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Some research results on the tightness and strength of flange joints

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Abstract. The paper presents recently published literature research on the tightness and strength of flange joints. These could influence both the calculation and the maintenance of flange joints. Some of the research results have been compared to normative stipulations currently used in the calculation of flange joints. The presentation covers: – the influence of sealing gasket design and screw load on sealing; – the influence of creep on screw relaxation, and thereby on gasket sealing; – cracking in flange joints; – the influence of an additional axial force of flange joints on the rotation and on the deformation of flange joint components; – flanges without sealing gasket; – the influence of the method of flange joint bolt tightening on the uniformity of bolts loading and tightness.

Keywords: flanges, bolt strength, tightness, sealing gaskets, creep.

1. Introduction

• The feature that makes classical flanges stand out is the flange welded to the shell or pipe; an exception is represented by flanges bolted onto a connection or pipe and free flanges [1].
Welded flanges have the following disadvantages: – residual stresses in the thermal influence zone of the weld; – distortion of pipe/shell against the flange caused by the weld. A slight distortion in the welding process, after assembling the flanges with screws may result in damage to the sealing gasket, uneven grip in the circumferential direction, leading to premature untightening and fluid leakage.

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• A new type of pipe/flange joint (Fig. 1) consists in the plastic deformation of the pipe inside the grooved flange [2] - [4]. The pipe is inserted inside the grooved flange, then undergoes cold plastic deformation by means of a hydraulic device.

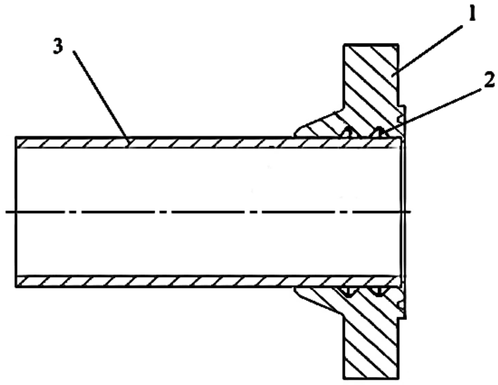

Fig. 1. Weld neck flange, *1*, with grooves, *2*, joined to a pipe, *3*, by cold plastic deformation of pipe [5].

This happens exactly as in the case of the joint between the pipe and the tubular plate of heat exchangers [1]. This type of flange was patented by Quickflange Technology AS 3. In this technology, the pipe is inserted inside the grooved flange; one introduces a segmented tool inside the pipe, axially maintained by a threaded fastener, A hydraulically actuated piston pushes a cone into the expandable tool. By doing so, the tool segments move in the circumferential direction and radially expand. The radial expansion induces the pressure that generates the plastic deformation of the end and its "anchoring" into the flange grooves.
The deformation of the pipe end with a hydraulic mandrel inserted therein, as one does when fixing pipes onto the tubular plate, could be, however, more efficient.
In the case of these flanges one achieves both the anchoring of the pipe onto the flange grooves and a very good metal-to-metal interface sealing. The heating of the welding neck flanges and consequently their likely distortion are eliminated.
On the other hand, local plastic deformations can be generated in the plastic deformation process, which reduces the strength of the pipe-flange joint.

2. Research on flange joint sealing

• At present, the tightness of a flanged joint is achieved by using a sealing gasket that is inserted between the two adjacent flanges (Fig. 2). Depending on the operating parameters (pressure, temperature) and the nature of the sealed fluid, various design solutions of sealing surfaces are being used (Fig. 3).

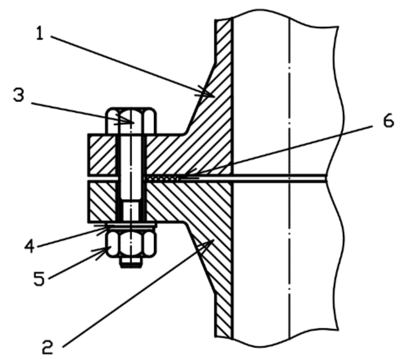

Fig. 2. Flange joint (*1; 2*) provided with a sealing gasket (6) and with clamping bolts (*3*); washer (*4*); nut (*5*) [1].

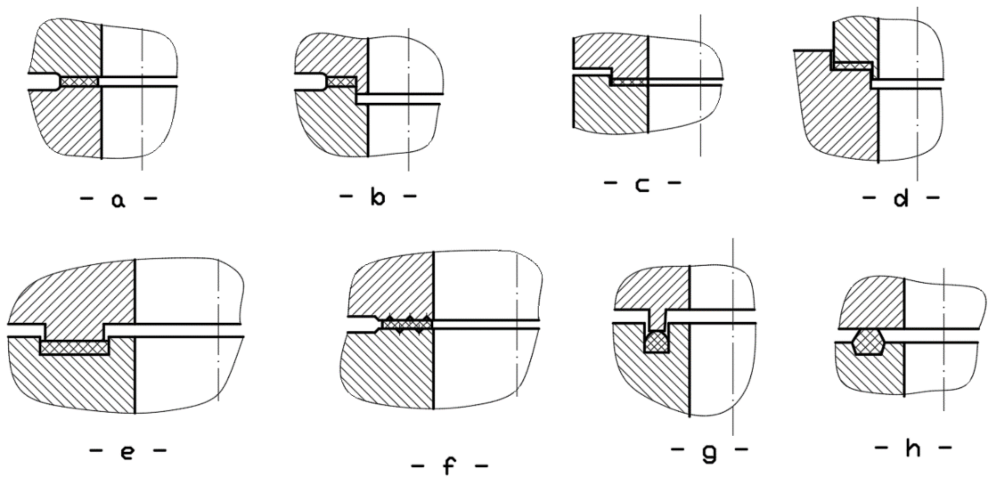

Fig. 3. Design solutions for sealing surfaces*:* *a* - with a simple plane surface and rectangular section gasket; *b* - with plane shoulder sealing surface, with rectangular section gasket; *c, d* - with plane sealing surface with threshold and hollow, with rectangular section gasket; *e* - with channel sealing surface with wedge, with rectangular section seal; *f* - plane surface with annular grooves; *g* - with channel sealing surface and wedge, with circular section seal; *h* - with an annular groove and a plane surface with a profiled section seal [1].

The quality of the flange surfaces that accommodate the sealing gasket, evaluated by their roughness, influences the tightness of the flange joints. On the other hand, with the increase of the seal width, the sealing improves.

Roughness marks result in an imperfect contact between the gasket and the flange; the surfaces are only partially in contact, i.e. in the top roughness areas. The fluid may "slip" between the roughness marks that are not in contact with the other surface. The contact between the flange and the sealing gasket is imperfect so that the fluid may flow from inside the container between the flange and the sealing gasket. Roughness marks may be divided into three categories: - roughness that does not come into contact with the other surface; - roughness that comes into contact with the other surface and is elastically deformed; - roughness that comes into contact with the other surface and are plastically deformed when tightening the sealing gasket. The best flange roughness is the one that ensures the lowest leakage in the sealed fluid.

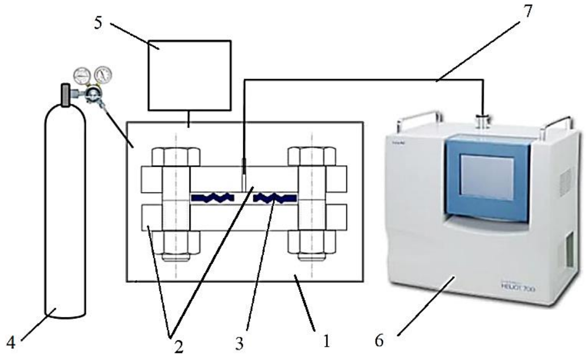

Fig. 4. Experimental installation for the measurement of helium leakage flow rate:
1 - test chamber; 2 - flange joint; 3 - sealing gasket; 4 - helium bottle;
5 - helium density sensor; 6 - helium leak detector; 7 - pipe [6].

Investigations of helium leakage between flanges and a corrugated, elastically deformed gasket and a plastically deformed metallic one (Fig. 4), in the case of three different roughness of a flange (1.5, 2.5 and 3.5 μm) and axial forces in bolts featuring 40; 60; 80; 100 and 120 kN have yielded the following [6]:

– for an axial force in bolts relatively small, the modification of the roughness causes relatively large variations in the helium leakage rate. At relatively high axial forces there was found no significant influence of flange roughness on leakage flow;

– with a plastically deformed gasket, in the case of a 3.5 μm flange roughness, one has found a helium leakage flow rate in all tested axial forces (Fig. 5). Consequently, this roughness is not suitable for such a seal;

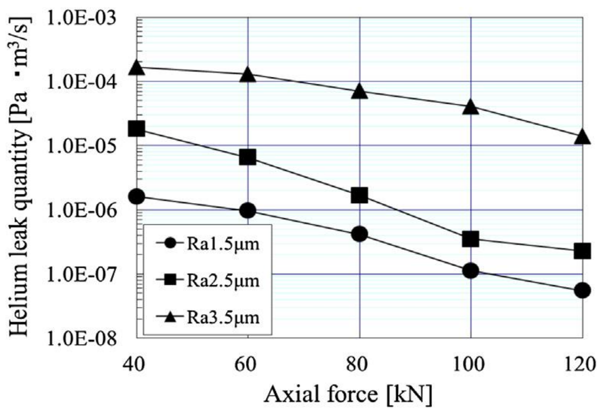

Fig. 5. Measured helium leaks in a flange joint with an elastically deformed gasket [6].

– plastically deformed corrugated metal gaskets are more efficient than the elastically deformed ones.

Such an analysis of flange joint performance, whose objective is the reduction of sealed fluid losses, was also performed for graphite gaskets corrugated in the radial direction. There was tested the possibility of reducing fugitive emissions of volatile organic compounds (VOC), for which an average flow rate of about $10^{-3}\,\mathrm{mg \cdot m^{-1} \cdot s^{-1}}$ [7]; [8] is allowed. The VOC emissions are given off in two ways: – along the contact surface between the flange and the gasket; – by the sealing gasket material itself.

The fugitive emissions flow rate increases approximately linearly with increased sealing fluid pressure. It has been found that for graphite gaskets [8] the fugitive emission rate (VOC) can be reduced by:

– making fine graphite pores; – impregnating graphite with substances to be maintained during their operation; - the use of flexible gaskets whose capillary diameters decrease when the gasket is compressed; – reduction of seal thickness.

• Flange joints with two concentric (double) gaskets (Fig. 6) were investigated to determine the influence of gasket pressure and sealing performance of the gasket on the joint [9]. Double gaskets of different widths and with different distances, b_g, from each other were analyzed. Double gaskets – flanged joints have been found to withstand higher internal pressures than those with a single gasket.•

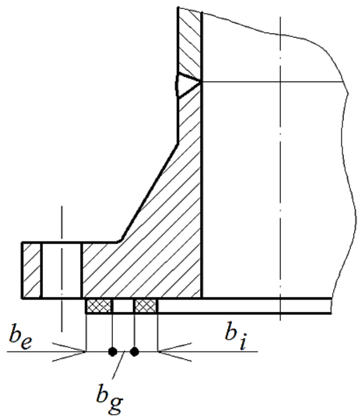

Fig. 6. Flange joints with two concentric seals spatially separated by width b_g.
The sealing gaskets have the following widths b_i - inner lining; b_e - outer lining. b_i and b_e may be equal or different [9].

In general, the performance of the sealing gasket depends on the gasket pressure produced by the flange joint and working conditions

One of the major problems of flange joints is fluid leakage through the pores between the gasket and the flange. The cause of these leaks is - after a while - the reduction of the contact pressure between the gasket and the flange due to bolts loosening and the permanent deformation of the gasket. When assessing the pressure on the gasket, some authors took into consideration the nonlinear behavior of the gasket [10]; it has been found that fluid leakages occur in flange rotations even below $0.3°$. Flange rotation and contact pressure value on the gasket depend on the gasket material, the pre-clamping force between bolts and the sealing fluid pressure.

Flange rotation in the case of a single gasket is higher than that of two gaskets. Untightening occurs at higher internal pressures if the inner gasket has a greater width than the outside width [9].

• Mixed sealing gaskets. When undergoing stresses at high temperatures, high pressures or pressure fluctuations, non-metallic gaskets may have low sealing performance

Under creep conditions the stress in bolts sometimes relaxes up to 70%. The decrease in bolt stress results in a reduction in the pressure on the gasket which further results in loss of tightness and leakage [11]. To avoid these drawbacks, one resorted to mixed gaskets [11]; [12] in three concentric rings: a metal outer ring, a middle graphite ring thicker than the metallic one, and a narrower inner metal ring (Fig. 7). In paper [12] such gaskets were investigated in cases of loading under creep conditions.

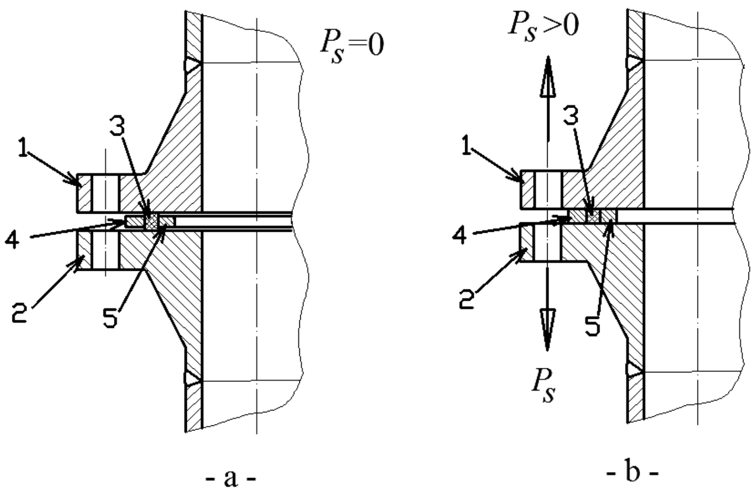

Fig. 7. Flange joint (*1* and *2*) with mixed gasket from three concentric rings: *3* - graphite gasket; *4* - outer metal gaskets; 5 - inner metal ring [11]:
a – stress free joint ($P_s = 0$); *b* – joint tightened with force $P_s > 0$.

The thickness of the graphite ring, acting as a sealing ring, is greater than that of the metal rings. In the case under study in paper [11] one used the sealing gasket shown in Fig. 8, wherein the thickness of the non-deformed graphite gasket is 2 mm larger than the one of the outer metal ring

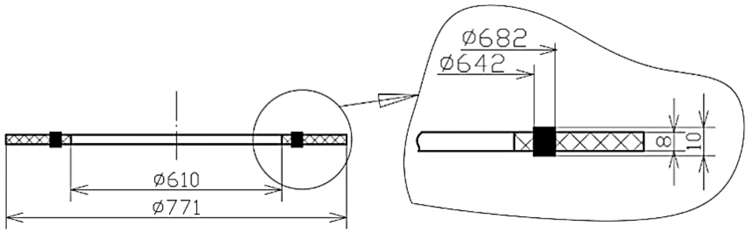

Fig. 8. Mixed metal-to-metal contact gasket [[11]].

The inner metal ring is intended to constrain the graphite ring to the inside diameter. Metallic rings constrain deformation and prevent the fracture of the graphite gasket. The behavior of the graphite gasket under compressive stress is nonlinear (OA in Fig. 9). When unloading (continuous reduction of pressure on the gasket, q) the latter remains residually deformed (OB) when $q = 0$. When the graphite ring has reached the thickness of the outer metal ring (point A), the deformation of the graphite ring remains constant. In the case under study, that happens with a 2 mm deformation of the graphite gasket, which is equal to the difference between its thickness and the thickness of the outer metal ring.

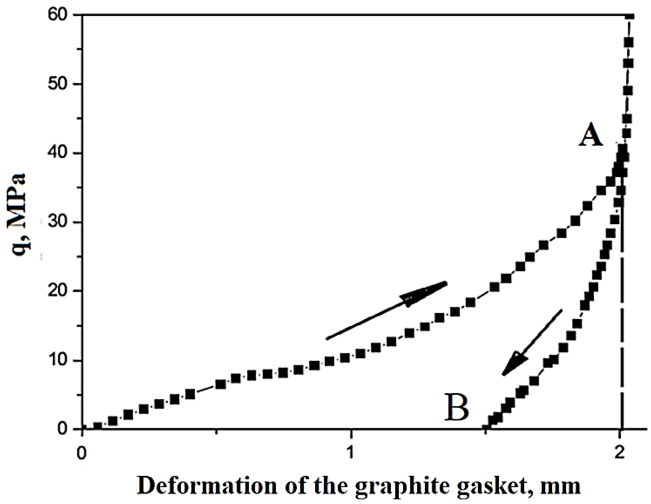

Fig. 9. Non-linear behavior of the graphite gasket: the dependence of seal pressure, q, on the deformation of the graphite gasket in the direction of its thickness [[11]].

At this point the flanges come into contact with the outer metal ring, which provides additional sealing and compensation for the pressure drop on the gasket as a result of relaxation the stress inside the bolts. Researches carried out [11] proved the necessity to provide a stress inside the bolts ranging $(0.3...0.4)\cdot\sigma_Y$ (where σ_Y is the bolt material yield point) under creep conditions. The use of mixed gaskets (Fig. 8) can reduce the relaxation effect induced by creep.

3. Flange joints without sealing gaskets

In recent years, research has been carried out on flange joints without a sealing gasket, so that the sealing is done by direct contact between the two flanges (Fig. 10). The contact pressure is ensured by tightening the bolts.

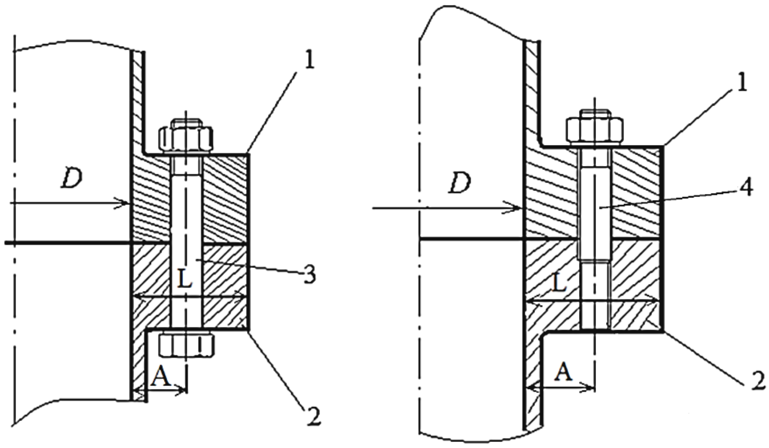

Fig. 10. Flange joint without sealing gasket:
1, 2 - flanges; 3 - bolt; 4 – stud bolt [[13]].

The inner pressure fluid, *p*, tends to "slip" at the interface between the two flanges; it creates an "opening" similar to a crack, along length a_0 (Fig. 11).

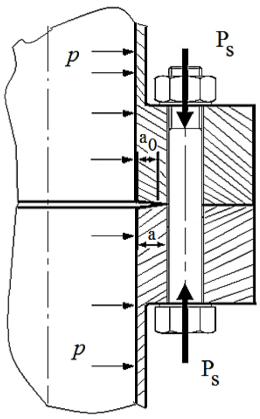

Fig. 11. "Opening" similar to a crack
generated by fluid under pressure [13].

The fluid drainage occurs when $a_0 = a$, where *a* is the distance from the inner surface of the flange to the edge of the bolt hole; the fluid comes out through the flange bolt hole. Generally, the bolt axis is very close to the inner surface of the flange/vessel. It has been found [13] that untightening occurs at inner pressure, *p*, that:

– it is directly proportional to the pre-loading force in the bolts;

– it decreases with an increase in distance, *A*, between the bolt axis and the inner surface of the vessel (Fig. 10);

–it decreases as the pitch between the bolts increases. It is recommended that one should choose as small a pitch as possible;

–it increases by increasing the flange width to a certain value, after which the unsealing pressure remains constant;

–it is little sensitive to the flange thickness; the only exceptions are the very thin, flexible flanges, that influence the unsealing pressure.

4. Researches concerning flange joints under creep conditions

Flanges under creep conditions were analyzed in works [14]-[16].

The effect of sealing gasket relaxation on the annular flange joint was investigated in works [14]; [15]. The continuous relaxation of the gasket over time reduces the stress in the gasket to a critical value sealing-wise, which can cause the gasket to break due to the leakage flow rate. In order to avoid fluid leakage, the pressure on the gasket must be higher than the sealing pressure $q_{et} = m \cdot p_i$, where p_i is the maximum fluid pressure during the flange joint operation, while $m > 1$ - a characteristic the gasket factor. A finite element analysis, in a given case, showed a 30 - 40% reduction in the stress on the sealing gasket after 15,000 hours [14].

It has been found that in flanges of relatively large diameter, the rotation of the flange becomes important for the value of the contact pressure between the flange and the gasket [15].

Current design norms do not take into consideration the effect of high temperatures on fluid leakage and on the strength of flange joints. It is necessary to introduce creep influence in the current design method [16]. The theory developed in paper [[16]] for determining the creep effect on the flanges was verified with the 3-D - finite elements method. Fig. 13 shows the relaxation of the bolts (Fig. 13, *a*) and the relaxation of the sealing gasket (Fig. 13, *b*) for a hub flange assembled with a plane plate, as components of a heat exchanger (Fig. 12).

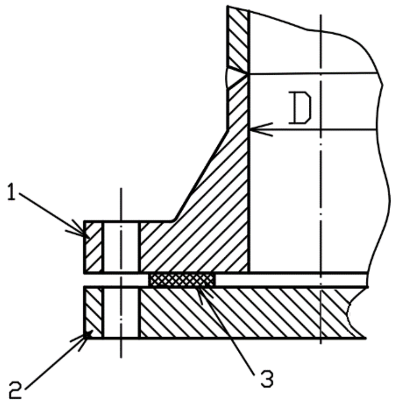

Fig. 12. Assembling a hub flange (*1*) with a plane plate (*2*) in a heat exchanger;
3 – sealing gasket [16].

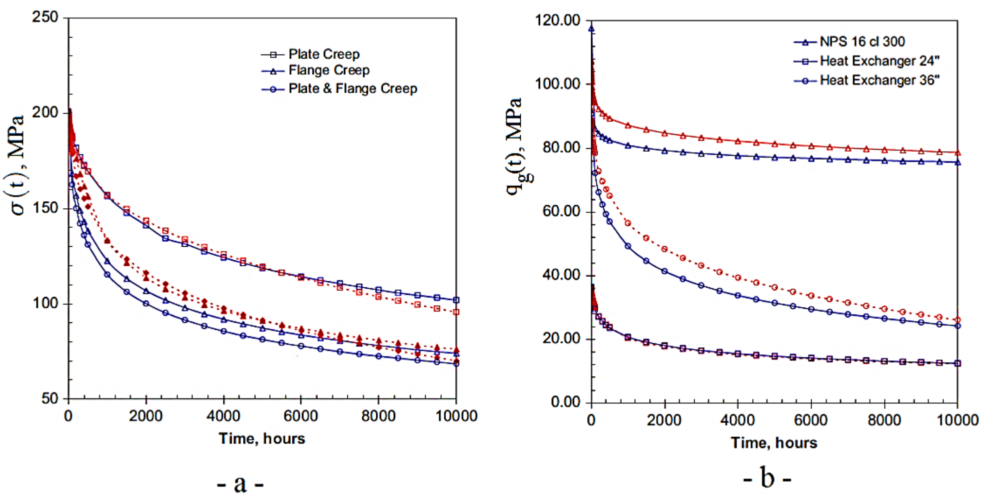

Fig. 13. For the heat exchanger with *D* = 914.4 mm, the decrease in time due to creep (relaxation) of: a- tensile stress in the bolts (*a*); - the compressive stress in corrugated metal sealing gasket (*b*) [16].

5. Flange joints with cracks

For flange joints used in pipeline systems in the oil, gas, chemical, nuclear industries, the presence of cracks may have catastrophic effects. Such flange joints are statically under load, but in some cases they may be fatigue or seismically loaded. In such cases, sometimes, the flanges are built with non-standard sizes [17]. Additional effects on flange joints may occur due to: a – flange rotation due to the bending produced by the pipes to which they are being joined (Fig. 14, *a*), or the axial displacement caused by additional traction forces induced by the pipes (Fig. 14, *b*).

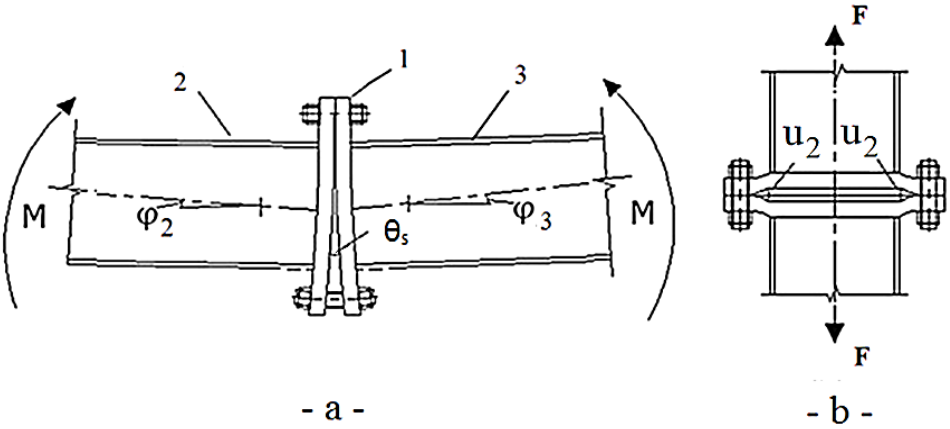

Fig. 14. Additional effects upon flanges: a - additional rotation θ_s of flanges (*1*) due to rotation φ_2 and φ_3 of pipes *2* and *3* to which they are joined; b - axial displacements u_1 and u_2 determined by the axial force in the pipes, *F* [17].

Fig. 15 shows photographs of flange joints where there occurred some fracture in the area of thermal influence of the welding (Fig. 15), or fracture and buckling under fatigue loading with a very small number of cycles (Fig. 15, *b*).

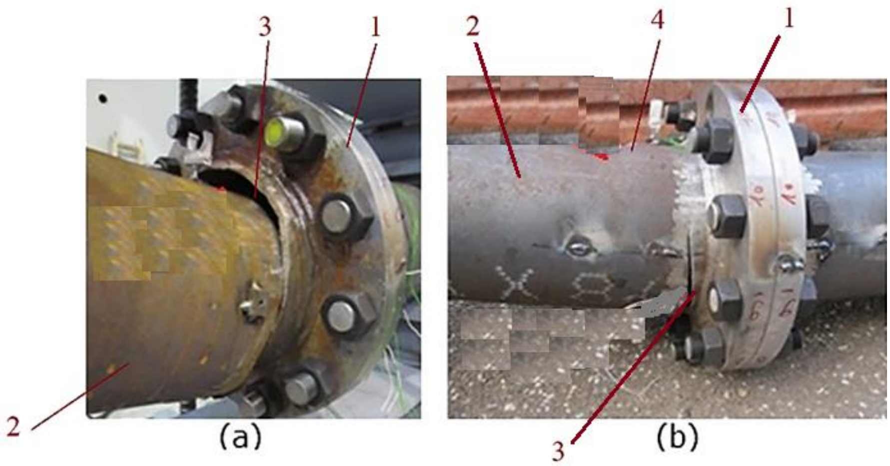

Fig. 15. Fracture (*a*), fracture and buckling (*b*) in the zone of thermal influence of the weld between flange (*1*) and pipe (*2*); 3 - fracture; 4 - buckling [17].

In such cases, for the materials of the flange joint components are important: the flange leakage rate, the pipe material ductility and the capacity of load-induced energy dissipation.

In the elbow with flanges (Fig. 16), cracks have been found in the thermal influence zone of the weld between the flange and the elbow [18].

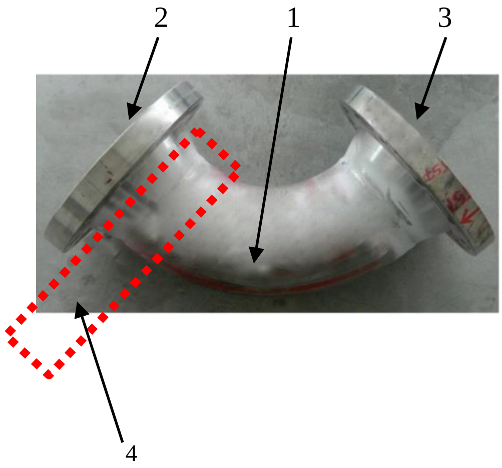

Fig. 16. Elbow (*1*) with welded flanges (*2* and *3*); 4 - crack area [18].

The flanges are used for a removable assembly when long multi-sectional cylinders are made and, for this purpose, no welding of cylinder sections is used. An example is the rotary dryer in Fig. 17 [19].

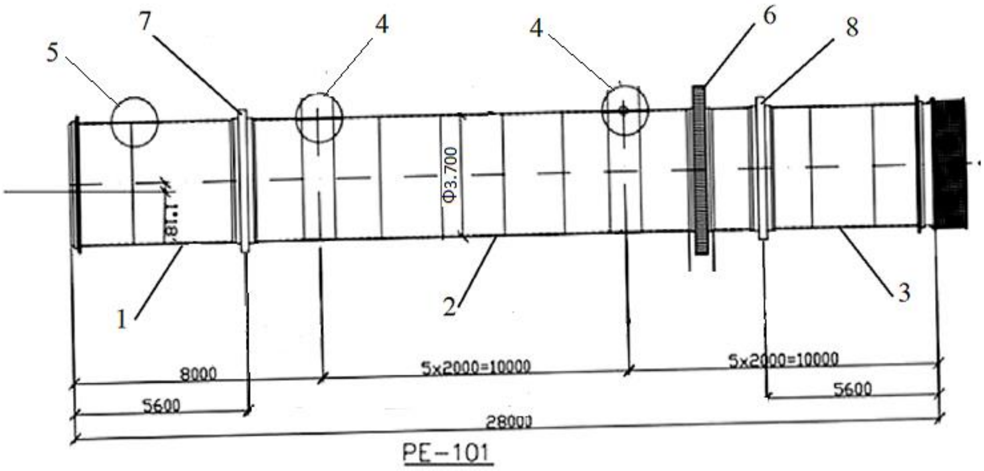

Fig. 17. A rotary dryer made up of three sections (1-3)
assembled together with a removable flange (*4*; *5*); *6* - toothed crown
for triggering drum rotation (2.5 rotations/minute); *7*; *8* – support rings [19].

These flange joints are statically as well as variably loaded as they are subjected to 2.4×10^6 cycles throughout their use.

After about two years of dryer use, cracks were observed at the surface of the annular flange, near the weld with the shell (Fig. 18).

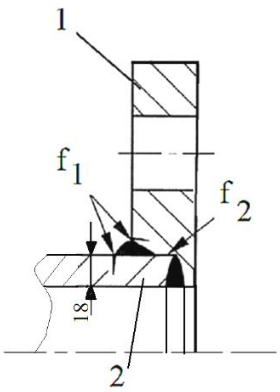

Fig. 18. Cracks (f_1 și f_2) at the joint of the flange (*1*) with cylindrical shell (*2*). [19]

One of the causes of cracking was the use of a discontinuous weld across the entire thickness of the flange. Consequently, one should not use partial welds along the flange thickness on large, non-standard flanges, but weldings that go across the entire thickness of the flange.

The non-standard flanges used to assemble tubular elements, structural components such as bridges, windmill towers, radio/television antennas, offshore structures, etc., undergo various types of loads (induced by forces in the operating conditions, wind or wave loads, etc.) or loads of a different nature (monotonic, cyclic ...) and they have been examined in numerous papers, including [20]-[29]; loads for a small number of cycles and unlimited cycles have been taken into consideration.

A structure made of a tube with welded end flanges (Fig. 19, *a*) was subjected to torsion featuring pulsating and symmetrically alternating pulses, respectively (Fig. 19, *b*). The initiation of cracks, determined on a fatigue testing gauge, occurred at the root of the weld (Fig. 20).

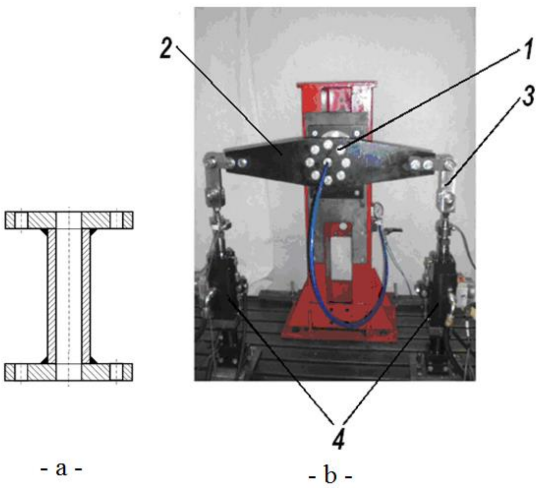

Fig. 19. Tubular structure with welded end flanges (a); under torsional fatigue loading (b): *1* - flange bolts; *2* - arm; *3* - connecting rods; *4* - hydraulic actuator [30].

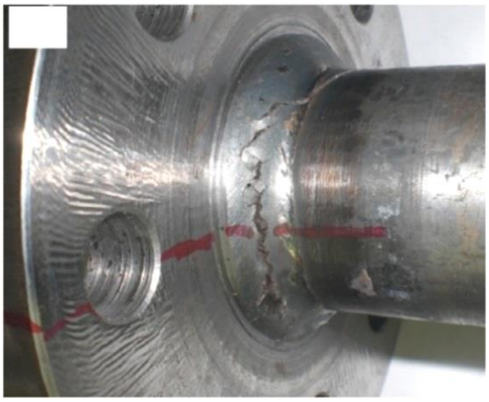

Fig. 20. Example of torsional fatigue failure by pure torsion
at the root of the weld [30].

6. Flange joints additionally loaded with an axial force

• Often literature, but especially design rules analyze and/or deal with the problem of flange joint strength, while the problem of leakage has remained a matter of great interest to industry. Similarly, the superposition of inner pressure loading and axial forces is little studied in the literature.

Paper [31] looks into the behavior of a flanged joint under the load of bolt pre-loading force, inner pressure and axial force by following two hypotheses: one where the pre-loading calculated according to ASME code [32] and another where the force of pre- loading is calculated according to industrial strategy [33].

Generally, the performance and strength of a flange joint, as previously shown, depends on the bolt pre-loading and the bolt sequence of assembly. The analysis of a hub flange joint with a spiral winding gasket under bolt force loading, inner pressure and additional axial force (Figure 21) has yielded the following results [31]:

– an increase in the bolt pre-loading force improves tightness but causes higher stresses in the flange hub, which reduces its mechanical strength;

– a relatively low pre-loading force may result in reduced flange joint tightness;

– the pressure on the gasket is not influenced by the additional axial force. However, the additional axial force along with the internal pressure causes the pressure on the gasket to be reduced, which has the effect of relaxing the sealing gasket;

– above a certain value of the additional axial force (lower in the pressure test than in the case of in-service pressure during operating conditions), the contact pressure between the flange and the gasket may become less than the recommended tightening pressure for the gasket, which affects tightness.

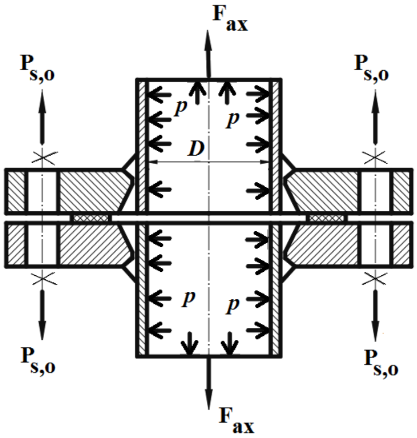

Fig. 21. Flange joint under bolt force load ($P_{s,0}$), inner pressure (p) and additional axial force (F_{ax}).

Similar conclusions were drawn in the wake of the analysis of a flange joint with two annular gaskets separated by a space wherein the fluid passing through the first sealing ring [34] was to be discharged.

• Some flange joints are provided with pins or shear cones (Fig. 22) designed to ensure the strength of the joint under transverse loading forces; strength under axial loads is ensured by bolts.

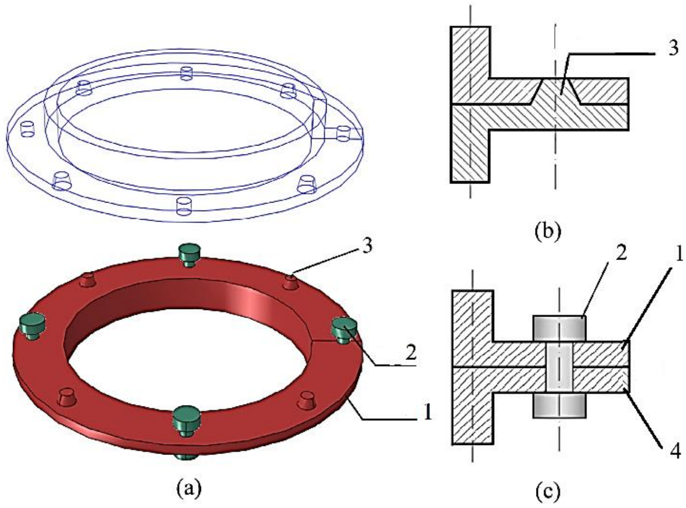

Fig. 22. Bolted flange (1; 4) connection with shear pin/cone (3):
a – assembly structure ; b – section view of shear pin/cone; c – section view of bolted flange connection [35];[36].

Such flange joints are used in dynamically loaded structures. Paper [36] has undertaken simulations of the behavior of such a joint under the harmonic or impact loads produced by a bending moment. It has been found that part of the transverse force is redirected by the pins to axial loading.
An analysis was made of the dependence of the maximum equivalent stress in the pin joint on the ratio of bending frequency and the axial loading frequency.
In one of the analyzed cases it was found that the maximum stress value is affected by the inclination of the pin generator, by the circular frequency of the vertical movement and by the pulse of the shear stress.
Some telecommunication towers are provided with circular flanges welded by tubular elements and tightened together with pretensioned bolts. When the flanges are axially loaded, F_{ax}, the bolts undergo both axial stresses as well as bending moments (Fig. 23). The value of the stress induced by the bend in the bolt depends on the bolt pretensioning and the flange thickness [37].

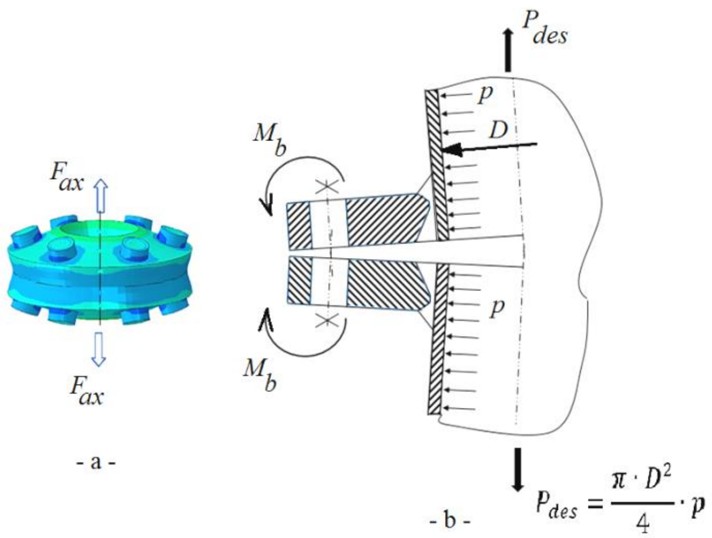

Fig. 23. Flange joint with bolts subject to bending as a result of their pretensioning and axial force application, F_{ax}, to the pipes to which they are connected [[37]] (*a*) and the unloosening force P_{des}, produced by inner pressure, *p*(*b*).

In the case of an axial load, F_{ax}, without internal pressure, for a given construction, the following [37] were found :
– displacement *u* - measured at the level of the weld between the flange and the pipe flange - increases with the increase in axial stress in the bolts, $\sigma_s = F_{ax}/A_s$, where A_s is the area of the bolt cross section (Fig. 24);

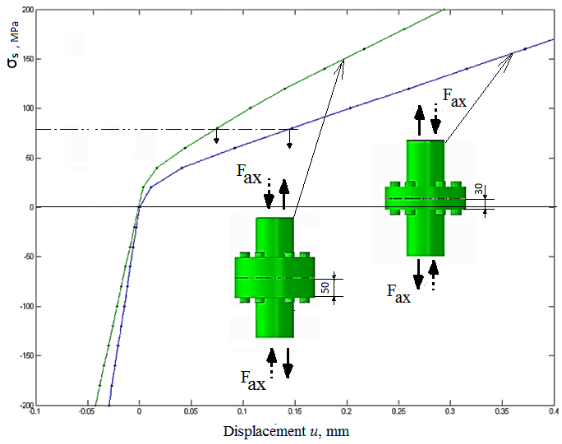

Fig. 24. Dependence of axial bolt stress, σ_s and displacement *u* for two flange thicknesses (*50* and *30 mm*, respectively) [37].

– under compressive axial force ranging in the analyzed case between zero and -200MPa, the diagram $\sigma_s - u$ is linear;

– under axial traction force, ranging between zero and 200MPa, the diagram $\sigma_s - u$ is partly nonlinear and dependent on the flange thickness. For the same stress value, σ_s, the displacement is greater in the thinner flange which has less stiffness;

– the pre-loading force for a bolt is written as, $P_{s,0}$ (Fig. 25), and $\sigma_{s,0} \sim P_{s,0}$; pre-loading the bolts with $\sigma_s > 0$ also influences the flange deformation. The higher the stress, σ_s, the greater deformation *u*;.

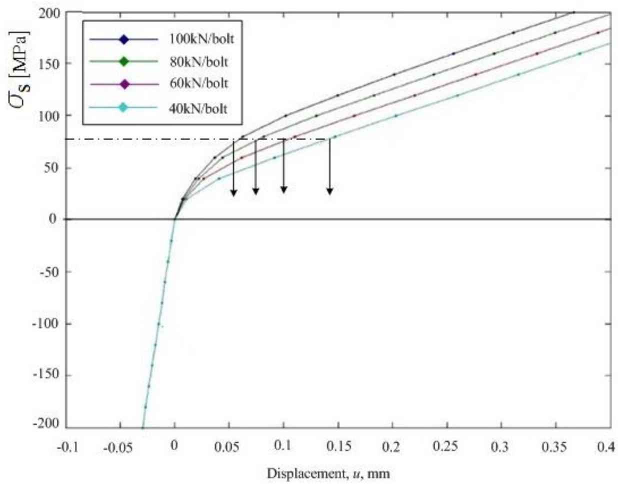

Fig. 25. Dependence of bolt axial stress, σ_s, upon displacement *u* for four values of the bolt

pre-loading force, for the 30 mm flange (Fig. 24) [37].

In the case of vibration a tubular structure occurring in pipes connected to each other by flanges, the stress in the bolts varies between $\sigma_{s,0}$ (bolt pre-loading stress) and a maximum value, σ_{max}, which results in bolt material fatigue, with the risk of breaking one or more bolts. As a result, the stiffness of the flange joint decreases, which increases the opening or rotation θ of the flanges, as well as the additional loading of the other bolts, which should be avoided.

7. The influence of the bolt tightening method upon the tightness and strength of flange joints

From research on the importance of stress and stress uniformity in bolts in the case of hub flange joints, the following conclusions [38] have emerged:
– the axial stress variations induced by pre-loading between the bolts of the same joints are lower in the case of *controlled tightening stress* (with a hydraulic bolt tensioner and the manual tightening of the nut) as compared to tensioning by torque control (with the key torque). In the first case the tightening of the bolts can be done simultaneously, whereas in the second case the tightening is done successively;
– when stress occurs via controlled tightening, the bolts are subjected to close bending moments, whereas in the case of torsional torque control there are differences between the bending moments applied to each bolt;
– with increased flange diameters, stress variations in bolts, gasket and flange decrease;
– the torques required to tighten the bolts provided in the ASME code [39] are larger than for the industrial ones [40]] With the ASME recommended torque moments, the flange joints are overloaded and the yield is started earlier.
Due to the limitations imposed by experimental work on flange joint research, many researchers have performed detailed studies by using the finite element method [38], [41]-[55].
The tightening of the bolts and the order in which it is carried out affects both the uniformity of bolt loading as well as its tightness. In [12] these problems were analyzed in the case of a flange joint provided with a mixed gasket of three concentric elements
It is estimated that 50 - 80% of the leakage between flange joints elements was due to the uneven grip of the bolts [12]. Bolt tightening control is essential to ensure sealing in flange joints. Tightening the bolts cannot be done simultaneously. It is done in a certain order that "dictates" the degree of unevenness in bolt tightening.
There have been compared three types of bolt tightening in the case of an 8 bolt flange joint [12]. The ASME recommendations for clamping methods refer to "floating bolted flanges joint".
The star assembly pattern (Fig. 26, a) is recommended by ASME PCC-1 [32] as being the best. The load on the bolt increases in three consecutive tightening rounds to reach 100% of the prescribed preloading, after which the bolts are tightened

successively still two more times in a clockwise direction to obtain a uniform bolt loading [12]:
- *first turn:* 20 - 30% of the prescribed preloading. Tightening order: 1, 5, 3, 7 - 2, 6, 4, 8.
- *second turn*: 50 - 70% of the prescribed preloading. Tightening order: 1, 5, 3, 7 - 2, 6, 4, 8.
- *third turn*: 100% of the prescribed preloading. Tightening order: 1, 5, 3, 7 - 2, 6, 4, 8.
- *fourth and fifth turn*: 100% of the prescribed preloading. Order of tightening: successive, clockwise.

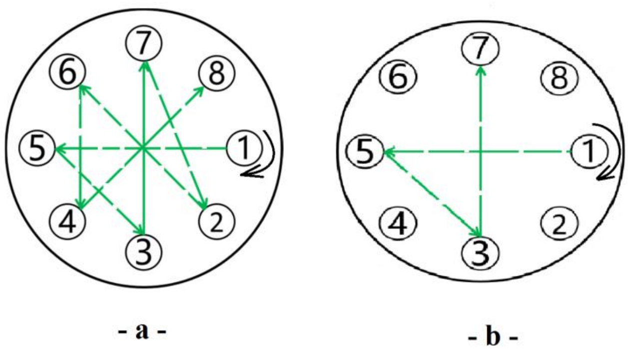

Fig. 26.a – Star assembly pattern of flange joint bolts;
b - Tightening with method #3 [12]]

The tightening marked as # 3 recommended by ASME [32] is the simplest; at the beginning only four bolts are fastened (Fig. 26, b), after which the tightening is performed in clockwise direction
- *first turn:* 20 - 30% of the prescribed preloading. Tightening order: 1, 5, 3, 7;
- *second turn:* 50 - 70% of the prescribed shaving. Tightening order: 1, 5, 3, 7
- *third turn*: 100% of the prescribed preloading. Tightening order: 1, 5, 3, 7;
- *fourth and fifth turn*: 100% of the prescribed preloading; tightening clockwise.

The SH method of tightening was devised in [12] for mixed seals (Fig. 27). In the first step of bolt tightening, the gasket acts as a floating bolted flanges joint because the tightening is transmitted to the graphite ring. After three clamping rounds, the graphite ring is compressed to the thickness of the outer metal ring. After that, the sealing gasket acts like a hard metallic gasket. The sequencing of tightening is shown in Fig. 27 [12]:
first turn: 50% of F_{MMC}, in the sequence: 1, 5, 3, 7 - 2, 6, 4, 8;
second turn: 110% tightening of F_{MMC}, in the order 1, 5, 3, 7 - 2, 6, 4, 8 (provides the metal to metal contact between the flange and the outer ring of the joint gasket);

- *third turn*: 110% of F_L, in the order: 1, 5, 3, 7 - 2, 6, 4, 8 (causes more uniform bolt loading);
- *fourth and fifth turn*: 100% load of F_L clockwise.

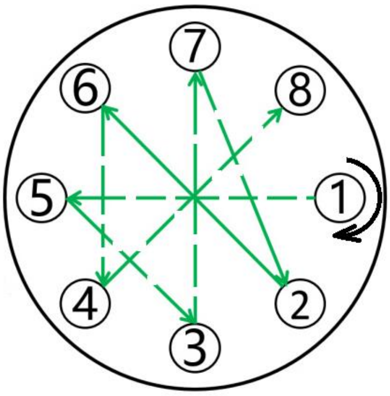

Fig. 27. SH method for tightening mixed seals: The F_{MMC} is the minimum preloading force that ensures metal-to-metal contact (flange-metal outer ring); F_L – bolt preloading force in bolts, prescribed [12].

Fig. 28 shows the experimental installation for determining the correlation between bolt and flange joint sealing.

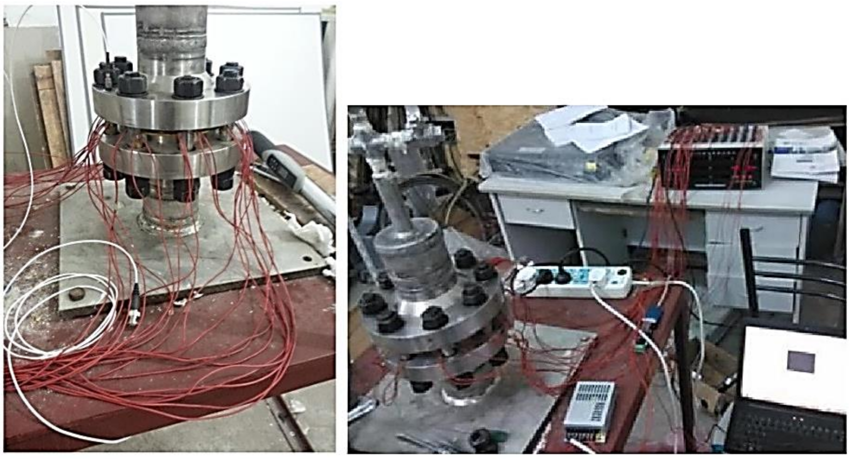

Fig. 28. Experimental setup for examining the correlation between bolt loading and flange leak rate [12].

Bolt tightening was performed with a dynamometric torque wrench, by torsion. Tensiometers were used to determine bolt loads and a data acquisition system for specific bolt strains. A sensor for measuring the pressure difference in the system allows you to indicate the moment of unsealing.

From the comparison of the results for the three tightening alternative, one found that the dispersion of clamping force values and leakage rates were the lowest in the case of fixing by using the SH method (Fig. 27), followed by Method # 3 (Fig. 26, b) and the star assembly pattern method (Fig. 26, a). The SH tightening method has obvious advantages over the other two investigated methods, including the use of a smaller number of tightenings.

8. Results used to assess the strength and tightness of flange joints

Some published results [63]-[67] can be used by expanding the research initiated in these papers.
For example, some proposals for improving the strength calculation methods are outlined in works [64]; [65]; [67]. The leakage problem, in correlation with sealing gasket behavior, is addressed in paper [63] concerning the fatigue loading of sealing gaskets. The behavior of the sealing gasket when considering the reduction in sealing pressure between the two bolts of a flange joint is dealt with in [66] which defines and proposes a pressure attenuation factor on the gasket in the circumferential direction of the flange.

9. Conclusions

Some previous works emphasized the additional calculations required to complement the official calculation method of flange joints. Work [61], for example, rationalizes and suitably adopts the proposals for rounding off calculations for flange joints. In order to complement the official calculation methods for flange joints, further experimental checks are required regarding:
– the temperature difference between the flange and the bolt while in service and in the transient mode;
– in-service behavior of the sealing gasket;
– rotation of the flanges and rotation of the bolts during pre-tightening and after pressurization;
– bolts bending after pressurizing the vessel or the pipes;
– bolt relaxation under creep conditions.

References

[1] Jinescu V.V., *Utilaj tehnologic pentru industrii de proces*, Vol. 3, Ed. Tehnică, București, 1988.
[2] Homilius C., *The benefits of cold-work in middle eastern brownfield projects and high pressure subsea piping operations. Society of Petroleum Engineers,* Abu Dhabi International Petroleum Exhibition and Conference 2012, ADIPEC 2012, Sustainable Energy Growth: People, Responsibility, and Innovation 2012, p. 2116-2122.
[3] Quickflange. Quickflange Tehnology & Products. www.quickflange.com
[4] Barsoum I., Khalaf A.M., *Evaluation of a Pipe-Flange Connection Replacing Fusion Welding*, Procedia Engineering, **130**, 2015, p 446 – 452.

[5] Henriksen J., Nordhagen H.O., Hoang H. N., Hansen M.R., Thrane F.C., *Numerical and experimental verification of new method for connecting pipe to flange by cold forming*, Journal of Materials Processing Technology, **220**, 2015, p. 215–223.

[6] Haruyama S., Nurhadiyanto D., Choiron M.A., Kaminishi K., *Influence of surface roughness on leakage of new metal gasket*, International Journal of Pressure Vessels and Piping, **111-112**, 2013, p. 146-154.

[7] Bramsiepe C., Schembecker G., *Fugitive emissions from liquid-charged flange joints: a comparison of laboratory and field data*, Environ. Sci. Technol., **43**, 2009, p. 4498–4502.

[8] Bramsiepe C., Pansegrau L., Schembecker G., *A model to predict fugitive VOC emissions from liquid charged flange joints with graphite gaskets*, Chemical Engineering Journal, **159**, 2010, p. 11–16.

[9] Nelson N.R., Prasad N.S., *Sealing behavior of twin gasketed flange joints*, International Journal of Pressure Vessels and Piping, **138**, 2016, p. 45-50.

[10] Bouzid A., Derenne M., *Analytical modeling of the contact stress with nonliniar gasket*, International Pressure Vessels Technology, **124**, 2002, p.47-53.

[11] Chen Y., Xu Y.Y., Huang L.Y., Xu Y.H., Guan K.S., *Creep Behavior of Metal-to-Metal Contact Bolted Flanged Joint*, Procedia Engineering, **130**, 2015, p. 214-220.

[12] Zhang L.Z., Liu Y., Sun J.C., Ma K., Cai R.L.,Guan K.S., *Research on the Assembly Pattern of MMC Bolted Flange Joint*, Procedia Engineering, **130**, 2015, p.193-203.

[13] Beghini M., Bertini L., Santus C., Guglielmo A., Mariotti G., *Partially open crack model for leakage pressure analysis of bolted metal-to-metal flange*, Engineering Fracture Mechanics, **144**, 2015, p. 16–31.

[14] Mourya R.K., Banerjee A., Sreedhar B.K., *Effect of creep on the failure probability of bolted flange joints*, Engineering Failure Analysis, **50**, 2015, p. 71–87.

[15] Luyt P.C.B., Theron N.J., Pietra F., *Non-linear finite element modelling and analysis of the effect of gasket creep-relaxation on circular bolted flange connections*, International Journal of Pressure Vessels and Piping, **150**, 2017, p. 52-61.

[16] Nechacke A., Bouzid A.–H., *On the use of plate theory to evaluate the load relaxation in bolted flanged joints subjected to creep*, International Journal of Pressure Vessels and Piping, **85**, 2008, p. 486-497.

[17] Reza M.S., Bursi O.S., Paolacci F., Kumar A., *Enhanced seismic performance of non-standard bolted flange joints for petrochemical piping systems*, Journal of Loss Prevention in the Process Industries, **30**, 2014, p. 124-136.

[18] Sridhar Idapalapati, Kelvin Loh K.M., Stedston Yeo, *Failure investigation of welded high nickel stainless steel flange-elbow joint*, Engineering Failure Analysis, **78**, 2017, p 99–109.

[19] YAPICI A., SARACOGLU G., *Fatigue analysis of bolted flange joints of a rotary dryer*, Engineering Failure Analysis, **63**, 2016, p. 182–190.

[20] Hoang Van-Long, Jaspart Jean-Pierre, Demonceau Jean-François, *Behavior of bolted flange joints in tubular structures under monotonic, repeated and fatigue loadings I: Experimental tests*, Journal of Constructional Steel Research, **85**, 2013, p. 1–11.

[21] Bouwan L.P., *Bolted connections dynamically loaded in tension*, J Struct Div ASCE 1982, p.108.

[22] Couchaux M., *Comportement des assemblages par brides circulaires boulonnés*, PhD thesis, France, INSA of Rennes, 2010.

[23] Cao J.J., Bell A.J., *Determination of bolt forces in a circular flange joint under tension force*, Int J Press Vessel Pip, **68**, 1996, p. 63–71.

[24] Cao J.J., Packer J.A., *Design of tension circular flange joints in tubular structures*, Eng J First Q, 1997, p. 17–25.

[25] Hitube project, *Design and integrity assessment of high strength tubular structures for extreme loading conditions*, Grant N0 RFSR-CT-2008-00035, Deliverable D.4: Test data on bolted tubular connections, 9-2011.

[26] Igarashi S., Wakiyama N., Inoue K., Matsumoto T., Murase Y., *Limit design of high strength bolted tube flange joint, Part 1 and 2*, Journal of Structural and Construction Engineering Transactions of AIJJapan, Department of Architecture reports, Osaka University, August 1985.

[27] Jaspart J.P., Pietrapertosa C., Weynand K., Busse E., Klinkhammer R., *Development of a full consistent design approach for bolted and welded joints in building frames and trusses between steel members made of hollow and/or open sections: application of the component method*, Volume 1, practical guidelines, CIDECT Report 5BP — 4/05, 2005.

[28] Nemati N., Le Houedec D., Zandonini R., *Numerical modelling of cyclic behaviour of the basic components of steel end plate connections*, Adv Eng Softw., **31**, 2000, p. 837–849.

[29] Wardenier J., Kurobane Y., Packer J.A., Dutta D., Yeomans N., *Design guide for circular hollow section (CHS) joints under predominantly static loading*, CIDECT, 2008.

[30] Meneghetti G., De Marchi A., Campagnolo A., *Assessment of root failures in tube-to-flange steel welded joints under torsional loading according to the Peak Stress Method*, Theoretical and Applied Fracture Mechanics, **83**, 2016, p. 19–30.

[31] ES/090 Rev:1. Design and Engineering Practices (DEPs), DEP 31.38.01.15.Gen (Piping Class Exploration and Production) 1998.

[32] ASME PCC-1, Guidelines for Pressure Boundary Bolted Flange Joint Assembly, 2013.

[33] Khan N.B., Abid M., Jameel M., Wajid H.A., *Joint strength of gasketed bolted pipe flange joint under combined internal pressure plus axial load with different (industrial and ASME) bolt-up strategy*, Journal of Process Mechanical Engineering, 2015, p. 1 – 10.

[34] Sánchez A.M., Bertizzolo R., Chavan R., Gagliardi M., Goodman T., Landis J.-D., Saibene G., Silva P.S., Vaccaro A., *Mechanical analyses of the waveguide flange coupling for the first confinement system of the ITER electron cyclotron upper launcher*, Fusion Engineering and Design, **109–111**, 2016, p. 532–538.

[35] Lu X., Wang P., Wang J., Chang J., Zhang Y., Guan Z., *Nonlinear behavior of bolted flange connections containing shear pin (cone)*, Chin. J. Comput. Mech., **32**, 2015, p.503–511.

[36] Lu X., Zeng Y., Chen Y., Xie X., Guan Z., *Transient response characteristics of a bolted flange connection structure with shear pin/cone*, Journal of Sound and Vibration, **395**, 2017, p.240–257.

[37] Blachowski B., Gutkowski W., *Effect of damaged circular flange-bolted connections on behaviour of tall towers, modelled by multilevel substructuring*, Engineering Structures, **111**, 2016, p. 93–103.

[38] Abid M., Khan A., Nash D.H., Hussain M., Wajid H.A., *Simulation of Optimized Bolt Tightening Strategies for Gasketed Flanged Pipe Joints*, Procedia Engineering, **130**, 2015, p. 204-213.

[39] ASME Boiler and Pressure Vessel Code. Section VIII, Divison – 1, Rules for bolted flanges connection with ring type gasket, New York USA, 2007.

[40] Garlock Gasket Manufacturer, http://www.garlock.com.

[41] Cao D., Xu H., *3-D finite element analysis of bolted flange joint considering gasket non-linearity*, ASME Int. PVP Conf., **382**, 1999, p. 121-126.

[42] Takkaki T., Fukuoka T., *Bolt-up strategy for pipe flange connections using finite element analysis*, ASME Int. PVP Conf., **405**, 2000, p. 143-149.

[43] Takkaki T., Fukuaka T., *Finite element analysis of bolt-up operations for pipe flange connections*, ASME Int. PVP Conf. **416**, 2001, p. 141-147.

[44] Takkaki T., Fukuaka T., *Systematical PE analysis of bolt assembly process of pipe flange connections*, ASME Int. PVP Conf., 2002, p. 147-152.

[45] Takkaki T., Fukuaka T., *Effective bolting up procedure using fea and elastic interaction coefficient method*, ASME Int. PVP Conf., **478**, 2004, p. 155-162.

[46] Abid M., Experimental and Analytical Studies of Conventional (Gasketed) and Unconventional (Non-gasketed) Flanged Pipe Joints (With Special Emphasis on the Engineering of 'joint Strength' and 'sealing'), PhD. Thesis, 2000.

[47] Nagata S., Shoji Y., Sawa T., *A simplified modeling of gasket stress-strain curve for FEM analysis in bolted flange joint design*, ASME Int. PVP Conf., **433**, 2002, p. 53-58.

[48] Tsuji H., Nakano M., *Bolt preload control for bolted flange joint*, ASME Int. PVP Conf., 2002, p. 163-170.

[49] Sawa T., Matsumoto M., Nagat S., *Effects of scatter in bolt preload of pipe flange connections with gasket on sealing performance*, ASME Int. PVP Conf., 2003, p. 65-75.

[50] Fukuoka T., Sawa T., *Evaluation of the tightening process of bolted joint with elastic angle control method*, ASME Int. PVP Conf., 2004, p. 11-18.
[51] Zhang M., Jiang Y., Lee C., *Finite element modeling of self-loosening of bolted joint*, ASME Int. PVP Conf., 2004, p. 19-27.
[52] Bouzid A., Nechache A., *Creep modeling in bolted flange joints*, ASME Int. PVP Conf., 2004, p. 49-56.
[53] Shoji Y., *An effect of gasket stress distribution on tightness estimation in pipe flange connections by finite element analysis*, ASME Int. PVP Conf., **478**, 2004, p. 113-120.
[54] Takkaki T., *Effects of flange rotation on the sealing performance of pipe flange connections*, ASME Int. PVP Conf., **478**, 2004, p. 121-128.
[55] Brown W., *Efficient assembly of pressure vessel bolted joints*, ASME Int. PVP Conf., **478**, 2004, p. 163-168.
[56] Jinescu V.V., Teodorescu N., *Construcţia şi calculul îmbinărilor cu flanşe* (III), Revista de Chimie, **33**, nr. 7, 1982, p. 671-676.
[57] Jinescu V.V., Teodorescu N., Gărduş V., *Construcţia şi calculul asamblărilor cu flanşe* (IV), Revista de Chimie, **38**, nr. 8, 1987, p. 727-730.
[58] Jinescu V.V., Teodorescu N., Gărduş V., *Calculul şi construcţia asamblărilor cu flanşe*, V, Revista de Chimie, **39**, nr. 1, 1988, p. 75-78.
[59] Jinescu V.V., Teodorescu N., Gărduş V., *Calculul şi construcţia asamblărilor cu flanşe*, VI, Revista de Chimie, **39**, nr. 2, 1988, p. 185-188.
[60] Jinescu V.V., *Calculul şi construcţia utilajului chimic, petrochimic şi de rafinării*", vol. I, Editura Didactică şi Pedagogică, Bucureşti, 1983
[61] Jinescu V.V., *Utilaj tehnologic pentru industrii de proces*, Vol. 2, Ed. Tehnică, Bucureşti, 1984.
[62] Jinescu V.V., Urse G., Chelu A., *Evaluation and completion the design methods of pressure vessels flange joints*, în Revista de Chimie, **69**, nr. 8, 2018.
[63] Păunescu M., Tacă C.D., Iatan I. R., *Experimental study on the fatigue behaviour of the marsit gaskets under high constant temperature conditions*, Buletinul Institutului Politehnic Bucureşti, Seria Inginerie mecanică, nr. 1 – 2, **53**, 1991, p. 105 - 109.
[64] Iatan I. R., Alămoreanu E., Iordan N., Chiriţă R., *Calculus elements for ring neck flanges*, Romanian Academy, Branch office of Iaşi, Modelling and optimization in the machines building field, MOCM – 3 (ISBN: 1224 – 748o), 1997, pp.14 – 17.
[65] Iatan I. R., *Consideraţii asupra calculului flanşelor inelare sudate, în faza de montaj*, Buletinul Institutului Politehnic Bucureşti, seria mecanică, tom **53**, nr.1, 1981, p. 101 – 112.
[66] Iatan I. R., Roman G., *Consideratii asupra flanselor cu garnituri inelare. Presiunea de etansare*, în Revista de Chimie, **60**, nr. 11, 2009, p. 1119 – 1122.
[67] Iatan I. R., *Contribuţii la calculul mecanic al îmbinărilor cu flanşe plate inelare, sudate*, Buletinul Institutului Politehnic Bucureşti, Seria Mecanică, **55**, nr. 1, 1983, p. 50 – 60.