 | 211,1 |
| Circumferential membrane stress due to sleeve pressure, \( \sigma_{\text{r}, \text{m}} \), MPa | 5,1 | 16,6 | 325,5 |
| Plate radial stress, \( \sigma_{\text{r}, \text{c}} \), MPa | 50,5 | 73,4 | 145,3 |
| Plate                          |     |     |     |
| Circular stress in the plate, \( \sigma_{\text{r}, \text{T}} \), MPa | 14,4 | 17,9 | 124,3 |
| Equivalent stresses in a bolt, \( \sigma_{\text{B}, \text{2}} \), MPa | 56,1 | 112,3 | 200,2 |
| Bolt                          |     |     |     |
| Equivalent stresses in a bolt, \( \sigma_{\text{B}, \text{2}} \), MPa | 56,2 | 112,5 | 200,2 |
| Bolt                           |     |     |     |
| Equivalent stresses in a bolt, \( \sigma_{\text{B}, \text{2}} \), MPa | 56,7 | 113,5 | 200,2 |

As can be seen, the maximum divergence of values is observed for circumferential membrane stresses, which increase 2.5 times with an increase in the bolt load of one bolt by 25%, and 3.3 times with an increase in the bolt load up to 200% of the initial meridional membrane stress also increases by almost 3.3 times with increasing bolt load to \( R_{\text{B}2} = 2*R_{\text{B}} \).

Thus, the results of these studies show that with a constriction of bolts approximately 25% more, compared with the calculated tightening force, the stresses increase more than twice. Moreover, these data were obtained at a pressure of 0.5 MPa. Apparently, with increasing pressure, the stresses can exceed the permissible ones and reach the yield strength, which can cause FC depressurization or disrupt its strength. This is especially dangerous for the case when redistribution of the tightening force during operation can occur. To avoid this, it is necessary to identify those FCs in which the occurrence of an uneven tightening force and for them to control TF using devices that can be used both at the installation stage and during operation. For example, this is the IN-01m device, as well as devices that allow you to record the tightening force continuously throughout the entire operation period [1,4,5,6]. The results obtained indicate that it is necessary to continue studying the effect of uneven bolt tightening on the stress-strain state of a flat welded flange at high pressures. It is also of interest how the non-uniform force of bolt tightening affects the stress-strain state of flange connections of other sizes and of another type (welded butt).

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