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Load condition analysis of pipe flange connection with gasket flat gasket and loose clamping rings

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ABSTRACT

Purpose: This paper describes the study of the connection with loose fixing composite rings of pipe flanges. The elastic properties of the joint components - gasket, pressure flanges, rings and fixing bolts were determined experimentally.

Design/methodology/approach: The criterion of joint tightness was formulated. The constructed model was used to determine the analytical dependence of the joint leakage pressure on the assembly torque of bolts tightening. The correctness of the model was confirmed by the agreement of the analytical results with the experimental results.

Findings: Quantitative characteristics were determined for the selected type of joint.

Practical implications: The formulated conclusions indicate the possibility of using the model to support the design process and to select the structural parameters of the joint meeting the operational requirements and safety criteria.

Originality/value: Produced with recycled materials using.

Keywords: Pipe, Connection, Clamping rings, Ring stiffness, Polyester-glass composites

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PROPERTIES

1. Introduction

The subject of this paper is the analysis of pipe flange connection with a flat gasket and loose clamping rings. Such joints are characterized by the possibility of multiple disassembly, while providing tightness and static, dynamic and fatigue strength. Fulfilling the latter functions requires appropriate selection of design features, both dimensional and material [1-5]. The working load condition of the rings depends on the assembly load resulting from the use of bolted joints, the dimensions of the pipes and their flanges to be joined, the pressure in the system, and the characteristics of the seal used. Durability of connections is also important, especially in aggressive environment or required resistance to increased temperature [6-8]. Practically, the approval of a specific design solution requires carrying out the tests specified by the regulations in the conditions significantly exceeding the range of the indicated working parameters [1,2]. The tests described in this paper refer to special composite rings in order to check to what extent their innovative design features, especially material, allow to meet the operating requirements and technical tests. At the same time, the aim of the work is to build a model giving the basis for design analysis of the connections, as well as indicating the possibility of using computer-aided design methods [9,10].

2. Test object

The object of the analysis is the loading condition of a disconnected pipe flange joint, which uses flat, separated, loose composite rings, fixed circumferentially. The rings, connected by bolts, press against each other the thrust flanges integral with the pipe. The tightness of the joint is ensured by a flat annular gasket, seated between the thrust flanges [1].

Figure 1 shows a cross-section of a loose ring flange joint, which has a rectangular outline in this section. The combined thickness of the pipe flanges and connecting rings $2(h + h_s)$ can be seen.

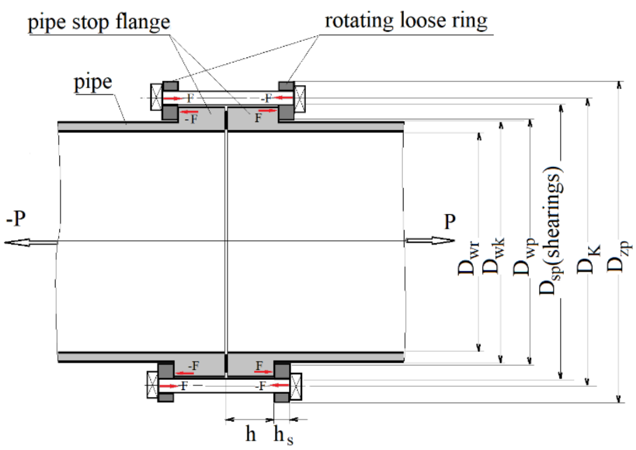

Fig. 1. Cross-section of a flanged joint

3. Preload condition analysis of joint components

The essential structural components of the joint are a pair of pipes ending in thrust flanges, pressure rings, an annular gasket between the pressure flanges, and assembly tie bolts. Figure 2 shows a diagram of the joint loading.

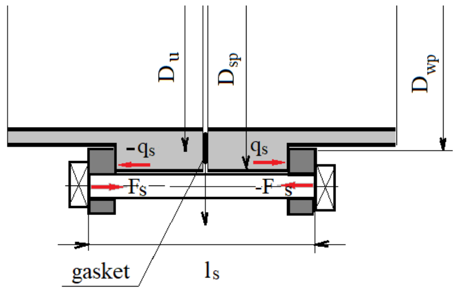

Fig. 2. Cross section of the connection with the marked loads of the rings F_s of the bolt pressure

The load on the fixed flange by the slack ring pressure is distributed over the annular surface of the contact with the thrust flange, under the tension force loading condition F_s. In the following analysis, this load, as well as the other analyzed reciprocal loads of the joint components - gasket, pipe flanges and rings with flanges, were reduced to uniformly distributed on the central circle of the gasket ring with diameter D_u. It was assumed that these loads, distributed uniformly, with a linear intensity q_s, which for n assembly bolts, in the absence of deformation, means the relationship:

$$q_s = \frac{nF_s}{\pi D_u}. \tag{1}$$

The load q_s depends on the value of the mounting torque of the bolts M_s. The assumption of uniform circumferential load qs is more reasonable the larger the number of mounting bolts. Figure 3 shows the flange connection diagram for 12 connecting bolts, together with the axial section of the bolts in the plane of symmetry.

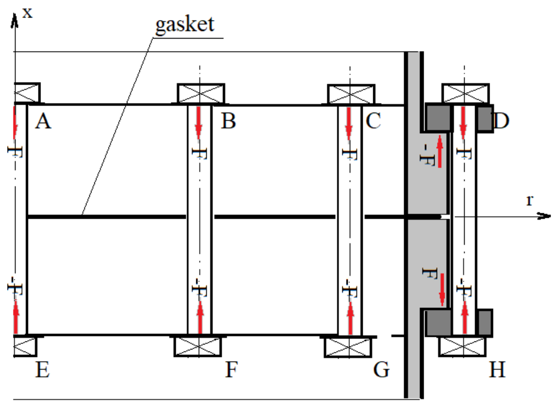

Fig. 3. Flange connection diagram for 12 connecting bolts

In the joint tests, the pipes are represented by short cylindrical pipe sections, closed at the free end, so that pressure probes can be conducted. Under unpressurized conditions in the closed chamber, the equivalent bolt tension intensity is equal to the mutual pressure intensity q_s of the rings, flanges and gasket. Figures 1, 2, 3 show the section and view of the gasket – bold line along the r-axis.

The pressure inside the pipe chamber causes loading of the gasket with linear intensity q_p reduced to a circle of diameter D_u.

$$q_p = \frac{D_w^2 \pi}{(4D_u)} = 0.048\pi. \quad (2)$$

This load is decomposed into a tensile component, q_1, to load the bolts, and a stress-relieving, pre-compressed gasket component, q_2.

4. Connection model under leak test conditions

In the description of the model, the load components corresponding to the bolt tension load q_s, pressure q_p and additionally the resultant intensity of the gasket compression qu will be used. The reason for the loss of tightness is to be found in the decrease in the intensity of the pressure on the pipe flanges to the gasket q_u, in the model case to 0.

The developed model of the joint and the loads occurring in it, refers to the phase of maintained joint tightness and is intended to investigate the conditions under which the loss of tightness occurs. A physical model was adopted as the basis, which was verified experimentally by subjecting to a leakage test a chamber consisting of two closed sections of flanged connected pipes, using a flat gasket and loose clamping rings.

The aim of the research was to develop and experimentally verify the joint model as a tool to support the design-construction process.

4.1. Load on fasteners

The load on bolts F_s tightened with the M_s torque depends on the geometrical features of the threaded connection and the coefficient of friction on the contact surface of thread and washers. In this study, this relationship was determined experimentally for bolts used in a dimensionally selected joint – 12 M20 bolts according to PN-EN 1092 standard [2,11].

The bolt was tightened in the hole of a rigid plate by measuring the elongation of the active section of the bolt for selected values of M_s nut tightening torque. After converting the elongation into tensile force F_s, a graph of Fs (M_s) relationship was prepared.

The values of the moments corresponding to the tension of the bolt under the conditions considering friction are summarized in Table 1.

The experimentally determined dependence has a non-linear character (Fig. 4), but in the range of the performed tests it is close to linear [12]. In the further analysis, the linear dependence of the bolt tension force F on the bolt tightening torque M_s was assumed, corresponding to the range of torque values used in the connections and adopted in the experimental studies for verification of the computational model described in Chapter 5 of the work. The linear plot of the relationship in the range of M_s [0, 100 Nm], determined by the least squares method, is shown in Figure 4, and the equation of the function is presented in relation (3).

$$F_s\ [\mathrm{N}] = 0.5M_s\ [\mathrm{kNm}], \quad (3)$$

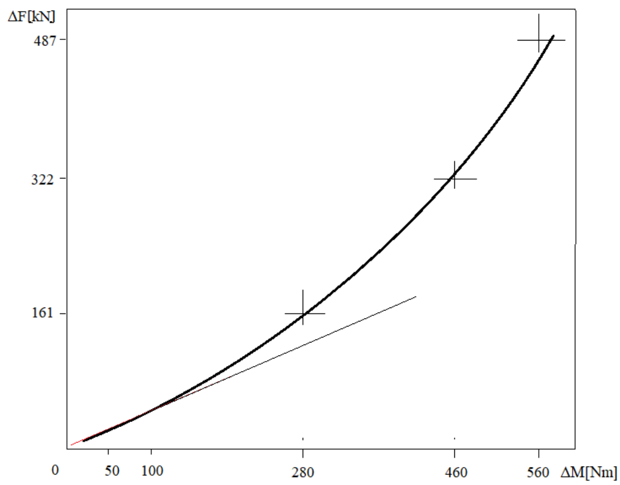

Fig. 4. Plot of bolt tension force versus tightening torque

The characteristic dimensions of the tested joint, are summarized in Table 2.

For a ring with the dimensions tabulated in Table 2, the gasket diameter is:

$$D_u = \frac{(D_{sp}+D_{wp})}{2} = 0.243\ [\mathrm{m}]. \quad (4)$$

Hence, the preassembly load intensities of the gasket and bolts are:

$$q_s = -q_u = \frac{(nF_s)}{\pi D_u},$$
$$q_s = -q_u = \frac{12 \cdot 0.5 M_s}{\pi D_u}, \quad (5)$$
$$q_s = -q_u = 7.86\ \left[\frac{\mathrm{kN}}{\mathrm{m}}\right].$$

Table 1.
The values of the moments corresponding to the tension of the bolt under the conditions considering friction

Screw outside diameter d_z = 20 mm	Tightening torque ΔM_s, Nm	Relative elongation of a tensile section $(l - l_o)/l \cdot 10^3$	Increase in tensile strength of the screw F_s, kN	F_s/M_s, 1/m
Cross-sectional area $A = 314$ mm^2	280	2.57	161	575
	460	5.13	322	700
Modulus of elasticity $E = 2 \cdot 10^5$ MPa	560	7.58	487	869

Table 2.
The characteristic dimensions of the tested joint

Pipe inside diameter for calculation of force P D_{wr}, mm	Working pressure p, MPa	Inner ring diameter D_{wp}, mm	Ring shear diameter D_{sp}, mm	Outer ring diameter D_{zp}, mm	Ring thickness hs/pipe flange h, mm, for n=8	Diameter under holes/hole diameter/ number of holes
200	$\leq$1.0	216	270	340	30/60	295/22/12

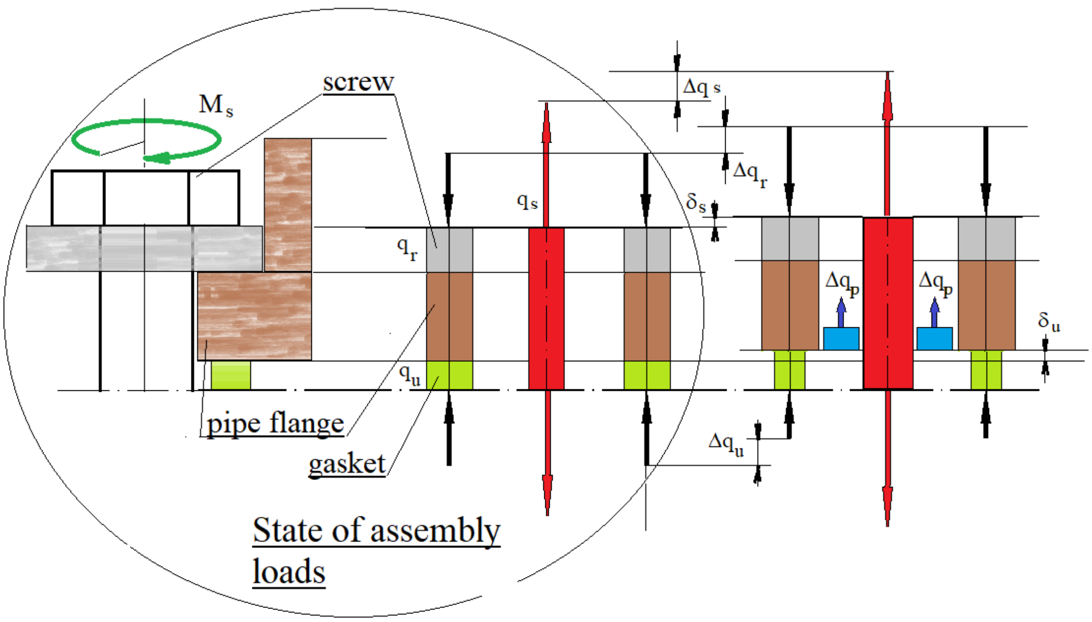

Fig. 5. Model of the elastic structure of the joint with the marked components of the loads from the pressure p and axial deformations of the elements

4.2. Loads on the remaining connection components

Figure 5 shows a diagram of the flange connection in axial section. The total bolt tension force F_s, the ring pressure on the pipe flange F_r and the gasket compression force F_u are represented by equivalent distributions q_s, q_r, q_u on the gasket centerline – a circle of diameter D_u. The part of the diagram encompassed by the oval corresponds to the preload of the elements resulting from the tightening torque of the bolts M_s. The values of the load intensities q_s, q_r, q_u result from the dependence of the tension of the assembly bolts on the tightening torque (6). A similar relation applies also to the changes of these loads caused by the change of the bolt tightening torque by o ΔM.

$$\Delta\theta_\sigma^M = -\Delta\theta_\upsilon^M = -\Delta\theta_\rho^M. \quad (6)$$

A change in the joint chamber pressure by Δq_p will cause changes in the assembly loads Δq_u^p, Δq_s^p, Δq_r^p of the gasket, bolts, and ring. In order to determine the effect of pressure changes on the load condition of the joint components, the model shown in Figure 5 was used. The displacements of the bolt-ring contact surfaces δ_s^p and the ring together with the flange and the gasket δ_u^p caused by the pressure change δ_u^p, are marked. They correspond to the change in bolt length and gasket thickness. Also the clamping ring can undergo deformation δ_r^p.

Due to the static indistinguishability of the connection, the load changes of the elements under the influence of the pressure change Δp will be determined from the relations (7), (8), related to the corresponding load change Δq_p. This load change will be decomposed into Δq_1 and Δq_2 components resulting from its distribution over the flange and gasket of the joint, as shown in Figure 6.

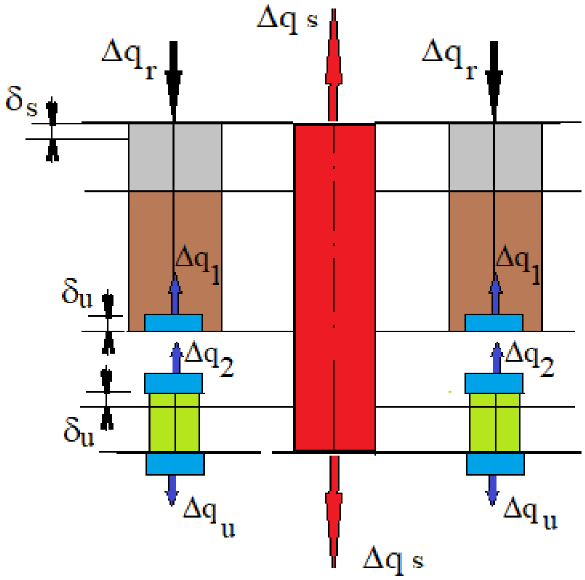

Fig. 6. Loads and deformations of joint components caused by change of pressure load by Δq_p

The determined values of load changes of the joint elements are:

$$\Delta q_u^p = \frac{\Delta q_p (c_s - c_r)}{(c_u + c_s - c_r)}, \tag{7}$$

$$\Delta q_s^p = -\Delta q_r^p = \frac{\Delta q_p (c_u)}{(c_u + c_s - c_r)}, \tag{8}$$

where:

Δq_1 and Δq_2 are the components of the load q_p in its distribution on the elastic ring and the seal;

$\Delta q_{s,u,r}^p$, N/m – linear load intensities on the screw, the gasket and the ring;

c_s, c_r, c_u, [Pa^{-1}] – axial compliance of the elements.

For n bolts the total susceptibility is:

$$c_s = \frac{D_u l_s}{(2E d_{sr}^2)} \ [\mathrm{Pa}^{-1}]. \tag{9}$$

For a gasket the susceptibility is:

$$c_u = \frac{g_u}{(E_u s_u)} \ [\mathrm{Pa}^{-1}], \tag{10}$$

where:

g_u – thickness,

s_u – width of the gasket,

l_s – length of the bolt,

E, E_u – elastic modulus of the screw and gasket materials, respectively.

From the analysis of these relations it can be concluded that an increase in the stiffness of the gasket means an increase in the proportion of its load in response to an increase in pressure. An increase in the susceptibility gives the opposite effect. The proportion of the load distribution on the gasket and the fastener including the ring will

$$w_{us} = \frac{\Delta q_u^p}{\Delta q_s^p} = \frac{(c_r - c_s)}{c_u}. \tag{11}$$

Knowledge of the susceptibility of the gasket c_u, the screw c_s and the ring c_r allows, on the basis of formulas (9), (10) to determine the effect of pressure change q^p on the gasket pressure.

The effect of simultaneous change of pressure p and tightening torque ΔM_s on the load condition is a superposition of the effects of these changes

The determination of the elasticity of the bolt system c_s can be done by assuming an elastic tensile model of the bar with an equivalent stiffness, which is the sum of the bolt system stiffnesses (Section 5.2).

The susceptibility c_u of the gasket, and indirectly the elastic modulus E_u of the material from which it is made, and the susceptibility c_r, of the ring will be determined experimentally. The knowledge of these elasticities will allow, together with the relation between the gasket pressure intensity and the bolt tightening torque (4), to determine the value of the bolt tightening torque M_s, necessary to ensure joint tightness under the load conditions of the assumed pressure p [13,14].

5. The results of experiments

The research program included designation of c_u susceptibility to selected types of gaskets, indirectly the elastic module E_u of the Plastic to be made and the susceptibility of the c_r ring.

5.1. Determination of the susceptibility of selected types of gaskets

In order to determine the susceptibility of gaskets, attempts to compress the gaskets made of various materials

[15]. The test was subjected to rubber shaped seals with a metal insert, rubber flat gaskets without inserts, rubber flat gaskets with a metal insert and flat fisheries from a bowling. The following Figures 7-10 show samples and cross-sections of the gaskets.

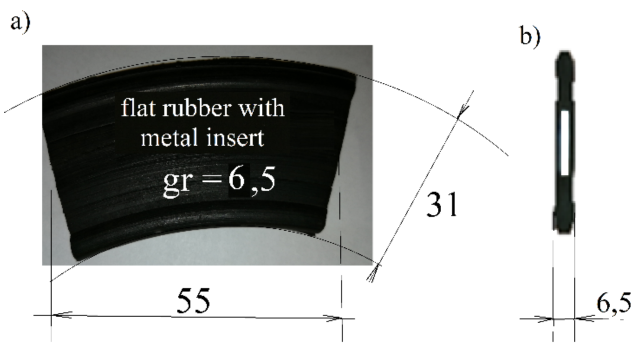

Fig. 7. Sample view of a shaped rubber gasket with a metal insert (a) and its cross-section (b)

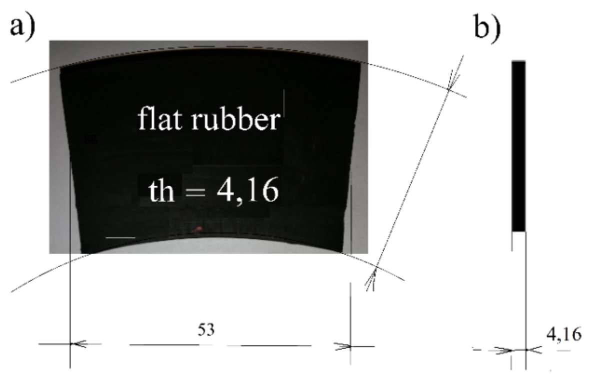

Fig. 8. View of flat rubber gasket sample (a) and its cross section (b)

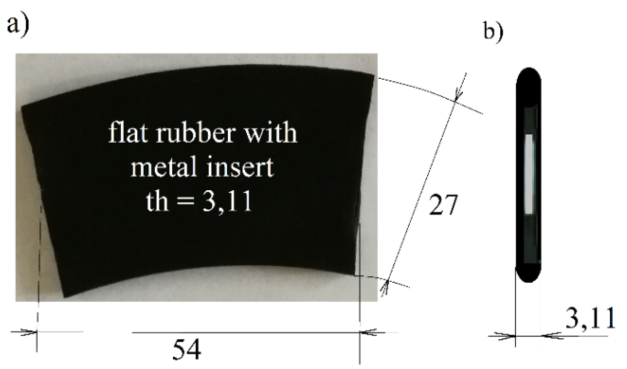

Fig. 9. Sample view of flat rubber gasket with metal insert (a) and its cross section (b)

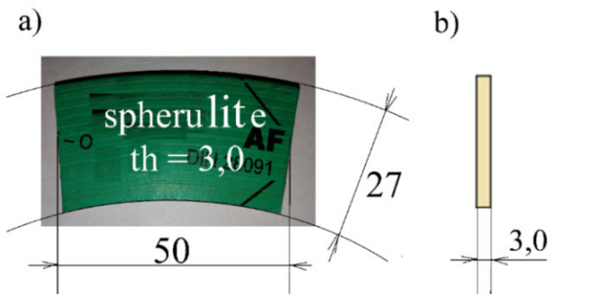

Fig. 10. View of a vertebralite gasket specimen (a) and its cross section (b)

The dimensions of the specimens are summarized in Table 3, next to the maximum compressive forces.

Table 3.
Compression test of rigid plastics acc. [17]

Nr	Gasket	Maximum strength F_{pr}, N	Sample thickness g_{pr}, mm	Width s_{pr}, mm
1	Rubber, shaped with metal insert	91	6.5	31.0
2	Rubber, flat	15026	4.16	34.0
3	Rubber, flat with metal insert	26411	3.11	27.0
4	Spherulite	23263	3.0	27.0

Figure 11 shows a summary plot of the trial runs.

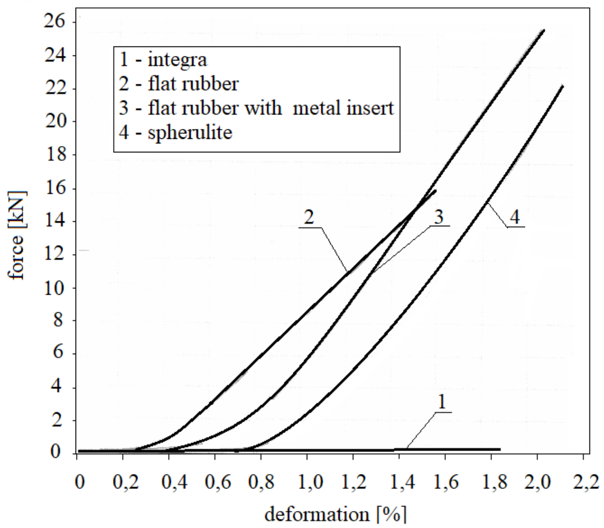

Fig. 11. A summary plot of the trial runs

Table 4.
The value of the susceptibility of the assembly of 12 bolts

No. of gaskets	Gasket type	Gasket thickness g_u, mm	Relative strain F_{pr} ε, %	Surface of specimen $A_{pr}\cdot 10^6$, m²	Force F_{pr}, N	Gasket width s_u, mm	E_u, MPa	Gasket compressibility $c_u\cdot 10^9$, m²/N
1	Rubber shaped with metal insert	6.50	16.5	1705	91	31	0.32	648.214
2	Flat rubber	4.16	11.5	1855	14.000	35	65.6	1.811
3	Flat rubber with metal inlay	3.11	10.5	1458	20.500	21	133.9	1.106
4	Spherulite	3.00	10.0	1350	17.000	27	125.9	0.882

The resulting susceptibility values of c_u gaskets, were determined from linear sections of compression plots.

The deformation of the gasket specimen:

$$\Delta g_{pr} = \frac{g_{pr}F_{pr}}{E_u A_{pr}}, \tag{12}$$

hence the Young's modulus of the gasket, against the specimen thickness equal to the gasket thickness:

$$E_u = \frac{g_{pr}F_{pr}}{D g_{pr} A_{pr}}. \tag{13}$$

The susceptibility of the gasket is expressed by the relation (10):

$$c_u = \frac{g_u}{(E_u s_u)}\ [\mathrm{Pa}^{-1}].$$

By substituting into this expression the values of the experimentally determined E_u:

$$E_u = \frac{(g_{pr}F_{pr})}{\Delta g_{pr} A_{pr}}\ [\mathrm{Pa}], \tag{14}$$

we obtain the relation specifying the value of the gasket susceptibility:

$$c_u = \frac{\Delta g_u A_{pr}}{(F_{pr} s_u)}\ [\mathrm{Pa}^{-1}]. \tag{15}$$

Calculated, on the basis of results of compression tests, values of the gaskets' susceptibility are listed in Table 4. The table contains also the dimensions of the gasket specimens and the maximum values of compressive forces, corresponding to linear ranges of compression characteristics.

5.2. Load on the seals assuming a rigid ring

To determine the load on gaskets under operating conditions, i.e. under force loading from pressure in a chamber closed with an annular connection, assuming a rigid ring, it is necessary to know the susceptibility of the screw and the gasket [16].

For a system of 12 steel bolts of diameter 20 mm (cross section 314 mm²), the stiffness is:

$$S = \frac{(3\pi d_s^2 E)}{l_s}, \tag{16}$$

while the compliance:

$$C = l_s(3\pi d_s^2 E)^{-1}, \tag{17}$$

the susceptibility reduced to the gasket alignment circle, with diameter D_u:

$$c_s = \frac{D_u l_s}{(3 d_s^2 E)}. \tag{18}$$

In the analysed connections (Fig. 1, Tab. 4), the value of the susceptibility of the assembly of 12 bolts will be:

$$c_s = 0.182 \cdot 10^{-9}\ [\mathrm{Pa}^{-1}]. \tag{19}$$

On the basis of relations (7) and (8) it is possible to determine the numbers of the influence of the change in the pressure in the specimen chamber on the change in bolt tension w_{sp} and gasket pressure w_{up}:

$$w_{sp} = \left(\frac{\Delta q_s}{\Delta q_p}\right) = \frac{-c_u}{(c_u + c_s - c_r)},$$
$$w_{up} = \left(\frac{\Delta q_u}{\Delta q_p}\right) = \frac{(c_s - c_r)}{(c_u + c_s - c_r)}. \tag{20}$$

For a rigid ring ($c_r = 0$), the relationships will take the form:

$$w_{sp} = \left(\frac{\Delta q_s}{\Delta q_p}\right) = \frac{-c_u}{(c_u + c_s)},$$
$$w_{up} = \frac{c_s}{(c_u + c_s)}, \tag{21}$$
$$w_{sp} + w_{up} = \frac{(c_s - c_u)}{(c_u + c_s)}.$$

Their values are given in Table 5.

Figure 12 shows the bar graph of load distribution from pressure, ring including bolts and gasket.

Table 5.
The numbers of the influence of the change in the pressure

Gasket No.	Gasket type	Susceptibility $c_u \cdot 10^9$, m²/N	Susceptibility $c_s \cdot 10^9$, m²/N	w_{up} $(\Delta q_u / \Delta q_p)$	w_{sp} $(\Delta q_s / \Delta q_p)$	(c_u/c_s) $(\Delta q_s / \Delta q_u) \times 10^3$
1	Moulded rubber with metal insert	648.214	1.8	0.003	0.997	0.000
2	Flat rubber	1.811		0.499	0.501	1.000
3	Rubber flat with metal insert	1.106		0.619	0.381	1.625
4	Spherulite	0.882		0.671	0.329	2.040

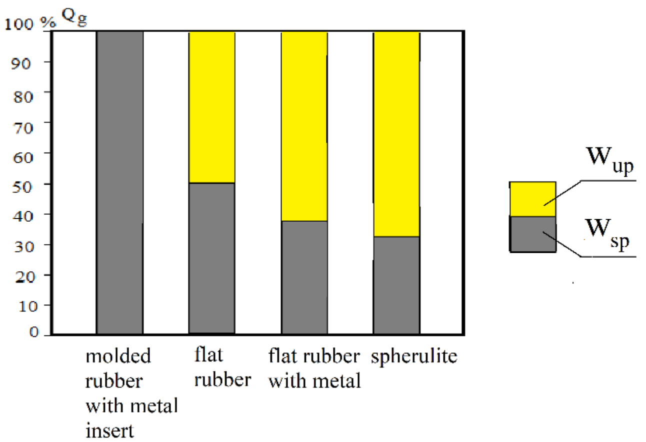

Fig. 12. Pressure load distribution of ring, including bolts q_s and gasket q_u

In the case of the initial load condition of the joint components, the load from the pressure change will result in load changes according to Table 5 and the diagram from Figure 12. For the bolt torque value M_s, the pressure p will increase the bolt load q_s to the value q_s^p:

$$q_s^p = q_s + q_p w_{sp}. \tag{22}$$

At the same time, the load on the gasket will decrease to the value:

$$q_u^p = q_u + q_p w_{up}. \tag{23}$$

The gasket is precompressed, so the component q_u has a negative sign. Its value, as well as the component q_s, is determined by the relation (5).

It should be emphasized that numerical values of values included in Table 5, as well as on graphs from Figure 12, concern particular constructional features of the joint – both dimensions and material of the pipe flange and the loose ring, as well as the type of gasket. A change in any of the parameters affects the values, q_s and q_p, and therefore the components of the pressure induced loads (22) and (23).

Changes in bolt and gasket loads correspond to their deformations, which for the satisfied assumption of significant ring stiffness are equal. The former result from an increase in tensile forces, the latter from a decrease in compressive forces.

In the state of the assembly load the gasket is in compression, which in the adopted convention of the signs (11) means that $q_u < 0$. From the relation (23) it follows that for a certain value of the compressive assembly pressure on the gasket q_u, there exists a pressure p which leads to loading the joint with the component q_p, for which the load on the gasket q_{up} will take the value of zero. This corresponds to the leakage loss condition of the joint. This condition is of the form:

$$\left(-q_u + q_p w_{up}\right) = 0. \tag{24}$$

5.3. Determination of c_r ring susceptibility

In order to compare and evaluate the influence of the ring compliance on the leakage loss conditions, experimental investigations were carried out to determine the compliance of the ring made of polyester-glass composite, manufactured by the company PLASTON. The dimensions of the tested ring are shown in Table 2. A schematic diagram of the test stand is shown in Figure 13.

Loading of the ring through circumferentially located pins, which corresponds to the bolt tension forces in the connection, causes the support reaction. In the adopted model, it is represented by a continuous distribution of q_s on the circular edge of the support. This condition of external loads causes axially symmetric lateral deflection of the ring [15]. This deflection is equal to the displacement of the rigid pressure plate of the testing machine. The dependence of the displacement on the clamping force allows us to determine the axial susceptibility of the ring under loading conditions at the ring joint.

$$c_r = \frac{\Delta \delta_p}{\Delta q_s}. \tag{25}$$

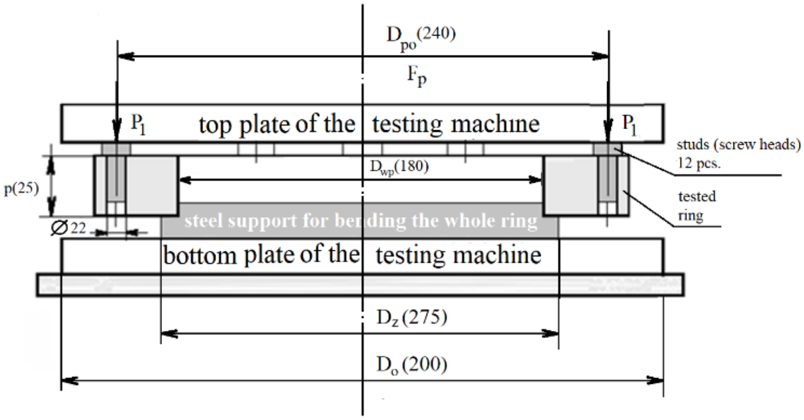

Fig. 13. Schematic of the test stand for ring stiffness, with the system of loads in the bend test

The susceptibilities of the ring, bolt and gasket, will be determined for a circumferentially distributed load, by the ratio of the axial displacement of the line of the circle of arrangement of the axes of the connecting bolts, to the intensity of the total bolt tension force distributed over this circle.

The ring is loaded by circumferentially distributed F_1 clamping forces on a circle of diameter δ_{po}, with a total value equal to the force F_p axially loading the top plate of the machine. Reduced to a uniform distribution over the plate support circle, the load representing bolt tension is:

$$q_s = \frac{F_p}{\pi D_{po}}. \tag{26}$$

The results of the two radial bending tests performed on the ring, included in Table 6, in the form of graphs are shown in Figure 14.

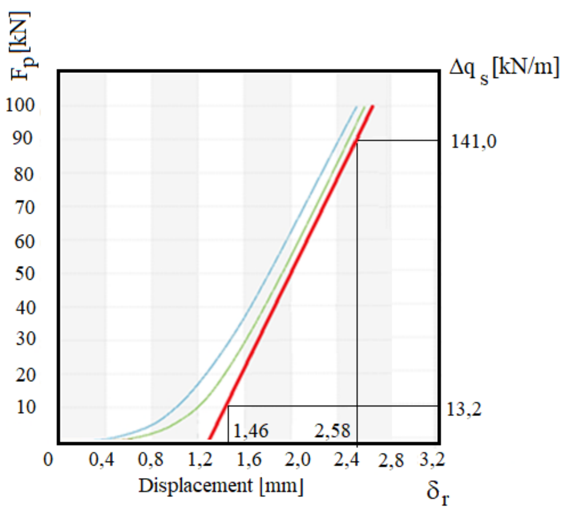

Fig. 14. Bending characteristics of the ring

Table 6.
Result of the bending test

Bending test as per scheme in Fig. 14; Flange with fine recyclate and two pieces of matte fabric 500 and two pieces of matte fabric 500; Estromal + Buff resin		
Size	Maximum force	Modulus of elasticity
Unit	kN	MPa
Average	99.7	10064

Based on the slope of the graphs in their linear part, the axial compliance of the ring c_r was determined. The ratio of the increment of the displacement $\Delta\delta_r$ to the load increment Δq_s caused by force ΔF_P, reduced to the circle of the holes arrangement, was used to determine the susceptibility of the ring c_r, under conditions of axially symmetrical radial bending.

$$\Delta q_s = \frac{\Delta F_p}{\pi D_{po}},$$

$$\Delta q_s = 1.33 F_p \left[\frac{\mathrm{N}}{\mu}\right]. \tag{27}$$

$$c_r = \frac{\Delta\delta_r}{\Delta q_s},$$

$$c_r = \frac{1.12}{127.8}\left[\frac{\mathrm{m}}{\mathrm{N}}\right]10^{-6}, \tag{28}$$

$$c_r = 9.84\left[\frac{\mathrm{m}^2}{\mathrm{N}}\right]10^{-9}.$$

The final verification of the connection model was carried out for the selected type of seal – a shaped rubber seal with a metal insert. The calculated values of the susceptibility of the connecting bolts c_s, the ring c_r and the seal c_u are summarized in Table 7.

The effect of changes in bolt tension Δq_s on changes in the intensity of gasket compressive forces is given by relation (11).

Table 7.
The calculated values of the susceptibility of the connecting bolts c_s, the ring c_r and the seal c_u

Element of the joint	Susceptibility c [Pa^{-1}]10^9	w_{up} $\frac{(c_r-c_s)}{(c_u+c_s-c_r)}$	w_{us} $\frac{(c_r-c_s)}{(c_u)}$
Bolt assembly	0.182		
Gasket c_u (molded rubber)	648.21	0.0151	-0.0149
Ring c_r	9.84		

For the determined susceptibilities, the value of the number of the effect of the change in bolt tension on the change in gasket load is:

$$\frac{\Delta q_u}{\Delta q_s} = -0.015. \tag{29}$$

It should be emphasized that the symbol D denotes the changes in the load on the gasket, the fasteners and the ring which result from a change in pressure. Determination of total loads of these elements requires taking into account the initial stresses-assembly. The negative value of the susceptibility (29) corresponds to a decrease in pressure on the gasket with an increase in pressure and tension on the fasteners.

The leakage coefficient, in the model taking into account the ring compliance has the form (24), where the ring compliance is determined by the relation (25). The number w_{up} of influence is determined by the formula:

$$w_{up} = \frac{(c_r-c_s)}{(c_u+c_s-c_r)}. \tag{30}$$

In the analysed case, the condition, based on the data from Tables 6 and 7, takes the form:

$$\left(q_u + 0.0151 q_p\right) = 0. \tag{31}$$

Based on relations (5) and (24):

$$\left(-7.86 M_s + 0.0151 q_p\right) = 0 \left[\frac{\mathrm{kN}}{\mathrm{m}}\right]. \tag{32}$$

Hence:

$$q_p = 0.52 M_s \left[\frac{\mathrm{MN}}{\mathrm{m}}\right], \tag{33}$$

and the pressure p at which the loss of tightness will occur under the conditions of installation torque of the bolts M_s, according to the relation (2):

$$p = \left(\frac{q_p}{0.048}\right) = 10.83 M_s \text{ [MPa]}. \tag{34}$$

This result, obtained on the basis of the assumed physical model of the ring connection, with the determined geometrical and material characteristics, answers the question about the value of the bolt tightening torque M_s necessary to ensure joint tightness in conditions of pressure load p. Experimental confirmation of correctness will at the same time allow its use as a tool to assist in the design of flange connections with design features different from those adopted for the analysis, e.g:

- pipe diameter
- dimensions of flanges
- stiffness of assembly rings used,
- number of connecting bolts and their dimensions.

It also allows solving inverse tasks, consisting in selection of constructional parameters and selection of gaskets and application of appropriate assembly screw tightening torques, ensuring joint tightness in test and operational conditions. In order to verify the correctness of the model, leakage tests were carried out on the annular joint with the design features assumed in the construction of the model.

6. Pressure test of ring joint

In order to verify the model, a pressure test was carried out, in which the values of pressure causing loss of joint tightness were determined, depending on the installation torque of bolts M_s. An annular connection of 12 bolts of pipe sections, with dimensions included in Table 2, was tested. A shaped rubber gasket with a metal insert was used, the characteristics of which are presented in Table 4.

The test was conducted for 7 bolt installation torque values. The initial bolt tightening torque value was 42 Nm, then the installation torque value was increased as described in Figure 15 and the data in Table 8. After loss of tightness, the chamber pressure was reduced to 0. The test was repeated for increasing torque values from 50 to 100 Nm. A graph of the pressure change in the joint chamber is shown in Figure 15. Note the slight stabilization of pressure after the initial short-term increase, which is characteristic of each M_s bolt torque value. This can be explained by the effect of transient self-sealing followed by a final permanent loss of tightness in a small time interval. Table 8 lists the pressure values at which the joint was unsealed.

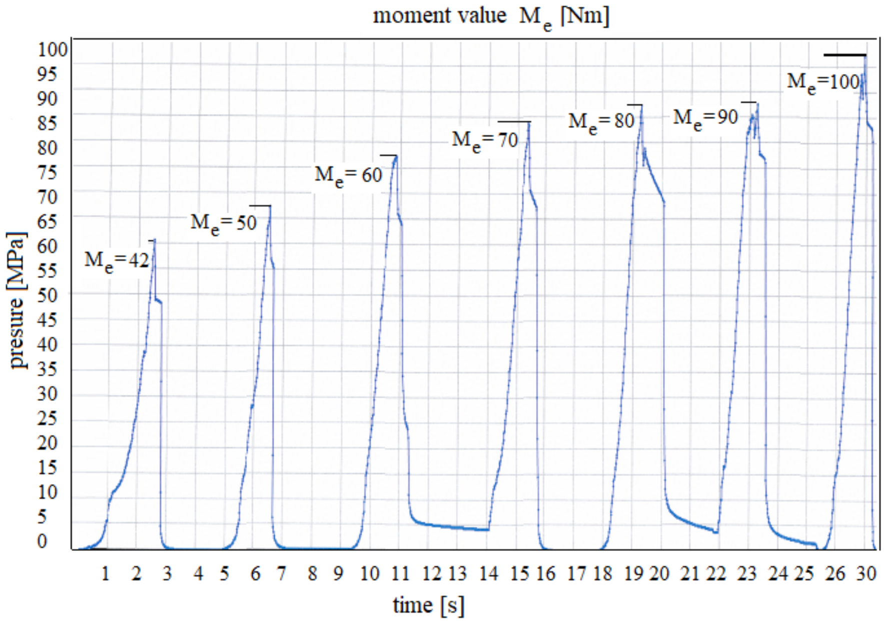

Fig. 15. Connection pressure test course

Table 8.
Pressure values for corresponding bolt torque values

No	Bolt tightening torque M_s^{exp}, Nm	Unsealing pressure p^{exp}, bar	Model unsealing pressure p, bar
1	42	49	45.50
2	50	56	54.15
3	60	65	64.98
4	70	68	75.81
5	80	70	86.64
6	90	78	97.47
7	100	83	108.30

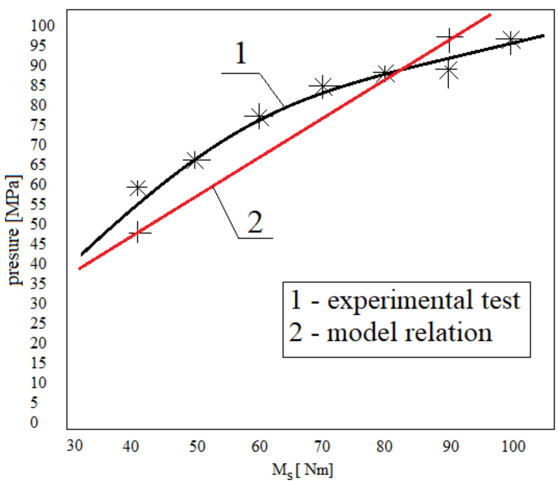

Fig. 16. Plots of leakage pressure on the assembly bolt tightening torque

In order to compare the obtained results, the values of the joint leakage loss pressures were determined on the basis of the results of the model analysis, obtained on the basis of the research-based elastic characteristics of the seal and the ring, written with the final relation (34) derived from the leakage loss condition (5). The results, in the form of pressure values for corresponding bolt torque values, are shown in Table 8.

For comparison, on the basis of the experimental and model test results, the dependence diagrams of the unsealing pressure on the bolt installation torque were prepared, shown in Figure 16. The curve (1) corresponds to the dependence resulting from the experimental tests, while the linear dependence (2) corresponds to the model relation defined by equation (34).

The discrepancy between the experimental and the model relation determined by the analysis, the measure of which is the ratio of the average values of the sum of the test

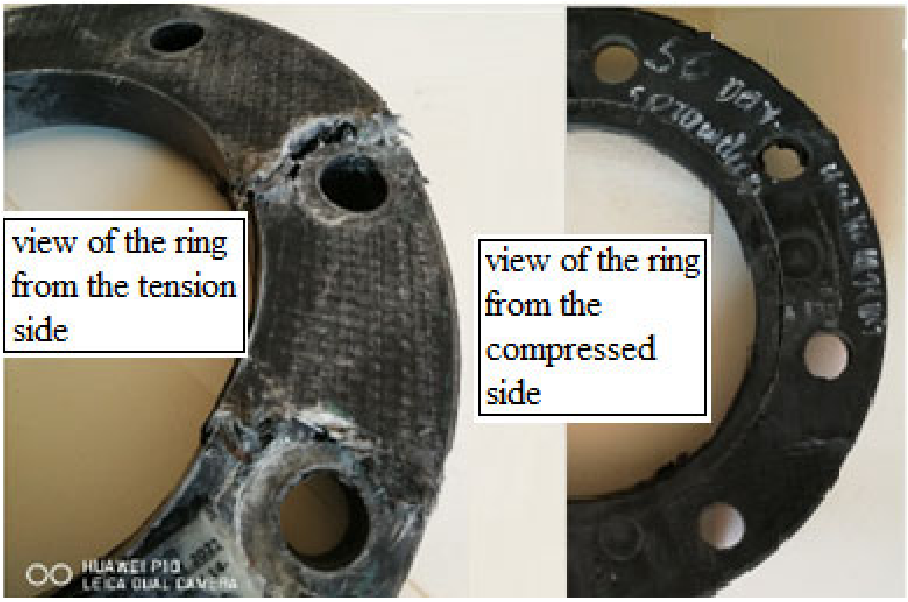

Fig. 17. View of the ring after the destructive test

results and the model calculations, gives the result of 14 to ensure the tightness of the joint under pressure test conditions.

The result of the model analysis can be considered safe from the point of view of the design operation assumptions The coefficient of random variation is 20%, which is satisfactory considering the small number of measurements.

The strength tests (Fig. 14) led to the destruction of the clamping ring. The ring destruction occurred when the value of the force F_p axially loading the upper plate of the machine was 227 kN. In the photographs from Figure 17 the picture of destruction is visible. In the weakened mounting holes, as expected, cracks occurred combined with delamination of the composite on the tensile stress side. The failure load significantly exceeded the required value, related to the need because the model results give underestimated values of the joint bursting pressures in the greater part of the test interval. This difference in 0.8 range of analysis are in favour of the experimental results, which should be considered as closer to the joint characteristics.

7. Conclusions

The conducted research allows to formulate the following conclusions:

1. The condition for the tightness of the annular joint is to ensure proper pressure of the pipe flanges on the gasket used.
2. The value of the clamping force depends on the gasket type, its cross-section and material characteristics 3.
3. The value of the clamping force is significantly influenced by the properties of the clamping rings, their rail and strength, as well as the number and dimensions of the applied clamping bolts.
4. The construction of the joint model, enabling the determination of the force depending on the assembly torque of bolts, ensuring joint tightness in the conditions of a given pressure, requires the knowledge of the elastic characteristics of the joint components – the gasket, clamping rings and bolts.
5. The study of the pressure dependence of the joint leakage loss from the bolt tightening torque, carried out on a joint with a shaped rubber gasket with a metal insert, has shown good agreement with the results obtained by calculation, using the physical model of the joint. In the analyzed case, the calculation results are underestimated in relation to the experimental tests, which is beneficial from the point of view of joint efficiency and safety.
6. The loading condition of the rings is favourably influenced by the increase in their stiffness, which can be achieved by increasing their thickness, reducing the bending moment resulting from the difference in the diameter of the outer edge of the rings and the diameter of the circle of arrangement of the tie bolts.
7. The determined leakage loss pressures confirm that the conditions for approving the tested joints for use in pressurized flow systems are met, i.e. it is possible to

ensure tightness of the joint in the scope of elastic deformation of rings and their strength.

8. The agreement of the pressure test results with those obtained by means of the model analysis confirms the possibility of using the developed model to support the design process of the connections under conditions of different dimensions of the connection elements and seals applied. In particular, it becomes possible to select the required dimensions of the clamping ring, the type of structure of its reinforcement, the materials used and the technology of its manufacture.

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