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EFFECTIVE DESIGN OF A BOLTED FLANGED JOINT WITH A FLAT RING GASKET

The paper is devoted to a bolted flanged joint with a flat ring gasket. Simple mathematical models of the flat ring gasket and the flange are formulated. Solutions to the models allowed determining numerically effective shapes of the flat ring and the flange. In the case of the gasket a minimal tension of the bolts was assumed as a criterion, while in the case of the flange the criterion of minimal angle of the flange rotation was applied. Results of the study, shown in the Figures, may serve for practical purposes in designing of pressure vessels and piping.

1. Introduction

Bolted flanged joints are used from years in pressure vessels and piping. A basic problem related to the topic consists in leakage tightness of these joints. Thomson [12] indicated the problem and discussed designing fundamentals of these joints in accordance with British Standards and American National Standards. Bouzid at al. [2], [3], [4] i [5] paid attention to the effect of gasket creep-relaxation on the leakage tightness and rotations of the flanges of these joints. Brown [6] presented a practical method enabling assessment of the effect of temperature on bolted joints. Jenco and Hunt [7] discussed improved plant leakage reduction techniques. Abid and Nash [1] numerically investigated deformation of flanges of the geometry optimised with a view to ensure safe stress and non-leaking conditions. Magnucki and Sekulski [10], Varga and Baratosy [13] determined optimal preliminary

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tension of the bolts of flanged joints. Their mathematical description takes into account rotation of the flanges.

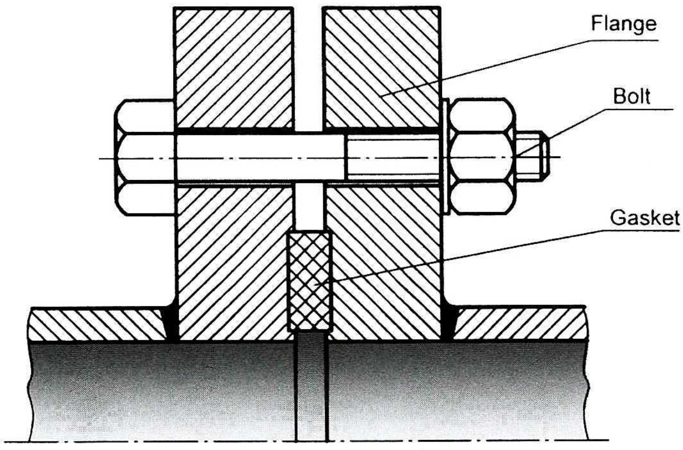

Fig. 1. Scheme of a typical bolted flanget joint

The subject of the study is a bolted flanged joint with a flat ring gasket presented in Fig. 1. Simple mathematical models of deformation of the flat ring gasket and the flange are formulated. Solutions to the models allowed us to propose effective shapes of the parts of the joint under strength constraints.

2. Effective overall sizes of the flat ring gasket

The study includes a flat ring gasket of an incompressible material. Axially symmetrical compression problem of the flat ring made of an incompressible material was presented by Wegner, Magnucki and Wasilewicz [8], [15], [16]. These works analytically describe the strain and the stress states and assume that plane cross sections before compression remain planar after compression. Wegner [14] generalized solution of the problem without the assumption of planar cross sections and developed an original discrete method using stress relaxation in nodes.

The equilibrium equations of the incompressible flat ring gasket shown in Fig. 2, considered as an axially symmetrical problem (Timoshenko and Goodier [11]), have the following form:

$$\frac{\partial \sigma_r}{\partial r} + \frac{\partial \tau_{rz}}{\partial z} + \frac{\sigma_r - \sigma_\varphi}{r} = 0, \qquad \frac{1}{r}\frac{\partial}{\partial r}(r\tau_{rz}) + \frac{\partial \sigma_z}{\partial z} = 0, \tag{1}$$

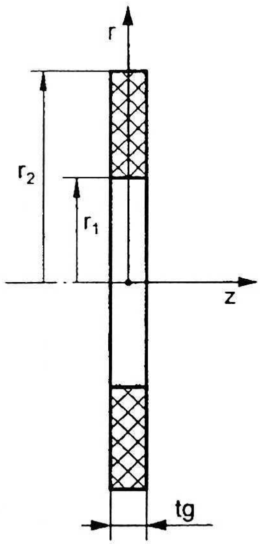

Fig. 2. Scheme of a flat ring gasket

where

σ_r, σ_z, τ_{rz} – components of the stresses state.

These equations of equilibrium (1) are included in the biharmonic equation

$$\nabla^4\Phi = 0 \tag{2}$$

where

$$\sigma_r=\frac{\partial}{\partial z}\left(\frac{1}{2}\nabla^2\Phi-\frac{\partial^2\Phi}{\partial r^2}\right),\quad \sigma_\varphi=\frac{\partial}{\partial z}\left(\frac{1}{2}\nabla^2\Phi-\frac{1}{r}\frac{\partial\Phi}{\partial r}\right),\quad \sigma_z=\frac{\partial}{\partial z}\left(\frac{3}{2}\nabla^2\Phi-\frac{\partial^2\Phi}{\partial z^2}\right),$$

$$\tau_{rz}=\frac{\partial}{\partial r}\left(\frac{1}{2}\nabla^2\Phi-\frac{\partial^2\Phi}{\partial z^2}\right),\quad \nabla^2\Phi=\frac{\partial^2\Phi}{\partial r^2}+\frac{1}{r}\frac{\partial\Phi}{\partial r}+\frac{\partial^2\Phi}{\partial z^2}$$ – the stress function.

The components of the strain state, with consideration of Hooke's law, are written in the form

$$\varepsilon_r=\frac{\partial u}{\partial r}=\frac{1}{E_g}\left[\sigma_r-\frac{1}{2}(\sigma_r+\sigma_z)\right]=-\frac{3}{2E_g}\frac{\partial^3\Phi}{\partial r^2\partial z},$$

$$\varepsilon_\varphi=\frac{u}{r}=\frac{1}{E_g}\left[\sigma_\varphi-\frac{1}{2}(\sigma_r+\sigma_z)\right]=-\frac{3}{2E_g}\frac{\partial^2\Phi}{r\partial r\partial z}, \tag{3}$$

$$\varepsilon_z = \frac{\partial w}{\partial z} = \frac{1}{E_g}\left[\sigma_z - \frac{1}{2}(\sigma_r + \sigma_\varphi)\right] = \frac{3}{2E_g}\frac{\partial}{\partial z}\left(\frac{\partial^2 \Phi}{\partial r^2} + \frac{1}{r}\frac{\partial \Phi}{\partial r}\right).$$

Therefore, the radial displacement $u(r, z)$ and the deflection $w(r, z)$ of the gasket, are as follows:

$$u(r, z) = -\frac{3}{2E_g}\frac{\partial^2 \Phi}{\partial r \partial z}, \qquad w(r, z) = \frac{3}{2E_g}\frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial \Phi}{\partial r}\right). \tag{4}$$

Mathematical model of the flat ring gasket is formulated based on paper [8]. It was assumed that deflection $w(r, z)$ of the compressed gasket between rigid flanges is a function of only one variable z, taking into consideration a planar cross section hypothesis stating, that $w(r,z) = w(z)$.

The incompressible linear condition ($\varepsilon^r + \varepsilon_\varphi + \varepsilon_z = 0$), with expressions (3) is in the form

$$\frac{1}{r}\frac{\partial}{\partial r}[r \cdot u(r, z)] + \frac{dw}{dz} = 0, \quad \text{form which} \quad u(r, z) = \frac{1}{r}f_1(z) - \frac{r}{2}\frac{dw}{dz}, \tag{5}$$

where $f_1(z)$ is a function, that is a result of the integration.

Substituting expression $(4)_1$ for $u(r, z)$ provides

$$\frac{\partial^2 \Phi}{\partial r \partial z} = -\frac{2}{3}E_g\left[\frac{1}{r}f_1(z) - \frac{r}{2}\frac{dw}{dz}\right],$$

that the stress function is of the following form:

$$\Phi(r, z) = -\frac{2}{3}E_g\left[f_3(r) + \int f_2(z)\,dz + \ln\left(\frac{r}{r_1}\right)\cdot\int f_1(z)\,dz - \frac{r^2}{4}w(z)\right], \tag{6}$$

where: $f_2(z)$ and $f_3(z)$ are functions, as integration results.

Substitution of this function into equation $(4)_2$ gives

$$\frac{d^2 f_3}{dr^2} + \frac{1}{r}\frac{df_3}{dr} = 0, \quad \text{from which} \quad f_3(r) = B_3 \ln\frac{r}{r_1}, \tag{7}$$

where B_3 is the integration constant.

Then, substitution of function (6) into the biharmonic equation (2) gives

$$\frac{d}{dz}\left(\frac{d^2 f_2}{dz^2} - 2\frac{dw}{dz}\right) + \frac{d^3 f_1}{dz^3}\ln\frac{r}{r_1} - \frac{r^2}{4}\frac{d^4 w}{dz^4} = 0. \tag{8}$$

Hence, this equation is true when:

$$\frac{d}{dz}\left(\frac{d^2 f_2}{dz^2} - 2\frac{dw}{dz}\right) = 0, \quad \frac{d^3 f_1}{dz^3} = 0, \quad \frac{d^4 w}{dz^4} = 0,$$

thus

$$w(z) = w_g\left(\alpha_0 + \alpha_1\zeta + \frac{1}{2}\alpha_2\zeta^2 + \frac{1}{6}\alpha_3\zeta^3\right), \quad f_1(z) = B_{10} + B_{11}\zeta + \frac{1}{2}B_{12}\zeta^2,$$

$$f_2(z) = t_g^2\left[B_{20} + B_{22}\zeta^2 + \frac{w_a}{t_g}\left(\alpha_1\zeta^2 + \frac{1}{12}\alpha_3\zeta^4\right)\right], \tag{9}$$

where $\zeta = \frac{z}{t_g}$ is the dimensionless coordinate $\left(-\frac{1}{2} \le \zeta \le \frac{1}{2}\right)$, t_g – thickness of the gasket.

The flat ring gasket is compressed by two rigid flanges, therefore, constraints of the displacements are as follows:

1^0 for $\zeta = 0$, $w(0) = 0$;

2^0 for $\zeta = 1/2$, $w(-1/2) = w_g/2$; and for $\zeta = 1/2$, $w(1/2) = -w_g/2$,

3^0 for $r = r_0$, $u(r_0, z) = 0$.

Hence, unknown displacement functions take the forms

$$w(z) = -\frac{1}{2}w_g(3 - 4\zeta^2)\zeta. \tag{10}$$

$$u(r,z) = \frac{1}{2}\left\{\frac{r_0^2}{r}\left[\frac{\alpha_3}{2}\zeta^2 - \left(1 + \frac{\alpha_3}{24}\right)\right] + r\left[1 + \frac{\alpha_3}{2}\left(\frac{1}{12} - \zeta^2\right)\right]\right\}\frac{w_a}{t_g}, \tag{11}$$

where

$$r_0 = r_1\sqrt{3\frac{r_{21}^2 - 1}{6\ln r_{21} + (r_{21}^2 - 1)(t_g/r_2)^2}}, \qquad r_{21} = \frac{r_2}{r_1}, \qquad \alpha_3 = 24\frac{1 - c_c}{2 + c_c},$$

$$0 \leq c_c \leq 1,$$

and $c_c = \dfrac{u(r, \pm 1/2)}{u(r, 0)}$ is the coefficient of flange reaction.

It is assumed that the flat ring gasket is supported in the seats of flanges and these facings do not slide each on other. The boundary condition in this case is in the form

$$4^0 \text{ for } \zeta = \pm 1/2, \quad u(r, \pm 1/2) = 0, \quad \text{i.e.} \quad c_c = 0.$$

Hence, the radial displacement (11) is in following form

$$u(r, z) = \frac{3}{4}(1 - 4\zeta^2)\left(r - \frac{r_0^2}{r}\right)\frac{w_g}{t_g}. \tag{12}$$

Taking the above into consideration, we express the axial compressive stress by the following function

$$\sigma_z = -E_g\left\{\frac{1}{3}\left[\left(\frac{r_0}{r_1}\right) + 4\right] - \left(\frac{r_1}{t_g}\right)^2\left[\left(\frac{r}{r_1}\right)^2 - 1\right] + 2\left(\frac{r_0}{t_g}\right)^2 \ln\frac{r}{r_1} - 4\zeta^2\right\}\frac{w_g}{t_g} \tag{13}$$

The axial force of the gasket for $\zeta = \pm 1/2$ is equal to

$$F_g = -2\pi \int_{r_1}^{r_2} \sigma_z r dr.$$

Substituting expression (13) for σ_z, one obtains

$$F_g = E_g A_g C_g \frac{w_g}{t_g} \tag{14}$$

where $A_g = \pi r_1^2 (r_{21}^2 - 1)$,

$$C_g = \frac{1}{3}\left[\left(\frac{r_0}{r_1}\right)2 + 4\right] - \left(\frac{r_1}{t_g}\right)^2\left[\frac{1}{2}(r_{21}^2 - 1) + \left(\frac{r_0}{r_1}\right)^2\left(1 - 2\frac{r_{21}^2}{r_{21}^2 - 1}\ln r_{21}\right)\right] - 1,$$

$$r_{21} = \frac{r_2}{r_1}.$$

Mathematical model of the bolted flanged joint (Magnucki and Sekulski [10]) describes

- an initial state ($p_0 = 0$), in terms of two equations

$$n_b F_b = F_g \text{ – equation of equilibrium,} \tag{15}$$

$$\delta_N = \Delta L_b + w_g \text{ – equation of displacements,} \tag{16}$$

where n_b is number of bolts, F_b is the force of the bolt, and δ_N is axial shift of the nut,

$\Delta L_b = \dfrac{F_b L_b}{E_b A_b}$ – elongation of the bolt, $w_g = \dfrac{F_g t_g}{E_g A_g C_g}$ – deflection of the gasket.

- an operational state ($p_0 \neq 0$), also with the help of two equations

$$n_b \Delta F_b + \Delta F_g = \pi r_1^2 p_0 \text{ – equation of equilibrium,} \tag{17}$$

$$\Delta(\Delta L_b) = \Delta w_a \text{ – equation of displacements,} \tag{18}$$

where: ΔF_b is the force increment in the bolt, ΔF_g is the force decrement in the gasket,

$\Delta(\Delta L_b) = \dfrac{\Delta F_b L_b}{E_b A_b}$ – elongation increment of the bolt, p_0 – design pressure,

$\Delta w_g = \dfrac{\Delta F_g t_g}{E_g A_g C_g}$ – deflection decrement of the gasket.

The tightness condition for the bolted flanged joint is in the following form

$$k_g p_0 \leq \frac{F_g - \Delta F_g}{A_g}, \tag{19}$$

where k_g is safety coefficient.

Solution of the equations (15)–(18) together with the condition (19) enabled us to determine:

- minimal pressure between flanges and the gasket

$$p_{g,\min} = \frac{F_{g,\min}}{A_g} = \left[k_g + \frac{1}{(1 + n_b k_0)(r_{21}^2 - 1)} \right] p_0, \tag{20}$$

where $k_0 = \dfrac{E_b A_b t_g}{E_g A_g C_g L_b}$ is dimensionless coefficient.

- minimal force of the bolt

$$F_{b,\min} = \frac{\pi r_1^2}{n_b}\left[(r_{21}^2 - 1)k_g + \frac{1}{1 + n_b k_0}\right]p_0, \tag{21}$$

- axial shift of the nut

$$\delta_N = \frac{\pi r_1^2 L_b}{n_b E_b A_b}\left[1 + (r_{21}^2 - 1)(1 + n_b k_0)k_g\right]p_0. \tag{22}$$

The rate of the external to internal radius $r_{21} = r_2/r_1$ considerably affects minimal pressure between flanges and the gasket (20), minimal force of the bolt (21) and axial shift of the nut (22). It was assumed that effective shape of the flat ring gasket corresponds to the smallest required axial shift of the nut (22). Hence, the optimal design criterion is of the following form

$$\min_{r_{21}}\left\{\tilde{\delta}_N\right\}, \tag{23}$$

where $\tilde{\delta}_N = 1 + (r_{21}^2 - 1)(1 + n_b k_0)k_g$ is dimensionless axial shift of the nut.

The gasket was shaped with the requirement of its constant volume (V_g = const.). Therefore, the thickness is equal to

$$t_g = \frac{V_g}{\pi r_1^2 (r_{21}^2 - 1)}. \tag{24}$$

Moreover, the geometrical constraint for rectangular cross section of the gasket was imposed in the form

$$e_g \leq \frac{r_1}{t_g}(r_{21} - 1), \tag{25}$$

where e_g is the parameter of rectangularity ($1 \leq e_g$).

Substitution of the thickness (24) gives

$$0 \leq r_{21}^3 - r_{21}^2 - r_{21} + 1 - \frac{V_g}{\pi r_1^3} e_g, \tag{26}$$

from which the minimal ratio $r_{21,\min}$ is obtained.

Numerical investigation was carried out for the following example data:

- the steel bolts with $E_b = 2.05 \cdot 10^5$ MPa, $A_b = 76.2$ mm^2, $n_b = 16$, $L_b = 40$ mm,
- the elastomer gasket with $E_g = 11$ MPa, $V_g = 1 \cdot 10^5$ mm^3, $k_g = 2$, $e_g = 2.5$, $r_1 = 105$ mm.

The equation (26) yields a value of minimal ratio $r_{21,\min} = 1.178$. Hence, $r_{21,\min} \leq r_{21}$. The effect of the ratio r_{21} on the dimensionless axial shift of the nut $\tilde{\delta}_N$ is shown in Fig. 3. Minimal value of $\tilde{\delta}_N$ for the numerical data adopted in this case occurs for $r_{21,opt} = 1.485$. Effective sizes of the gasket in this case are as follows: $r_1 = 105$ mm, $r_{2,opt} = 156$ mm, $t_g = 2.4$ mm. The gaskets corresponding to other numerical data were investigated on a similar way. It was found that a minimum of the dimensionless axial shift of the nut $\tilde{\delta}_N$ (23) always exists, but not always is located in the allowable space ($r_{21,\min} \leq r_{21}$). In such a case minimal value of $\tilde{\delta}_N$ occurs at the area boundary for $r_{21} = r_{21,\min}$.

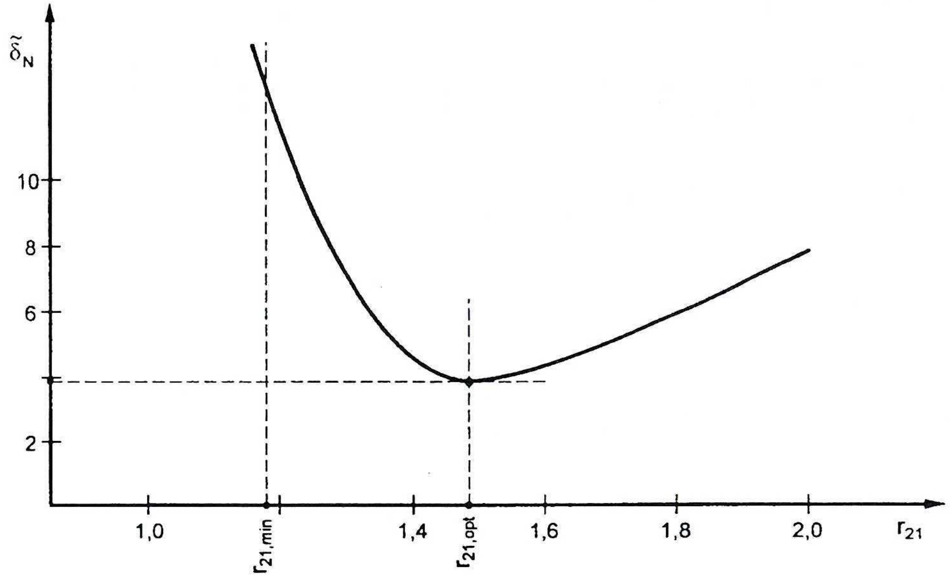

Fig. 3. Dimensionless axial shift of the nut

3. Effective shape of the flange

The cross section of a classical flange is a rectangular (Fig. 1). Bouzid and Chaaban (1993), Thomson (1994), Bouzid and Derenne (1999), Magnucki and Sekulski (1999) indicated, among others, disadvantageous effect of flange rotation on tightness of flanged bolted joints and the leakage problem.

The angle of rotation might be certainly reduced by increasing flange rigidity, increasing, for example, its thickness and, at the same time, its mass. Taking this into account, the angle of rotation could be presumably reduced in result of changes in the shape of the flange cross section maintaining its constant mass, that is subject to investigation. A classical rectangular cross section of the flange was generalized by "gluing" another rectangle (an additional part) (Fig. 4). The volume of this flange is as follows

$$V_f = \pi\left[\left(R_3^2 - R_1^2\right)a_1 + \left(R_2^2 - R_3^2\right)a_2\right], \tag{27}$$

where: R_1 and R_2 – internal and external radii of the flange, R_3 – internal radius of the additional part of the flange, a_1 – thickness of the main part of the flange, a_2 – thickness of the additional part of the flange.

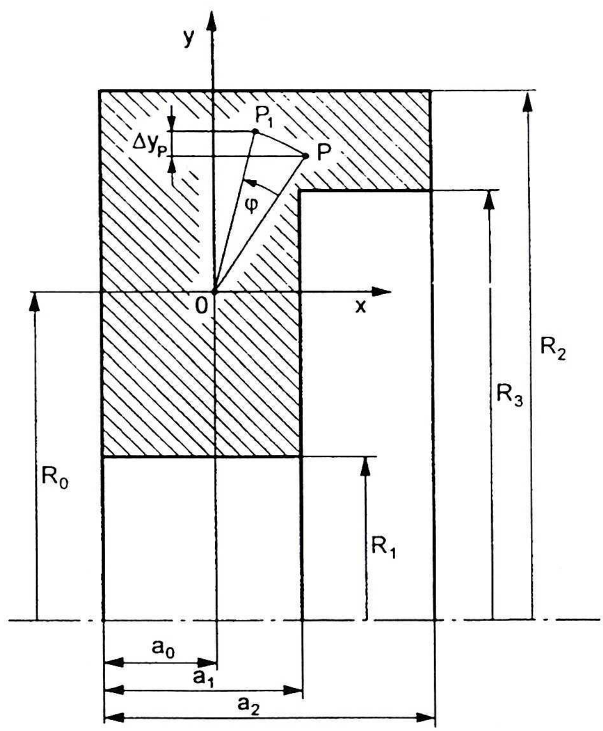

Fig. 4. A cross section of a flange

An arbitrary point P of the cross section shall move to its new location – the point P_1, in result of rotation of the flange by a certain angle φ with respect to the rotation center – the point O (Fig. 4). Coordinates of the point P_1 are as follows

$$x_{P1} = x_P \cos\varphi - y_P \sin\varphi, \qquad y_{P1} = x_P \sin\varphi + y_P \cos\varphi. \tag{28}$$

Hence, increment of the coordinate y_P is $\Delta y_P = y_{P1} - y_P = x_P \sin\varphi - y_P(1 - \cos\varphi)$. For the small rotation angle φ ($\sin\varphi \cong \varphi$ and $\cos\varphi \cong 1 - \frac{1}{2}\varphi^2$), this increment takes the following form

$$\Delta y_P = \varphi \cdot x_P - \frac{1}{2}\varphi^2 \cdot y_P. \tag{29}$$

Planar cross sections of the flange before rotation remaining planar after the rotation, the strain of any circumferential fiber, defined by coordinates x and y, are determined as

$$\varepsilon = \frac{x}{R_0 + y}\varphi - \frac{1}{2}\frac{y}{R_0 + y}\varphi^2, \tag{30}$$

where R_0 is the radius of the rotation center.

Location of the center of rotation is determined by two dimensions a_0 and R_0. The condition of zeroing the circumferential normal force in the cross section of the flange gives

$$N = E_f \int_\Omega \varepsilon dxdy = 0, \quad \text{where} \quad E_f - \text{Young's modulus.} \tag{31}$$

Integration at the area of the cross section (Ω) gives two equations:

$$\left[\left(a_1^2 - 2a_1 a_0\right)\ln\left(R_3/R_1\right) + \left(a_2^2 - 2a_2 a_0\right)\ln\left(R_2/R_3\right)\right]\varphi = 0,$$

$$\{a_1[R_3 - R_1 - R_0 \ln(R_3/R_1)] + a_2[R_2 - R_3 - R_0 \ln(R_2/R_3)]\}\varphi^2 = 0,$$

from which

$$a_0 = \frac{1}{2}\frac{a_1^2 \ln(R_3/R_1) + a_2^2 \ln(R_2/R_3)}{a_1 \ln(R_3/R_1) + a_2 \ln(R_2/R_3)},$$

$$R_0 = \frac{a_1(R_3 - R_1) + a_2(R_2 - R_3)}{a_1 \ln(R_3/R_1 + a_2 \ln(R_2/R_3)}. \tag{32}$$

The bolt forces induce flange rotation by a certain angle φ. Thus, the flange is subject to a uniformly distributed moment. The bending moment at any cross section of the bolt is

$$M_0 = E_f \int_\Omega x\varepsilon dxdy \,. \tag{33}$$

Integration at the area of the cross section (Ω) gives

$$M_0 = E_f\left(C_{f1}\cdot\varphi + C_{f2}\cdot\varphi^2\right), \tag{34}$$

where

$$C_{f1} = \frac{1}{3}\left[a_1\left(a_1^2 - 3a_1a_0 + 3a_0^2\right)\ln\left(R_3/R_1\right) + a_2\left(a_2^2 - 3a_2a_0 + 3a_0^2\right)\ln\left(R_2/R_3\right)\right],$$

$$C_{f2} = \frac{1}{4}a_1\left(a_2 - a_1\right)\left[R_3 - R_1 - R_0\ln\left(R_3/R_1\right)\right].$$

Hence, the rotation angle for small φ is expressed by

$$\varphi = \frac{M_0}{E_f C_{f1}}. \tag{35}$$

Maximum of the coefficient C_{f1} gives minimum of the rotation angle φ, hence, the shaping criterion takes the form

$$\max_{R_{32}}\{C_{f1}\}, \tag{36}$$

where $R_{32} = R_3/R_2$, in the range $R_1/R_2 \le R_{32} < 1$.

The volume of the flange (27) is constant $V_f = \text{const}$, that gives

$$a_1 = \frac{V_f}{\pi R_2^2\left[r_{32}^2 - r_{12}^2 + \left(1 - r_{32}^2\right)\alpha_{21}\right]}, \qquad \text{where} \quad \alpha_{21} = \frac{a_2}{a_1}. \tag{37}$$

Numerical investigation was carried out for the following example data: $V_f = 22\cdot 10^5\ \text{mm}^3$, $R_1 = 100$ mm, $R_2 = 195$ mm, $\alpha_{21} = 1.0; 2.5$. The changes in the C_{f1} (R_{32}) coefficient considered as a function are shown in Fig. 5. For $\alpha_{21} = 1$, i.e. for a rectangular cross section, this function is constant ($C_{f1} = 868.3\ \text{mm}^3$). For $1 < \alpha_{21}$ maximum of the function occurs, and, in particular, for $\alpha_{21} = 2.5$, the maximum $C_{f1,\max} = 1681.4\ \text{mm}^3$ for $R_{32,opt} = 0.91817$. In this case the flange rotation angle for its generalized cross section is almost twice smaller than for a classical flange shape. Growing parameter α_{21} leads to the increase of maximum $C_{f1,\max}$, but the thickness a_1 decreases. Optimal shapes of a flanged bolted joint for example numerical data are shown in Fig. 6.

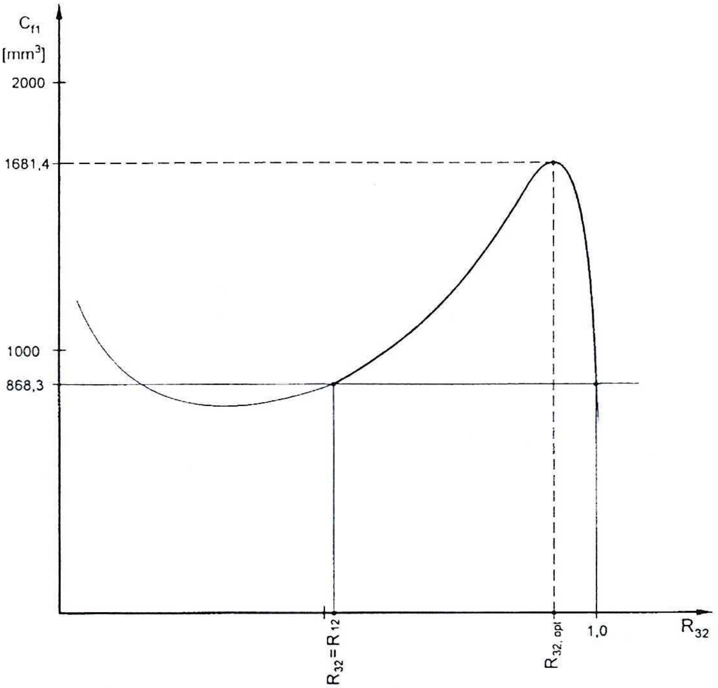

Fig. 5. The coefficient C_{f1} as a function of the dimensionless parameter R_{32}

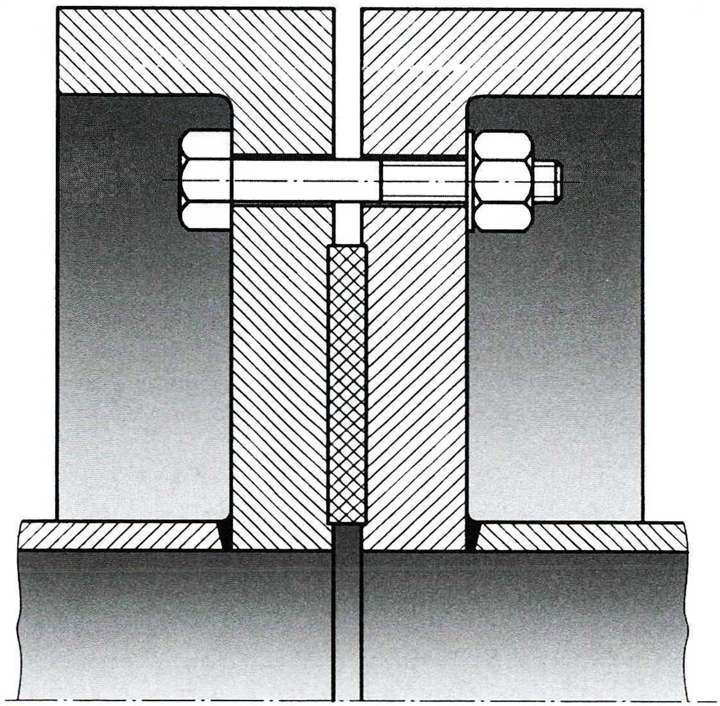

Fig. 6. A eross section of the bolted flanget hoint

4. Conclusions

Numerical studies of an analytical model of the flanged bolted joint with flat ring gasket enable us to formulte the following two conclusions:

- an optimal shape of rectangular cross section of the gasket of constant volume may be found for which the bolt tension required for the joint tightness is minimal.
- a generalized optimal shape of a cross section of the flange of constant volume may be found for which its rotation angle is minimal.

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Efektywne projektowanie połączenia kołnierzowego śrubowego z płaską pierścieniową uszczelką

Streszczenie

Przedmiotem pracy jest połączenie kołnierzowe śrubowe z płaską pierścieniową uszczelką. Sformułowane są proste modele matematyczne dla kołnierza i uszczelki. Rozwiązanie tych modeli umożliwiło na określenie efektywnych kształtów płaskiej uszczelki i przekroju poprzecznego kołnierza. Kryterium w kształtowaniu uszczelki przyjęto jako minimalny naciąg śrub, natomiast w kształtowaniu przekroju kołnierza przyjęto minimalny kąt jego obrotu. Wyniki tych badań są pokazane na rysunkach i mogą być przydatne w praktycznym projektowaniu połączeń rur i naczyń ciśnieniowych.