ANALYTICAL STUDY OF THE STATIC THERMOMECHANICAL STRESSES OF THE ASSEMBLIES WITH OPTIONAL RING FLANGES. ROTATION OF THE FLANGE RING AROUND THE CIRCUMFERENCE OF CENTERS FOR BOLT HOLES

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Abstract: This paper addresses the evaluation of thermal and mechanical stresses that occur in assemblies with flat ring flanges, of optional type. The flange rings are fixed by welding to the wall of the cylindrical body of the vessel, with a constant thickness along its length. Regarding of the above, the compatibility of the deformations of the component elements (radial displacements and rotations) is approached. A linear algebraic system is formed in which both external loads (pressure, temperature) and connecting loads (bending unit moments and unit forces) are present. The present analysis discusses the quantitative, cumulative effect of the deformed gasket and the stiffness of the curved/bent screws on the tightness of the system. The methodology is flexible by introducing selection factors so that the mentioned influences can be easily separated.

Keywords: circular flat flange assemblies, stresses, deformations

1. INTRODUCTION

The growing need to chemically or mechanically process various chemicals has led to the creation of complex mechanical equipment that can withstand low or high working parameters. In this context, a very important role is taken by the sealing of liquid or gaseous substances processed in static or dynamic equipment, for chemical and/or mechanical processing. The practical constructive types are particularly diverse: flat or neck ring flanges [1 - 39], respectively ring flanges with radial ribs [40 - 46], tightened with screws, with clips or with clamps [47 - 50], with ring seals flat [51 - 53] or lenticular [54 - 56], respectively without gaskets [57 - 60]). The choice of the material of a sealing gasket is a difficult problem, because it must meet a number of extremely important conditions, such as: a) be stable at working temperature, pressure and chemical and mechanical aggressiveness of the technological environment, while maintaining its economic characteristics; c) have resistance to friction and wear; e) deform elastoplastic, when the gasket is tight, in order to completely fill the micro-asperities of the sealing surfaces. For its importance, recognized experimental research has been carried out over time [61 - 67], as well as appropriate theoretical studies [39, 68, 69], for example, for the evaluation of the sealing pressure between the flange rings, in the radial and circumferential direction.

In this sense, the study of flange assemblies can be performed taking into account static and/or dynamic loads, simple or combined (pressures, temperatures, transient regime, fatigue, creep, earthquake) [70 - 79], as well as corrosive and/or erosive aggression for metallic or non-metallic materials (polymeric or composite). The theoretically established mathematical expressions clearly show the influence of the rigidity of the assembly on

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the tightness, the reason for which, as much as possible, a diametric gauge of the ring as small as possible is imposed, simultaneously with the increase of its thickness. In the same sense, the possible increase in the number of tightening screws is sought. All the elements mentioned above have in view, simultaneously, the assurance of a safe operation, in specific conditions, but also the existence of a minimum consumption of construction material, in this case the lowest possible price of construction material, but also of labor and maintenance.

2. STUDY HYPOTHESES

The following analytical study considers the following simplifying hypotheses:
1. The rings of the flat flanges are fixed to the cylindrical shell (with constant thickness) by welding (Figure 1); the cylindrical shell adjacent to the flange ring, along its entire thickness, is considered to have length $h_c < \ell_{sc} \approx 1.77 \cdot \sqrt{D_{mc} \cdot \delta}$, to use the theory of short structural elements; the thickness of the cylindrical shell is accepted constant throughout the deformation;

2. An assembly with flanges (rings) identical in terms of geometry (flat plates) and material is considered;

3. The effect of the welding cords for fixing the cylindrical ferrule with the flange ring on the general state of stress is neglected;

4. The metallic material of the assembly components is considered homogeneous, continuous and isotropic, required in the elastic field;

5. External loads (pressure, temperature) have static values;

6. Deformation of the rings in the radial direction and their rotation, according to the accepted hypothesis, is performed in a monobloc configuration; the effect of the screw holes on the rigidity of the flange ring is neglected;

Note: The rotation of the flange ring, in this case, occurs around the circumference of the centers of the screw holes, defined by the diameter D_s (Figure 1).

7. The value of the rotation angle of the flange ring is conditioned by the intensity of the internal connection loads, together with the effect of the external loads: a) the unit force resistant to the tendency of the expulsion of the gasket [27, 39]; b) by the bending, resistant unit moments, m_g, by the eccentric compression of the gasket, respectively m_s present due to the curvature of the screws [7, 9, 40 - 42], all related to the inner circumference of the ring;

8. A uniform radial compression of the gasket for sealing and, at the same time, an identical tightening of the screws, by using a specific methodology and suitable systems is considered [31, 35, 80-82];

9. Accepting the ring of a flange as a flat ring plate, with a constant thickness over its entire width, simply supported along the circumference of the screw centers, characterized by the diameter (Figure 1), and by the loads mentioned in point 7 (above), sets the following expression for the angle of rotation (around points C - Figure 2):

$$\vartheta_f^{\bullet\bullet} = \frac{1}{k_{gs}^{\bullet\bullet}} \cdot \Big\{ -M_{01}^{\bullet\bullet} + 0.5 \cdot h_f \cdot Q_{01}^{\bullet\bullet} + M_{02}^{\bullet\bullet} - 0.5 \cdot h_f \cdot Q_{02}^{\bullet\bullet} -$$
$$-0.5 \cdot \Big[\left(D_2 - D_{if} \right) \cdot \left(D_2 / D_{if} \right) + \mu_{fg} \cdot h_f \cdot c_{\vartheta}^{fg} \Big] \cdot \overline{P}_2 + 0.5 \cdot \left(D_s - D_{if} \right) \cdot \overline{P}_s -$$
$$- 0.5 \cdot \Big[\left(D_{mg} - D_{if} \right) \cdot \left(D_{mg} / D_{if} \right) + \mu_{fg} \cdot h_f \cdot c_{\vartheta}^{fg} \Big] \cdot \overline{P}_g \Big\}, \quad (1)$$

where:

$$k_{gs}^{\bullet\bullet} = \frac{1}{k_f^{\bullet\bullet}} + \frac{n_s \cdot E_s \cdot d_s^4 \cdot D_i \cdot c_\vartheta^s}{\left(6 \cdot \ell_s \cdot D_s^2\right)} + \frac{E_g \cdot c_g^3 \cdot D_{mg} \cdot \left(D_{eg} - D_{ig}\right)^3 \cdot D_{if} \cdot c_\vartheta^g}{\left(48 \cdot h_g \cdot D_{mg}\right)} \tag{2}$$

$$k_f^{\bullet\bullet} = 0.5 \cdot D_s^2 \cdot \frac{\left[1 - \nu_f + \left(1 + \nu_f\right)\alpha_{Df}\right]}{\left[\alpha_{Df} \cdot \left(\alpha_{Df}^2 - 1\right) \cdot \left(1 - \nu_f^2\right) \cdot \Re_f \cdot D_{if}\right]} \tag{3}$$

$$\alpha_{Df} = \frac{D_s}{D_{if}}$$

From the expression of the continuity of the deformations it is found that the effects of the previously mentioned quantities lead to the diminution of the total rotation of the flange ring, favoring the tightness of the assembly.

In the structure of factor $k_{gs}^{\bullet\bullet}$ the selection factors c_ϑ^g, c_ϑ^s, c_ϑ^{fg} are introduced, allowing to easily notice the influences of the "resisting" quantities to the deformation of the flange ring (when the mentioned coefficients take values equal to unity, the influences are present; when the values are zero, those influences disappear).

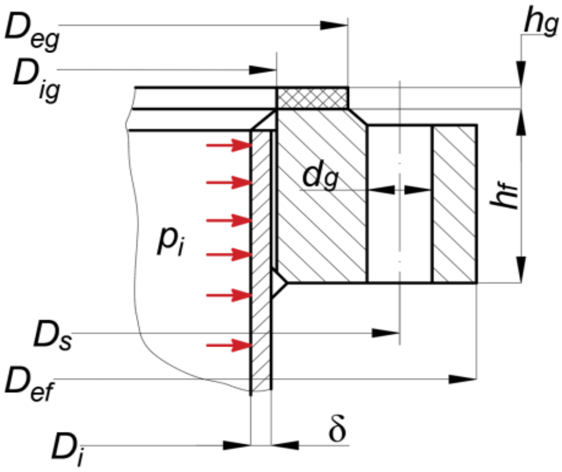

Fig. 1. Ring flat flange type A (dimensional characteristics - diagram).

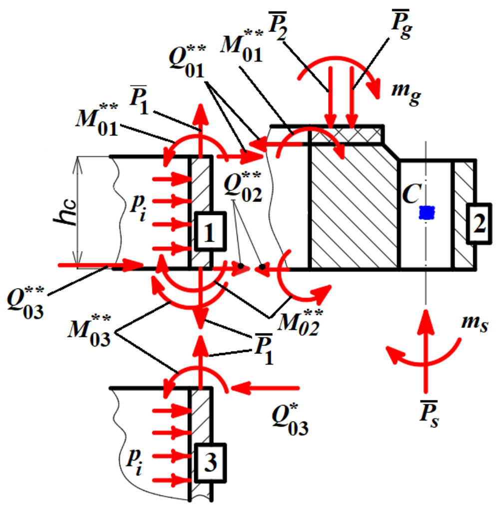

Fig. 2. Decomposition (hypothetical) of flange assembly elements (diagram):
1 - cylindrical shell adjacent to the flange ring; 2 - flange ring; 3 - cylindrical shell (vessel body).

3. ANALYTICAL STUDY. CONNECTION LOADS

To establish the expressions of the unitary bending moments and of the unitary connecting forces, the assembly is decomposed (fictitiously) into the component elements (Figure 2). In this sense, the equations of compatibility of deformations between the elements mentioned above are written as follows:

► connection between elements 1 - 2 (top):

- radial displacements:

$$a_{11}^{\bullet\bullet} \cdot Q_{01}^{\bullet\bullet} + a_{12}^{\bullet\bullet} \cdot M_{01}^{\bullet\bullet} + a_{13}^{\bullet\bullet} \cdot Q_{02}^{\bullet\bullet} + a_{14}^{\bullet\bullet} \cdot M_{02}^{\bullet\bullet} + a_{15}^{\bullet\bullet} \cdot Q_{03}^{\bullet\bullet} + a_{16}^{\bullet\bullet} \cdot M_{03}^{\bullet\bullet} = b_{1}^{\bullet\bullet} \tag{4}$$

- rotations:

$$a_{21}^{\bullet\bullet} \cdot Q_{01}^{\bullet\bullet} + a_{22}^{\bullet\bullet} \cdot M_{01}^{\bullet\bullet} + a_{23}^{\bullet\bullet} \cdot Q_{02}^{\bullet\bullet} + a_{24}^{\bullet\bullet} \cdot M_{02}^{\bullet\bullet} + a_{25}^{\bullet\bullet} \cdot Q_{03}^{\bullet\bullet} + a_{26}^{\bullet\bullet} \cdot M_{03}^{\bullet\bullet} = b_{2}^{\bullet\bullet} \tag{5}$$

► connection between elements 1 - 2 (bottom):

- radial displacements:

$$a_{31}^{\bullet\bullet} \cdot Q_{01} + a_{32}^{\bullet\bullet} \cdot M_{01} + a_{33}^{\bullet\bullet} \cdot Q_{02} + a_{34}^{\bullet\bullet} \cdot M_{02} + a_{35}^{\bullet\bullet} \cdot Q_{03} + a_{36}^{\bullet\bullet} \cdot M_{03} = b_{3}^{\bullet\bullet} \tag{6}$$

- rotations:

$$a_{41}^{\bullet\bullet} \cdot Q_{01}^{\bullet\bullet} + a_{42}^{\bullet\bullet} \cdot M_{01}^{\bullet\bullet} + a_{43}^{\bullet\bullet} \cdot Q_{02}^{\bullet\bullet} + a_{44}^{\bullet\bullet} \cdot M_{02}^{\bullet\bullet} + a_{45}^{\bullet\bullet} \cdot Q_{03}^{\bullet\bullet} + a_{46}^{\bullet\bullet} \cdot M_{03}^{\bullet\bullet} = b_{4}^{\bullet\bullet} \tag{7}$$

► connection between elements 1 –3:

- radial displacements

$$a_{51}^{\bullet\bullet} \cdot Q_{01}^{\bullet\bullet} + a_{52}^{\bullet\bullet} \cdot M_{01}^{\bullet\bullet} + a_{53}^{\bullet\bullet} \cdot Q_{02}^{\bullet\bullet} + a_{54}^{\bullet\bullet} \cdot M_{02}^{\bullet\bullet} + a_{55}^{\bullet\bullet} \cdot Q_{03}^{\bullet\bullet} + a_{56}^{\bullet\bullet} \cdot M_{03}^{\bullet\bullet} = b_{5}^{\bullet\bullet} \tag{8}$$

- rotations:

$$a_{61}^{\bullet\bullet} \cdot Q_{01}^{\bullet\bullet} + a_{62}^{\bullet\bullet} \cdot M_{01}^{\bullet\bullet} + a_{63}^{\bullet\bullet} \cdot Q_{02}^{\bullet\bullet} + a_{64}^{\bullet\bullet} \cdot M_{02}^{\bullet\bullet} + a_{65}^{\bullet\bullet} \cdot Q_{03}^{\bullet\bullet} + a_{66}^{\bullet\bullet} \cdot M_{03}^{\bullet\bullet} = b_{6}^{\bullet\bullet} \tag{9}$$

Equalities (4) - (9) form the algebraic system written in the form:

$$\left[A^{\bullet\bullet}\right] \cdot \left\{S_{l}^{\bullet\bullet}\right\} = \left\{T_{l}^{\bullet\bullet}\right\} \tag{10}$$

where:

$$\left[A^{\bullet\bullet}\right] = \begin{bmatrix} a_{11}^{\bullet\bullet} & a_{12}^{\bullet\bullet} & \cdots & a_{16}^{\bullet\bullet} \\ a_{21}^{\bullet\bullet} & a_{22}^{\bullet\bullet} & \cdots & a_{26}^{\bullet\bullet} \\ \cdots & \cdots & \cdots & \cdots \\ a_{61}^{\bullet\bullet} & a_{62}^{\bullet\bullet} & \cdots & a_{66}^{\bullet\bullet} \end{bmatrix} \tag{11}$$

represents the matrix of influencing factors $a_{ij}^{\bullet\bullet}$ ($i = 1 \cdots 6$; $j = 1 \cdots 6$);the transposed vector of the connection load.

$$\left\{S_{l}^{\bullet\bullet}\right\} = \left\{Q_{01}^{\bullet\bullet} \quad M_{01}^{\bullet\bullet} \quad Q_{02}^{\bullet\bullet} \quad M_{02}^{\bullet\bullet} \quad Q_{03}^{\bullet\bullet} \quad M_{03}^{\bullet\bullet}\right\}^{T} \tag{12}$$

transposed vector of free terms (radial displacements and rotations under the action of external loads - pressure, temperature):

$$\left\{T_l^{\bullet\bullet}\right\} = \left\{b_1^{\bullet\bullet} \quad b_2^{\bullet\bullet} \quad \ldots \quad b_6^{\bullet\bullet}\right\}^T \tag{13}$$

From equality (9) the way of evaluating the values of the unknowns of the present problem - unitary shear forces $Q_{0k}^{\bullet\bullet}$ $(k = 1 \cdots 3)$ and unitary bending moments $M_{0k}^{\bullet\bullet}$ $(k = 1 \cdots 3)$ is deduced- written in the form

$$\left\{S_l^{\bullet\bullet}\right\} = \left[A^{\bullet\bullet}\right]^{-1} \cdot \left\{T_l^{\bullet\bullet}\right\} \tag{14}$$

where $\left[A^{\bullet\bullet}\right]^{-1}$ represents the inverse of the $\left[A^{\bullet\bullet}\right]$ matrix, whose determinant has a value other than zero.

The expressions of the influencing factors from the equalities (4) - (9) have the forms:

$$a_{11}^{\bullet\bullet} = f_{1q} \Big/ \left(2 \cdot k_c^3 \cdot \Re_c\right) - k_{wf} - h_f^2 \Big/ \left(4 \cdot k_{gs}^{\bullet\bullet}\right); \quad a_{12}^{\bullet\bullet} = f_{1m} \Big/ \left(2 \cdot k_c^2 \cdot \Re_c\right) + h_f \Big/ \left(2 \cdot k_{gs}^{\bullet\bullet}\right);$$
$$a_{13}^{\bullet\bullet} = f_{qd}\left(h_c\right) \Big/ \left(2 \cdot k_c^3 \cdot \Re_c\right) - k_{wf} + h_f^2 \Big/ \left(4 \cdot k_{gs}^{\bullet\bullet}\right);$$
$$a_{14}^{\bullet\bullet} = f_{md}\left(h_c\right) \Big/ \left(2 \cdot k_c^2 \cdot \Re_c\right) - h_f \Big/ \left(2 \cdot k_{gs}^{\bullet\bullet}\right); \; a_{15}^{\bullet} = f_{qd}\left(h_c\right) \Big/ \left(2 \cdot k_c^3 \cdot \Re_c\right);$$
$$a_{16}^{\bullet} = f_{md}\left(h_c\right) \Big/ \left(2 \cdot k_c^2 \cdot \Re_c\right); \; a_{21}^{\bullet\bullet} = -f_{23q} \Big/ \left(2 \cdot k_c^2 \cdot \Re_c\right) - h_f \Big/ \left(2 \cdot k_{gs}^{\bullet\bullet}\right);$$
$$a_{22}^{\bullet\bullet} = -f_{2m} \Big/ \left(k_c \cdot \Re_c\right) + 1 \big/ k_{gs}^{\bullet\bullet}; \; a_{23}^{\bullet} = f_{qr}\left(h_c\right) \Big/ \left(2 \cdot k_c^2 \cdot \Re_c\right) + h_f \Big/ \left(2 \cdot k_{gs}^{\bullet\bullet}\right);$$
$$a_{24}^{\bullet\bullet} = f_{mr}\left(h_c\right) \Big/ \left(2 \cdot k_c \cdot \Re_c\right) - 1 \big/ k_{gs}^{\bullet\bullet}; \; a_{25}^{\bullet\bullet} = f_{qr}\left(h_c\right) \Big/ \left(2 \cdot k_c^2 \cdot \Re_c\right);$$
$$a_{26}^{\bullet\bullet} = f_{mr}\left(h_c\right) \Big/ \left(2 \cdot k_c \cdot \Re_c\right); \; a_{31}^{\bullet\bullet} = f_{qd}\left(h_c\right) \Big/ \left(2 \cdot k_c^3 \cdot \Re_c\right) - k_{wf} + h_f \Big/ \left(4 \cdot k_{gs}^{\bullet\bullet}\right);$$
$$a_{32}^{\bullet\bullet} = f_{md}\left(h_c\right) \Big/ \left(2 \cdot k_c^2 \cdot \Re_c\right) - h_f \Big/ \left(2 \cdot k_{gs}^{\bullet\bullet}\right);$$
$$a_{33}^{\bullet\bullet} = f_{1q} \Big/ \left(2 \cdot k_c^3 \cdot \Re_c\right) - k_{wf} - h_f^2 \Big/ \left(4 \cdot k_{gs}^{\bullet\bullet}\right); \; a_{34}^{\bullet} = f_{1m} \Big/ \left(2 \cdot k_c^2 \cdot \Re_c\right) + h_f \Big/ \left(2 \cdot k_{gs}^{\bullet\bullet}\right);$$
$$a_{35}^{\bullet\bullet} = f_{1q} \Big/ \left(2 \cdot k_c^3 \cdot \Re_c\right); \; a_{36}^{\bullet\bullet} = f_{1m} \Big/ \left(2 \cdot k_c^2 \cdot \Re_c\right);$$
$$a_{41}^{\bullet\bullet} = f_{qr}\left(h_c\right) \Big/ \left(2 \cdot k_c^2 \cdot \Re_c\right) - h_f \Big/ \left(2 \cdot k_{gs}^{\bullet\bullet}\right); \; a_{42}^{\bullet\bullet} = f_{mr}\left(h_c\right) \Big/ \left(2 \cdot k_c \cdot \Re_c\right) + 1 \big/ k_{gs}^{\bullet\bullet};$$
$$a_{43}^{\bullet\bullet} = -f_{23q} \Big/ \left(2 \cdot k_c^2 \cdot \Re_c\right) + h_f \Big/ \left(2 \cdot k_{gs}^{\bullet\bullet}\right);$$
$$a_{44}^{\bullet\bullet} = -f_{2m} \Big/ \left(k_c \cdot \Re_c\right) - 1 \big/ k_{gs}^{\bullet\bullet}; \; a_{45}^{\bullet\bullet} = -f_{23q} \Big/ \left(2 \cdot k_c^2 \cdot \Re_c\right);$$
$$a_{46}^{\bullet\bullet} = -f_{2m} \Big/ \left(k_c \cdot \Re_c\right); \; a_{51}^{\bullet\bullet} = f_{qd}\left(h_c\right) \Big/ \left(2 \cdot k_c^3 \cdot \Re_c\right);$$
$$a_{52}^{\bullet} = f_{md}\left(h_c\right) \Big/ \left(2 \cdot k_c^2 \cdot \Re_c\right); \; a_{53}^{\bullet\bullet} = f_{1q} \Big/ \left(2 \cdot k_c^3 \cdot \Re_c\right); \; a_{54}^{\bullet} = f_{1m} \Big/ \left(2 \cdot k_c^2 \cdot \Re_c\right);$$
$$a_{55}^{\bullet} = \left(f_{1q} - 1\right) \Big/ \left(2 \cdot k_c^3 \cdot \Re_c\right); \; a_{56}^{\bullet\bullet} = \left(f_{1m} - 1\right) \Big/ \left(2 \cdot k_c^2 \cdot \Re_c\right);$$
$$a_{61}^{\bullet\bullet} = f_{qr}\left(h_c\right) \Big/ \left(2 \cdot k_c^2 \cdot \Re_c\right); \; a_{62}^{\bullet\bullet} = f_{mr}\left(h_c\right) \Big/ \left(2 \cdot k_c \cdot \Re_c\right);$$
$$a_{63}^{\bullet\bullet} = -f_{23q} \Big/ \left(2 \cdot k_c^2 \cdot \Re_c\right); \; a_{64}^{\bullet} = -f_{2m} \Big/ \left(k_c \cdot \Re_c\right);$$
$$a_{65}^{\bullet\bullet} = \left(1 - f_{23q}\right) \Big/ \left(2 \cdot k_c^2 \cdot \Re_c\right); \; a_{66}^{\bullet} = \left(1 - f_{2m}\right) \Big/ \left(k_c \cdot \Re_c\right), \tag{15}$$

while the vectors of radial displacements and rotations have the configurations:

$$b_1^{\bullet\bullet} = \left[p_i \Big/ \left(4 \cdot k_c^4 \cdot \Re_c \right) \right] \cdot \left[1 - \mu_c \cdot D_i^2 \Big/ \left(2 \cdot D_{mc}^2 \right) \right] + 0{,}5 \cdot \alpha_c \cdot D_{mc} \cdot \Delta T_c +$$
$$+ \left[h_f \Big/ \left(4 \cdot k_{gs}^{\bullet} \right) \right] \cdot \{ - \left[\left(D_2 - D_{if} \right) \cdot \left(D_2 / D_{if} \right) + \mu_{fg} \cdot h_f \cdot c_{\vartheta}^{fg} \right] \cdot \overline{P}_2 +$$
$$+ \left(D_{ef} - D_s \right) \cdot \overline{P}_s - \left[\left(D_{mg} - D_{if} \right) \cdot \left(D_{mg} / D_{if} \right) + \mu_{fg} \cdot h_f \cdot c_{\vartheta}^{fg} \right] \cdot \overline{P}_g \Big\} -$$
$$- \Delta R \left(\Delta T_f \right)_{D = D_{if}} ;$$
$$b_2^{\bullet\bullet} = \left[1 \Big/ \left(2 \cdot k_{gs}^{\bullet\bullet} \right) \right] \cdot \Big\{ - \left[\left(D_2 - D_{if} \right) \cdot \left(D_2 / D_{if} \right) + \mu_{fg} \cdot h_f \cdot c_{\vartheta}^{fg} \right] \cdot \overline{P}_2 +$$
$$+ \left(D_s - D_{if} \right) \cdot \overline{P}_s - \left[\left(D_{mg} - D_{if} \right) \cdot \left(D_{mg} / D_{if} \right) + \mu_{fg} \cdot h_f \cdot c_{\vartheta}^{fg} \right] \cdot \overline{P}_g \Big\} ;$$
$$b_3^{\bullet\bullet} = \left[p_i \Big/ \left(4 \cdot k_c^4 \cdot \Re_c \right) \right] \cdot \left[1 - \mu_c \cdot D_i^2 \Big/ \left(2 \cdot D_{mc}^2 \right) \right] + 0{,}5 \cdot \alpha_c \cdot D_{mc} \cdot \Delta T_c -$$
$$- \left[h_f \Big/ \left(4 \cdot k_{gs}^{\bullet\bullet} \right) \right] \cdot \Big\{ - \left[\left(D_2 - D_{if} \right) \cdot \left(D_2 / D_{if} \right) + \mu_{fg} \cdot h_f \cdot c_{\vartheta}^{fg} \right] \cdot \overline{P}_2 +$$
$$- \left[\left(D_{mg} - D_{if} \right) \cdot \left(D_{mg} / D_{if} \right) + \mu_{fg} \cdot h_f \cdot c_{\vartheta}^{fg} \right] \cdot \overline{P}_g + \left(D_s - D_{if} \right) \cdot \overline{P}_s \Big\} -$$
$$- \Delta R \left(\Delta T_f \right)_{D = D_{if}} ;$$
$$b_4^{\bullet\bullet} = \left[1 \Big/ \left(2 \cdot k_{gs}^{\bullet\bullet} \right) \right] \cdot \Big\{ - \left[\left(D_2 - D_{if} \right) \cdot \left(D_2 / D_{if} \right) + \mu_{fg} \cdot h_f \cdot c_{\vartheta}^{fg} \right] \cdot \overline{P}_2 +$$
$$+ \left(D_s - D_{if} \right) \cdot \overline{P}_s - \left[\left(D_{mg} - D_{if} \right) \cdot \left(D_{mg} / D_{if} \right) + \mu_{fg} \cdot h_f \cdot c_{\vartheta}^{fg} \right] \cdot \overline{P}_g \Big\} ;$$
$$b_5^{\bullet\bullet} = 0 ; \; b_6^{\bullet\bullet} = 0 \tag{16}$$

In the above equations the auxiliary values (the corresponding geometric characteristics are noted in Figures 1 and 2) are found:

$$P_1 = 0.25 \cdot \pi \cdot D_{ec}^2 \cdot p_i ; \; \overline{P}_1 = P_1 \Big/ \left(\pi \cdot D_{mc} \right) ; \; P_2 = 0.25 \cdot \pi \cdot \left(D_{mg}^2 - D_{ec}^2 \right) \cdot p_i ;$$
$$\overline{P}_2 = 0.5 \cdot \left(D_{mg} - D_{ec} \right) \cdot p_i ; \; P_g = 0.25 \cdot \pi \cdot \left(D_{eg}^2 - D_{ig}^2 \right) \cdot p_{sg} ; \; D_{ec} = D_i + 2 \cdot \delta ;$$
$$\overline{P}_g = 0.5 \cdot \left(D_{eg} - D_{ig} \right) \cdot p_{sg} ; \; \Re_c = E_c \cdot \delta^3 / \sqrt{12 \cdot \left(1 - \mu_c^2 \right)} ; \; k_c = \sqrt[4]{12 \cdot \left(1 - \mu_c^2 \right)} \Big/ \sqrt{D_{mc} \cdot \delta} ;$$
$$D_{mc} = D_i + \delta ; \; D_{mg} = 0.5 \cdot \left(D_{ig} + D_{eg} \right) ; \; D_{mf} = 0.5 \cdot \left(D_{if} + D_{ef} \right) ;$$
$$k_{wf} = 0.5 \cdot D_{if} \cdot \left[\left(1 - \mu_f \right) \cdot D_{ef}^2 - \left(1 - 2 \cdot \mu_f \right) \cdot D_{if}^2 \right] \Big/ \left[h_f \cdot E_f \cdot \left(D_{ef}^2 - D_{if}^2 \right) \right] \tag{17}$$

where E_c, E_f, E_g, E_s - the modules of longitudinal elasticity of the materials of the cylindrical shell, of the flange ring, of the sealing gasket, respectively of the screws material, N/mm^2;

$\overline{F}_{fg} = \mu_{fg} \cdot \left(\overline{P}_g + \overline{P}_2 \right)$ - unitary friction force between flange ring and sealing gasket, N/mm; P_s - the force developed in the screws of the flange assembly, in different operating conditions; $\overline{P}_s$ - the existing unit force on the diameter circumference D_s; c_g - the "reduction" coefficient of the initial gasket width, which can be chosen considering recommendations from the literature; $c_{\vartheta}^{g}, c_{\vartheta}^{s}, c_{\vartheta}^{fg}$ - selection factors used to store or eliminate the influence of gasket rotation, bolt rotation, respectively the friction between gasket and flange ring (accepting the value equal to zero); d_s - nominal diameter of the screw, mm; $f_{1m}, f_{2m}, f_{md}\left(h_c \right), f_{mr}\left(h_c \right), f_{1q}, f_{23q}, f_{qd}\left(h_c \right), f_{qr}\left(h_c \right)$ - factors of influence of the connecting loads (unitary shear forces and bending unitary moments) [74]; $k_{gs}^{\bullet}$ - influence factor, with significant force, of the quantities that can diminish the value of the rotation of the flange ring under the action of the loads given by the value of the internal pressure and of the connection loads, respectively the eccentric compression of the sealing gasket, the rotation tendency of the screws and ring, N; k_{wf} - influencing factor the unitary shear forces (for connection) on the width of the flange ring, mm^2/N; l_s - calculation length of the screws, mm; l_{sc} - half wavelength, mm; n_s - the number of flange assembly bolts; $\Re_c$ - cylindrical bending stiffness of pressure vessel body, N·mm; ΔT_c, ΔT_f - the thermal gradient of the vessel body, related to the temperature of the external environment, respectively the thermal gradient of the flange ring at its inner surface, K; $\Delta R\left(\Delta T_f\right)_{D=D_{if}}$ - radial displacement of the flange ring under the influence of the thermal gradient, for the accepted law for temperature variation [75]; α_c, α_f - thermal deformation factor for cylindrical shell and for ring flange, K^{-1}; μ_c, μ_f - the Poisson coefficient (of the transverse contraction) of the cylindrical shell material, the flange ring material; μ_{fg} - the coefficient of friction between the sealing gasket and the flange ring.

4. CONCLUSIONS

Based on the previous relations, establishing the expressions of the connection loads between the considered assembly elements, hypothetically separated (Figure 2) it is possible to deduce the expression of the flange ring rotation, taking into account the equalities (1) - (3). The value of this angle provides an assessment of the tightness of the system. In an unsuitable state, the geometry of the assembly components is modified (increasing the thickness of the shell on a certain length, greater than the length of the half-wave, under the flange ring, respectively increasing the thickness of the considered flange ring).

The results of the theoretical and experimental works are known, which indicate that the state of tension is more intense along the length of the cylindrical ferrule 3 (Figure 2), below the plane of connection with the flange ring. Once the values for $Q_{03}^{\bullet\bullet}$ and $M_{03}^{\bullet\bullet}$ are known and taking into account the effects of pressure and temperature, we proceed to evaluate the variable stresses on the inner and outer surface of the shell of the cylindrical body [3, 5]. The maximum equivalent stress, established by using the 5th resistance theory, is compared with the allowable strength of the metallic material. If the resistance condition is met, the analytical study is completed. Otherwise, the geometry of the assembly is remodeled until the accepted technical safety is confirmed.

An adequate calculation program can lead to the optimization of the construction, with the minimization of the material consumption and the assurance of a safe operation. The previously exposed methodology allows, at the same time, an evaluation of the stresses and by using the discrete values of the external loads in case of a transient regime of the pressure, respectively of the temperature. The results obtained on the exposed path can be developed in further research on the behavior of flange assemblies in areas such as creep, fatigue, the effect of residual stresses in weld seams, or the prevention of cracking [83 - 86].

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