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Determination of the strength characteristics of a flanged joint with a flat gasket and loose retaining rings using a 3-point bending test

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ABSTRACT

Purpose: To evaluate the effectiveness of determining the strength of loose rings made, polymer laminate, using a 3-point bending test of rectangular material specimens differing in form and dimensions from actual structural components.

Design/methodology/approach: The analysis carried out indicated a method of quantitative evaluation that could be directly related to the operating conditions of the rings. A similarity relation between the critical states of the ring and the specimens subjected to bending tests was presented. The assumed hypothesis of maximum tensile stresses of the rigid outer laminate layers provided the basis for formulating a criterion for the allowable loads on the rings, depending on the assembly tension of the bolts and the operating pressure in the installation.

Findings: The results of the bending tests, both in the elastic and failure range of the samples, confirmed the validity of the hypothesis about the proportionality of the breaking stresses in the ring and the bending, cuboid sample, made of a similar laminate.

Practical implications: The area of dimensional optimisation of the joint with regard to its strength and stiffness criterion is indicated.

Originality/value: The method provides a basis for evaluating the strength and tightness of a flange connection by means of experimental testing of specimens made of a laminate similar to the material of the rings. The results of the 3-point bend tests allow for the substitution of difficult tests of complex assembly components such as rings in a flanged connection, including the operating conditions of the connection in a pressure piping system.

Keywords: Pipe, Connection, Clamping rings, Ring stiffness, Composites, Bend test

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PROPERTIES

1. Introduction

One of the common disconnecting pipe connections is flanged connections [1-6]. The connecting elements are bolts, the number and dimensions of which are selected depending on the pressure in the system and the leakage conditions. Connections in which additional loose pressure rings are used to enable the required compression force of the gasket to be achieved in order to improve the strength of the pipe flange zone are used [1, 7-9]. It finds particular justification in the case of laminated polymer composite pipes. Experiments have confirmed that the loose rings can also be made of composite in such design solutions. The described tests were carried out on rings made with an innovative material solution in which composite waste from the pipe production line accounts for a significant proportion of the material used. Figure 1 shows an example of a joint (a) and loose rings (b).

a) b)

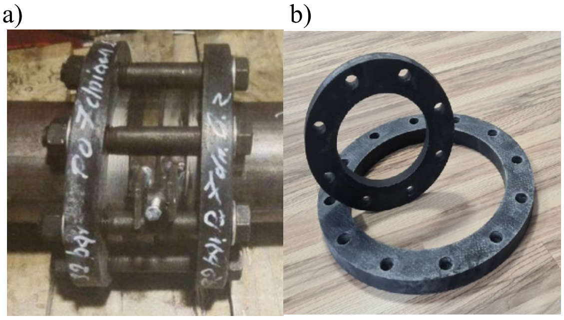

Fig. 1. View of a flanged connection with loose rings (a) and rings (b)

Composite pipe technology has a long-standing tradition. The use of rings, especially with innovative material solutions, required experimental research in addition to the development of innovative manufacturing technology. The next step, aimed at optimising ring dimensions with regard to strength and stiffness criteria, is a strength analysis [10,11]. In order to carry this out, a physical model of the ring is developed under working load conditions [3,4]. The strength analysis aims to determine the influence of bolt tension forces on the loading condition of the ring of the flange connection. In particular, to determine the zones of the ring where the loads are highest. Based on general relationships, it will be possible to determine the value of bolt tension that threatens to destroy the ring. The tension in the bolts is the result of a superposition of the component from the installation tension and due to the pressure in the installation, in which the flange connection with loose rings is used. The value of the installation tension should correspond to the leak-tightness condition of the joint under the operating pressure in the installation. The condition depends on the type of seal used in the connection. The next steps of the analysis are:

- construction of a physical model of the flanged joint with a loose assembly ring;
- a model of the external loads on the ring in the joint;
- analysis of the stress state in the ring with an indication of the critical load zone.

Verification of the analysis results will be carried out on the basis of strength tests of the ring on a test bench, on which sample rings with design features consistent with those adopted for the analysis have been tested. The verified ring model will be used to design joints of different diameters for different pressures and sealing arrangements. Finally, leakage tests were carried out on joints with loose composite rings.

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2. Physical model of the ring joint

Figure 2 shows a half-section of the model symmetrical part of the joint, comprising the end section of the pipe terminated by the flange and the applied loose compression ring.

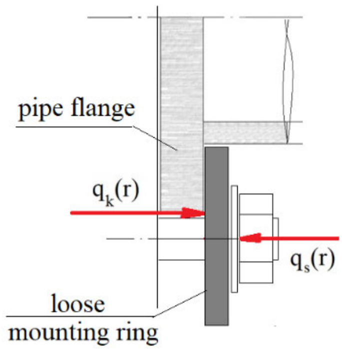

Fig. 2. Flanged connection with loose rings

The load $q_k(r)$ shown in the figure corresponds to the distributed, reciprocal pressure of the ring and the pipe flange of radius R_k. The linear distribution of this load on the edge of the pipe flange corresponds to the assumed form of deformation of the ring corresponding to axisymmetric bending. The load $q_s(r)$ is the bolt tension intensity, assumed in the model to be the axisymmetric distribution of the 8 bolt

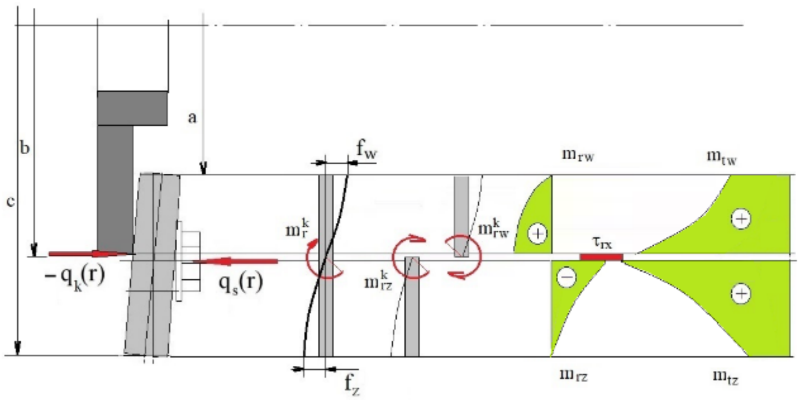

Fig. 3. Cross-section of a joint with external loads and components of the internal load condition with circumferentially distributed bending moment with radial m_t and circumferential m_r components

tension Q, on a circle of radius Rs, close to the radius of the axis placement circle of the mounting holes.

$$q_s = 8Q/2\pi R_s. \quad (1)$$

Figure 3 shows a half-section of the joint with circumferentially distributed external axisymmetric ring loads q_k and qs, derived from the flange pressure and bolt tension forces Rs, respectively. The diagrams of the radial components of the ring bending moments, radial m_r, circumferential m_t, and the radial component of the shear stress τ_{rx} correspond to the loads on the inner and outer parts of the ring, separated by the line of pressure against the edge of the pipe flange. The characteristic dimensions of the joint are marked, assuming, for the sake of simplicity, the notation of the formulae defining the dependence of the components of the state of internal loads in the ring: R_z=c, R_k=b, R_w=a. The outer radius of the pipe flange R_k belongs to the nominal dimensions of the pipe. In contrast, the radius Rs is the radius of the circle on which the resultant pressure from the bolt tension forces Q, via the washers and from the pressure in the pipeline is distributed. The difference in length of these radii is small because the bolt tensions are transferred to the flange through the bolt heads and washers, which, under the radial deformability of the ring, results in a significant approximation of these radii.

For similar values of the radii R_k and R_s, it occurs:

$$q_k \approx q_s. \quad (2)$$

The load on the ring is a reduced, distributed bending moment of linear intensity on a circle of radius $b=(R_k+R_s)/2$:

$$m_r = q_s (R_s – R_k). \quad (3)$$

The moment vector is tangent to the line of the circle, hence its designation as circumferential. The load is axisymmetric. The form of the deflection line of the radial axis of the ring section is shown in Figure 3.

In the following analysis, the ring will be divided into two rings: an outer and an inner ring, with inner and outer radii (b, c) and (a, b), respectively. On a circle of radius b, these rings are connected rigidly, which means that the radial deflection angles are congruent φ:

$$\varphi_w(b) = \varphi_z(b) = \varphi(b). \quad (4)$$

and the axial displacement, assumed to be equal to 0. The load mr causes the inner and outer parts of the ring to bend with a moment of intensity depending on the radius r, $m_{rw}(r)$ and $m_{rz}(r)$, respectively. For radius r=b the following occurs:

$$m_r = m_{rw}(b) + m_{rz}(b). \quad (5)$$

3. Analysis of the load condition in the ring with an indication of the critical load zone

In order to analyse the internal load condition of the ring, an axisymmetric ring model with distributed moment load near the edge of the flange, on a circle of radius R_k = b, will be used (Fig. 3).

The rotation angles of the contacting, connected ring sections, depending on the radius, according to [4], will be:

- for the outer part of the ring:

$$\varphi_z = (m_{rz}b^2r_z)[(1 - \nu)/(1 + \nu) + c^2/r_z^2]/ /\{(1 - \nu)D(c^2 - b^2)\}, \quad (6)$$

- for the internal part:

$$\varphi_w = (m_{rw}a^2r_w) [(1 - \nu)/(1 + \nu) + b^2/r_w^2] / / [(1 - \nu)D(b^2 - a^2)], \quad (7)$$

where: $D = E(1 - \nu^2)$,
the a, b, c values for the test ring in Table 1.

Table 1.
Ring dimensions

Inner ring diameter D_w, mm a	Diameter of flange (ring shear circle) D_k, mm b	Outer ring diameter D_z, mm c	Ring/flange thickness for n=12 g, mm	Diameter under holes D_s, mm/hole diameter, mm/number of holes	Dimensions of the washer d_z/d_w, mm/mm
216	270	340	30/60	295/22/12	21/37

It follows from the condition of compatibility of the rotation of the two parts of the ring around the edge of the quill:

$$(m_{rz}b^2r_z)[(1 - \nu)/(1 + \nu) + c^2/r_z^2] /[(1 - \nu)D(c^2 - b^2)] =$$
$$= (m_{rw}a^2r_w) [(1 - \nu)/(1 + \nu) + b^2/r_w^2] / [(1 - \nu)D(b^2 - a^2)]$$

and from there:

$$m_{rz}= m_{rw}a^2/2(b^2 + c^2) \tag{8}$$

At the same time, the radial components of stress are accompanied by circumferential components. The formulae (9), resulting from the equilibrium condition (5) and inseparability (8), determine the dependence of the moment components for the outer and inner parts of the ring (9). The calculated values refer to the dimensions of the example ring contained in Table 1.

$$m_{rz} = m_r\, 2a^2 /(b^2 + c^2 + 2a^2) = 0.33\, m_r,$$
$$m_{tz} = (m_{rz}b^2)(1 + c^2/ b^2)/(c^2 - b^2) = 1.27\, m_r,$$
$$m_{rw} = m_r\, (b^2 + c^2)/(b^2 + c^2 + 2a^2) = 0{,}67\, m_r, \tag{9}$$
$$m_{tw} = (m_{rw}b^2)(1+ a^2/ b^2)/(b^2 - a^2) = 3.05\, m_r.$$

4. State of stress in the ring

The characteristic waveforms of the bending moment components for the inner and outer parts of the ring are shown in Figure 3. They all assume the highest values in the ring contact zone with the pipe flange edge. The structure of the ring is of the sandwich type – the outer anisotropically laminated facings have a significantly higher stiffness than the core, which contains only chopped recycle fibres. The bending of the ring, therefore, induces a predominantly planar stress state in the cladding. The main directions are radial and circumferential. The radial component σ_{rw} induces stretches across the width of the ring. The circumferential component σ_{tw} induces tension on the inner part and σ_{tz} compression on the outer part of the ring. The distribution of stresses over the thickness of the ring is illustrated in Figure 4.

Equations (9) determine the maximum values of the respective components of the bending moments. These occur near the edge of the flange.

For a sandwich type slab of thickness d, in which rigid facings of thickness g carry the main loads, their loading condition is flat and the bending moment, with components m_t and m_r, induces stresses in the facings with components:

$$\sigma_t = \pm\, m_t/(dg)\ [Pa],$$
$$\sigma_r = \pm\, m_r/(dg)\ [Pa]. \tag{10}$$

The opposite signs of the corresponding stress state components in the opposite layers of the ring indicate the compression and tension directions.

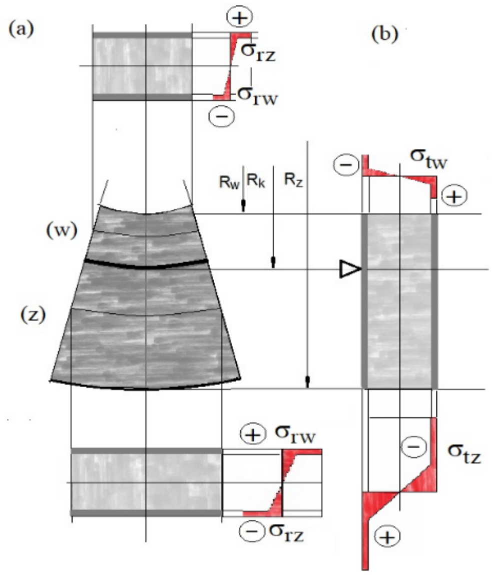

Fig. 4. Stress distribution in the radial (a) and circumferential (b) sections of the inner (w) and outer (z) parts of the ring

It should be noted that in the outer section, they do not take into account the stress concentrations in the vicinity of the mounting bolt holes. Careful analyses show that the stresses in the zone of the concentrations can be three times higher than those calculated outside this zone [2,3,12]. As a result, their maximum values will occur in the outer, hole-weakened part of the ring and may reach, for k = 3:

$(\sigma_t)_{max} = k \cdot (\sigma_t)_{max}/dg \quad = 1.0\ m_r,$

$(k \cdot \sigma_r)_{max} = \ 3.8\ m_r\ /dg \quad .$ (11)

The diagrams of bending moments in Figure 3 show the proportions of the individual components, taking into account the occurrence of stiffness near the holes. Figure 4 shows the distribution of stresses over the thickness of the ring. Under such loading conditions, the critical mechanism for potential ring failure may be brittle fracture along a line passing through the centres of the holes on the tensile side [12,13] of the laminated ring cladding. Figure 5 shows the load distribution of the ring cladding around the hole. Point K indicates the zone of highest radial and axial loads It is the area of probable crack initiation of the cladding. It is to be expected that the cladding subjected to the highest tensile stress will fracture first.

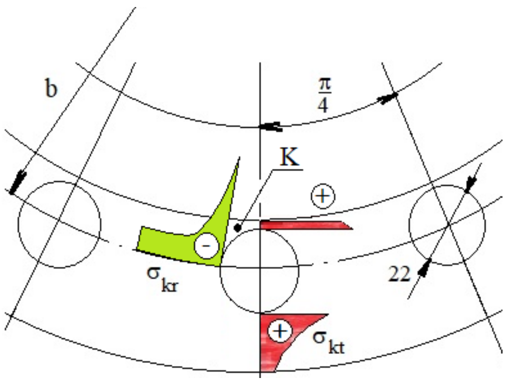

Fig. 5. Stress diagrams around the hole

5. 3-point bending test of a ring laminate specimen

A measure of the strength of the rings is the maximum value of the mounting bolt tension forces under mounting load and pipeline pressure conditions. In practice, especially in the design phase, there is a need to determine the strength of the designed ring based on the results of laminate tests [1]. For this purpose, bending strength tests are carried out in a 3-point bending test. Specimens with a ply structure and cladding, as in the target rings, are tested. However, the thickness of the core, which retains a structure similar to that used in the rings, is reduced. The assumption of equivalence of the results of the destructive bending test of cuboidal specimens with the strength of the ring is based on the hypothesis of maximum tensile stresses [7] related to the stress state in the cladding of the composite. As in the ring model, with the dominant stiffness of the cladding, the equivalence condition for the failure states of the ring and specimen is of the form:

$(\sigma_r)_{max} = (\sigma_p)_{max},$ (12)

where: $(\sigma_r)_{max}$, $(\sigma_p)_{max}$ are, respectively, the maximum tensile stresses in the outer cladding of the ring and the specimen under 3-point bending conditions.

The tensile component $(\sigma_r)_{max}$ - equation (11, 12) takes the highest value for the ring. After taking into account the relationship (1), (3), for g=0.03 m, d= 0. 54 m:

$(\sigma_r)_{max} = 3{,}8\ m_r/(dg) = 3{,}8 q_s (R_s - R_k)\ /(dg),$

$q_s = 4Q/\pi R_s,$

and for R_s [m], Q [N]:

$(\sigma_r)_{max} = 0.17(R_s - R_k)Q$ [MPa]. (13)

For the specimen, the maximum stress was determined in a 3-point bending test. A schematic of the loading system is shown in Figure 6. For the sample dimensions: l = 56 mm, b = 15 mm, h = 7 mm value $(\sigma_p)_{max}$, taking the value of the force P in [N], will be:

$(\sigma_p)_{max} = 0.3P$ [MPa]. (14)

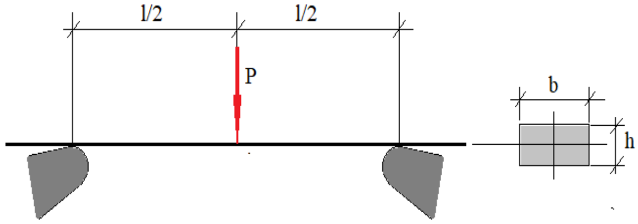

Fig. 6. Schematic of the 3-point bending test system

Figure 7 shows diagrams of the eight bending tests carried out on the specimens.

An example of failure form of a specimen subjected to 3-point bending shown in the Figure 8.

Table 2 shows the averaged results of the bending tests.

Table 2.
Bending test results

Modulus of elasticity E, MPa	Bending strength, MPa	Destructive deformation, %
21328	239.5	1.42

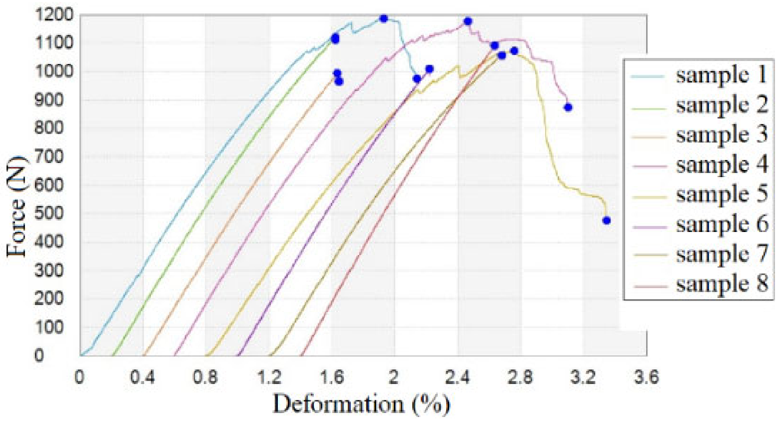

Fig. 7. Diagrams of the bending tests of 8 specimens

Fig. 8. Example of failure form of a specimen subjected to 3-point bending

The maximum tensile stresses in the central cross-section of the lower cladding of the laminate specimen will be:

$$(\sigma_p)_{max} = 3Pl/2bh^2. \quad (15)$$

Assuming that the determined bending strength corresponds to the maximum tensile stress $(\sigma_p)_{max}$ on the surface of the specimen, referring to hypothesis (12), from which the equality of the maximum tensile stresses reached in the specimens and in the ring is derived, from relation (13) the maximum value of the force Q of the working:

$$Q = 5.9(\sigma_r)_{max} / (R_s – R_k) \text{ [MPa]}. \quad (16)$$

In practice, this relationship can be used to determine the bolt torque based on the following data: thread geometry, surface condition and working pressure of the pipeline. The numerical value of W=5.9 is derived from the flange and ring diameters and must be calculated as described. The difference in radii (R_s - R_k), resulting from the design form of the ring, allows a reduction in the bolt tension force Q. Instead, it increases the bending moment on the ring (3) and, thus, the mounting stresses in the ring (11). The final choice of design solution for the ring should be based on the optimisation result corresponding to the minimum of the maximum ring loads.

6. Conclusions

The innovative rings, manufactured using waste glass fibre-reinforced polyester composite material, show very good strength when applied to pipe flange connections. There is a need and opportunity to optimise their form, in particular the selection of characteristic dimensions, number and types of bolts, depending on the dimensions of the pipe flanges and the operating pressure of the installation. A key characteristic of the rings is the bending strength, which is crucial to the possibility and range of applications of loose ring flange connections in a complex ring arrangement. It is shown that the determination of this strength, which can be measured by the maximum pressure on the flange and gasket, is possible from the results of a 3-point bending test on samples of the laminate from which the ring is made. Due to the limitations of the standards specifying the form and dimensions of the specimens, it is furthermore not possible to carry out tests on the ring material of the thickness used. This paper shows that this difficulty can be overcome by testing specimens with a smaller thickness of the filling laminate core.

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