Dedicated to the memory of Prof. Mihail Renert

Considerations about Flange with Ring Gasket

The sealing pressure

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The paper highlights the importance of the sealing of the flange with ring gasket, the requirements that demand such constructions and the directions in which the theoretical and experimental researches characteristic to this field are performed. Original aspects of deformation of the gasket, in putting into operating or working, are detailed. We establish the expressions for evaluate the load pressure for each referred phase, taking into account the existing geometry and the deformability of the structure, both in the right of the bolts, and in the field between two consecutive bolts.

Keywords: flange with ring gasket, sealing pressure

The problem of the sealing of the flanges constituted and will constitute further a concern of a high risk both for the designers of the under pressure equipment and for the users of such mechanical structures.

The processes industrial diversity, on one hand, but also the working parameters values (pressures, temperatures, corroding and / or erosion), on other hand, require the essence of the mentioned preoccupations and without neglecting, the importance of the above, the reliability of the industrial equipment having such constitutions and, obviously, the environmental protection (industrial and human).

As it is known, the necessity of the flanges for processing equipment is required by a number of considerations related to manufacturing technology, the conditions of transportation, setting and working.

Even if the production possibilities, including those of assembling on working places allow the achievement of chemical apparatus with necessary dimensions, they are often demountable, to facilitate the setting and technical repairs during the operation.

The flanges, used in the structure of the industrial equipment under pressure, must satisfy the following requirements [1]:

- to ensure the sealing of the flanges at the temperature and working pressure;
- to be resistant (to mechanical loading and to corrosion);
- to allow the quick activation and repeated mounting or demounting;
- to be done relatively easy and to be able to be controlled;
- to be cheap enough.

The flanges fulfilled the majority of the mentioned conditions, although they are not fast enough mounting or demounting, and some types, such assembling involve a relatively high cost. Spread wide in the industrial use and the importance of flanges in the construction of the technological equipment make those adequate research themes to be met frequently.

The reliability of the flanges conditions in an essential measure, the good working of vessels, from which they are part. It aims, at the same time, to the achievement of such efficient economical constructions [1]. The existence of several methods for calculating the flanges makes more difficult the development of technical documentations with normative feature (worldwide) and even the organization of a series productions.

Analyzing the standards and norms of foreign and Romanian calculation, regarding the methods for calculating the flanges, it is found that, they are fully methods of verification and not sizing even less optimization methods.

As a result of this fact, different profiles of flanges - of different masses - may rejoin to the same conditions and check by one or other method of calculation.

As it is known, the assembling with flanges is composed of simple constructive elements: flanges, bolts, gaskets.

At the first sight the assembling with flanges seems simple, but a careful analysis proves it very complex.

Regarding the theoretical and experimental study of the flange with ring gasket is noted that it was developed, essentially, after two directions.

The study of the bearing capacity of the flanges in the elastically domain of axial symmetrical loading [6 - 12] or not [13 - 14] and in the elastic - plastic zone of loading [15].

As part of the calculations that are made, even in the said case, it is considered a safety factor related to yield limit of 1,5, which leads us to the idea that in working conditions is possible that the flow of material construction will be not produced in any area of the considered flange.

In this case, of the increasing of the external load for producing super elastic strains (yield strain), the so-called limit strain [16] is introduced, instead of the yield limit of the material, set on test tubes, through experimental tests.

Yield limit is considered that maximum stress, fictitious, which would occur in a pure elastic behaviour of the construction element, under the effect of a bending load, which appear, but in reality, certain specific remaining strains 0.2%.

In some cases the value of 1% can be assumed, too [17]. Based on the specific calculations optimal profiles of flanges have been able to be determined, from the loading point of view[15, 18].

The study refers to the sealing of the flanges and the determination of the influence of various geometrical and working factors acting on it [18 – 24, 27].

In the mentioned research upon the types of flanges, having the best packing properties and, in addition, being simple and having the cost as little as possible are studied.

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Among the factors that may lead to loss of sealing, may be listed [18]:

- the incorrect correlation between the status of the sealing areas and the pre - load of the gasket at setting;
- the pressure value for sealing during the working, conditions which must be higher than the working pressure [25, 26]; the unevenly distribution of this pressure on the surface of the gasket because, among other things of the inappropriate choice of step of the bolts;
- the stress relaxation or the creeping of the gasket and / or of the bolts;
- the difference of temperature between the constructive elements, constitutions of the assembly and the same element;
- the unwanted changes in the structure of the construction materials of the constructive elements, constitutions of the flanges and in particular, the degradation of the material of the gasket due to an inappropriate choice or a defective working.

The study of the mentioned factors is very difficult. Some approximations closer to reality are determined in the case of factor b), when the distance of the rings of the flanges and their rotation under the internal pressure should be considered.

The necessity of the uniform allocation of pressure on the gasket, both in radial direction and circumference is set off, also.

To ensure the sealing the elastic bolts have proved useful, in higher number.

This situation has the advantage that allows reducing the width of the ring of the flange (and implicitly the load arm of bolts, which produces the bending and the rotating of the ring of the flange), and as a consequence, the economy of metallic material and increase of the sealing of the structure.

Should not be overlooked the fact that a correct calculation regarding the mechanical loading of the flanges is not enough and the perfect correlation between research, design, manufacture and setting is essential.

Aspects regarding the compression of the gasket

To ensure the sealing of the flanges it is necessary that the pressure which acts on the gasket, will not fall below a certain value, established varying with the pressure and the temperature of the working environment, the material and the gasket type, the shape and the state (quality) of the areas for sealing, etc.

This pressure, acting on the areas of the gasket is accomplished by loading of the bolts, provided for this purpose.

The uniformity of the assessment of this pressure on the width of the gasket, in radial direction and circumference, depends on the number of bolts and the rigidity of the flanges.

So, the importance of the adequate choice of the step of the bolts is revealed. The bolts step must be less or equal to the given value of the formula [1]:

$$p_s = s + 2 \cdot h_f \cdot \tan \alpha_{sf}, \qquad (1)$$

where,

$$\tan \alpha_{sf} = \left(3{,}87 \cdot d_g - s\right) / \left(2 \cdot h_f\right). \qquad (2)$$

For the usual constructions of flanges, $42^o \leq \alpha_{sf} \leq 53.2^o$ [1]. The effect of the loading of the bolts, on the gasket, is “extinguished”, practically, at a distance, measured in the circumferential direction, on one side and the other side of the considered bolt axis equal with $p_s / 2$.

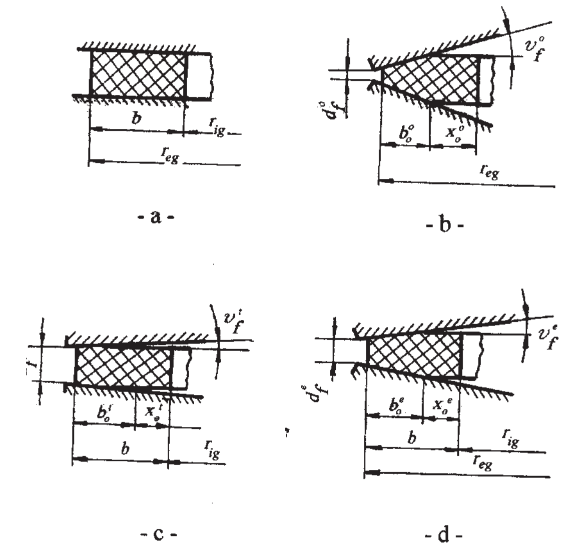

Fig. 1. Deformation of the sealing gasket at setting, putting into operating and in working:
a – no compression gasket; b - phase of setting; c - phase of putting into operating; d – working phase

It was noted that till now the researches have not yet established a relationship of calculation which determines with precision the value of the deformation of the gasket under and around the bolts [1] and still less with the distance of the area.

The difficulty derives from the fact that it is not yet fully elucidated the problem of the contact between the nut, respectively the bolt head and flange ring, regarding the pressure distribution (inconstant), that arises through the constriction.

The value of the compression pressure, in the middle of the distance between two consecutive bolts, represents, in the case of preserving of the packing, 80% from that of the gasket next bolts [1].

The gasket deformation, in a diametric plan, at setting, at putting into operating of the equipment, and during the working, too, is indicated in figure 1 [1].

Both at setting and putting into operating, respectively the working of the equipment, at the nominal value of the working parameters (pressure, temperature) the openings values between the rings of the flanges, d^o_f, d^t_f, d^e_f (at setting, put into operating and working) and those of the rotation of the rings of the flanges, ϑ^o_f, ϑ^t_f, ϑ^e_f, (at setting, put into operating and working) can be determined by experimental measurement.

These sizes, as will be seen in the following, can permit the determination, with good precision, of the values of the pressures on the gasket in the various phases of work.

It is mentioned that in the following calculations, the deformation in a radial direction of the gasket (constant width), compared with its axial deformation is neglected.

It is neglected, also, the radial displacement, under the pressure and working temperature, of the assembling composed of the rings of the flanges, bolts and the gasket (relative displacements between the mentioned elements are not taken into consideration).

Having in view the expressed simplifications, we can write [1]:

$$h^i_g = d^i_f + 2 \cdot \begin{pmatrix} r_{ef} - r_{eg} + \\ + \Delta r^i_g \end{pmatrix} \cdot \tan \vartheta^i_f, \qquad (3)$$

where:

$$0 \le \Delta r^i_g \le \left(r_{eg} - r_{ig}\right);\quad i = 0,t,e.$$

Because the angle of rotation of the flange ring is small (max. 3^0 [1]), it might be, with good approximation, $\vartheta^i_f \approx \vartheta^i_f$, so that the relationship (3) shall be accordingly adapted.

The width the gasket is compressed can be determined from the relationship (3) - modified - putting in the place of h^i_g , the h_g size. It follows:

$$\Delta r^i_g = \frac{h_g - d^i_f}{2\cdot\vartheta^i_f} - r_{ef} + r_{eg}. \tag{4}$$

In this way we can determine the reduction factor of the width of the gasket, expressed as a report between the effective width (compressed) and the initiate width of the gasket (fig. 2),

$$c^i_g = \Delta r^i_g \big/ \left(r_{eg} - r_{ig}\right) = \Delta r^i_g \big/ b. \tag{5}$$

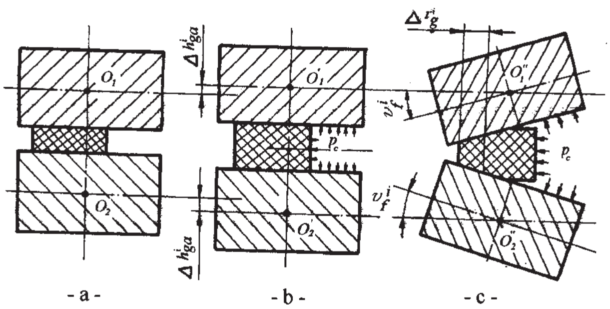

Fig. 2. Deformation of the sealing gasket a - axial compression of the gasket; b - axial return of the gasket, under the action of the internal pressure; c - rotation of the flanges rings, under the action of the bending moments

The distance between the centers of mass of the flanges rings O^1_1 and O^1_2, after the rotation with the ϑ^i_f angle, can be calculated, varying with the d^i_f opening, with formula:

$$d^i_{O^1_1,O^1_2} = d^i_f + \left(b - 2\cdot h_f\cdot\vartheta^i_f\right)\cdot\vartheta^i_f, \tag{6}$$

or, after neglecting the square of ϑ^i_f ,

$$d^i_{O^1_1,O^1_2} = d^i_f + b\cdot\vartheta^i_f. \tag{7}$$

To maintain the packing of the flange working, it is necessary that:

$$0,5\cdot\left(h_g - h^i_g\right) \rangle \max\left(h_a, h_f\right), \tag{8}$$

so that to inhibit fluid leakage to the outside under the pressure working environment, next to the gasket.

Taking into consideration a linear variation of the compression pressure of the gasket in a normal plan at the areas for sealing, diametrically, we can write:

$$\left(p_g\right)_r = \left(E_g/h_g\right)\cdot\begin{bmatrix}\Delta h^i_{mg} - \\ -\left(r_{mg} - r\right)\cdot\vartheta^i_f\end{bmatrix}, \tag{9}$$

where:

$$\Delta h^i_{mg} = 0,5\cdot h_g - 0,5\cdot d^i_f - r_{ef} + \\ + \left(1 + 0,5\cdot c^i_g\right)\cdot r_{eg}\cdot\vartheta^i_f - 0,5\cdot c^i_g\cdot r_{ig}\cdot\vartheta^i_f \tag{10}$$

The minimum and maximum values of the load pressure are obtained replacing in the (9) expression the *r* current radius, with $r^*_{i,g}$, respectively with r_{eg}. Given the expression of the $r^*_{i,g}$, size and of the (4) relationship, it results:

$$\left(p_{g\,min}\right)_{r=r^*_{ig}} = \left(E_g/h_g\right)\cdot \begin{bmatrix}\Delta h^i_{mg} - \left(c^i_g\cdot\vartheta^i_f - 0,5\right)\cdot r_{eg} + \\ + \left(0,5 + c^i_g\cdot\vartheta^i_f\right)\cdot r_{ig}\end{bmatrix}; \tag{11}$$

$$\left(p_{g\,max}\right)_{r=r_{eg}} = \left(E_g/h_g\right)\cdot\left(\Delta h^i_{mg} + 0,5\cdot\vartheta^i_f\right). \tag{12}$$

To determine the values of the load pressure of the gasket, in the field between two next bolts, is considered the relationship [1]:

$$p_g = \left(p_g\right)_{rs}\cdot e^{-k_{fg}\cdot r\cdot\omega}\cdot\begin{pmatrix}\cos k_{fg}\cdot r\cdot\omega + \\ + \sin k_{fg}\cdot r\cdot\omega\end{pmatrix}, \tag{13}$$

where,

$$0 \le \omega\; 2\cdot\pi/n,\quad r^*_{ig} \le r \le r_{eg}.$$

The attenuation factor of the load pressure upon circumference, between two consecutive bolts is calculated with the formula [1]:

$$k_{fg} = \frac{1,565}{h_f\cdot\sqrt{1 + \frac{h_g}{h_f}\cdot\frac{l_f}{b}\cdot\frac{E_f}{E_g}}}. \tag{14}$$

Notations

b_i - the initial width of the gasket;

c^i_g - the reduction coefficient of the width of the gasket, at setting, at putting into operating or working;

d^i_f - the openings between the rings of the flanges at setting, at putting into operating or working;

d_g - the diameter of the bolt hole, practiced in the flange ring;

h_a - the maximum height of the asperities of the sealing surfaces;

h_g , h^i_g - the initial thickness of the gasket, respectively the current thickness of it at setting, at putting into operating or working;

h_f - the thickness of the flange ring;

h_r - the height of the rough nesses ruling on the areas for sealing;

k_{fg} - the attenuation factor, calculated by the (14) relationship;

$l_f = r_{ef} - r_{if}$ - the width of the flange ring;

p_g - the load pressure of the gasket;

$(p_g)_r$ - load pressure dependent of the current radius, on the width of the gasket;

$(p_g)_{rs}$ - the load pressure in the plan of the considered bolt;

p_s - the step between the axes of two consecutive bolts;

r - the current radius between the radius of inside and outside of the gasket;

r_{ef}, r_{if} - the outside radius, respectively the inside radius, of the flange ring;

r_{eg}, r_{ig} - the outside radius of the gasket, respectively the inside radius of this;

r^*_{ig} - the inside radius of the gasket, to which its compression it is produced;

r_{ef} , r_{if} - the outside radius of the flange ring, respectively the inside radius of this;

$r_{mg} = 0.5\ .\ (r_{eg} + r_{ig})$ - the middle radius of the gasket, not being deformed;

s - the outside diameter of the bolt head, washer or the contact surface of the nut with the flange ring;

E_f, E_g - Young's of the material of the flange ring, respectively of the gasket;

α_{sf} - the angle under which expands the area of influence of the pressure under the bolt head, washer or nut on the thickness of the flange ring;

ϑ^i_f - the angle of rotation of the flange ring, at setting, at putting into operating or working;

ω - the current angle to center;

Δh^i_{mg} - “the settling” of the gasket under the action of the outside loads (at setting, at putting into operating or working);

Δr^i_g - the compressed width of the gasket at setting, at putting into operating or working.

Conclusions

The sealing of the mechanical structures under pressure is one of technical and scientifical problems - which have concerned and will focus more both the researchers in the field, and the users of such equipment.

The paper, after stating the specific requirements of the flange with ring gasket, indicate the directions in which both the theoretical research and the experimental ones, both in country and abroad are conducted.

In the study the expression of sealing pressure, taking into account both the geometry of the construction and the angle of deformation of the flange ring, on one side next to bolts of constriction, and on the other side in the field between two consecutive bolts, on the base of a factor of influence in the existing literature, are determined.

The obtained results will be used later in the evaluation of sealing force, using theoretical – experimental methods and the respective values will be compared to those given by the technical norms in the field.

References

1.IATAN, I, R., Cercetări teoretice şi experimentale privind construcţiile de îmbinări cu flanşe cu nervuri, Teză de doctorat, Institutul Politehnic din Bucureşti, 1979

2.ADDISON, H., Pipes and Pipelines International, 15, 1970, nr. 3, p. 22

3.KANEVSKI, V. M., Himiceskoe i hefteanoe maşinostroenie, 1978, nr. 1, p. 30

4.RENERT, M., POPA, A., BIT – ISCIR, 1972, nr. 2, p. 116

5.RENERT, M., POPA, A., Probleme de proiectare a îmbinărilor cu flanşe, Comunicare ştiinţifică la Sesiunea de comunicări Institutul de Petrol şi Gaze Ploieşti, 8 -9 octombrie 1971

6.BOIARŞINOV, S.V., Osnovî stroitelnoi mehaniki maşin, Maşinostroeniia, Moskva, 1973

7.JINESCU, V. V., PĂUNESCU, M., Rev. Chim. (Bucuresti), 26, 1975, nr. 7, p. 582

8.PARASCHIVESCU, M., Journal of Pressure Vessel Technology, 1970, febr., p. 1

9.POSEA, M., Studii şi Cercetări de Mecanică Aplicată, 31, 1972, nr. 4, p. 823 -850, 31, 1972, nr. 5, p. 1155 -1172; 32, 1973, nr. 1, p. 131 -146; 34, 1975, nr. 1, p. 77

10.IATAN, I. R., Buletinul Institutului Politehnic Bucureşti, 36, 1974, nr. 1, p. 71

11.IATAN, I. R., RENERT, M., Buletinul Institutului Politehnic Bucureşti, 40, 1978, nr. 7, p. 675

12.JINESCU, V. V., TEODORESCU, N., Rev. Chim. (Bucuresti), 32, nr. 3, 1981, p. 282 32, nr. 4, 1981, p. 385; 33, nr. 7, 1982, p.671

13.KARASEV, P. L., Himiceskoe maşinostroenie, 1961, nr. 3, p. 24

14.WINTER, R., Maschinenbautechnik, 12, 1963, nr. 4, p. 197

15.BÜHNER, H., Chemie Ing. Techn., 45, 1973, nr. 11, p. 811

16.SCHWAIGERER, S., Materialprüfung, 3, 1961, nr. 5, p. 171

17.RENERT, M., Rev. Chim. (Bucuresti), **29**, 1978, nr. 1, p. 85

18.LOK, H. H., Wiss. Zeit., T.H.Otto von Guericke Magdeburg, 9, 1965, nr. 5/6, p. 553

19.FERNLUND, I., Konstruktion, 22, 1970, nr. 6, p. 218

20.JINESCU, V. V., PĂUNESCU, M.,Rev. Chim. (Bucuresti), **41**, nr. 11, 1990, p.898

21.PĂUNESCU, M., RENERT, M., Rev. Chim. (Bucuresti), **43**, nr. 1 - 2, 1992, p. 78; 43, nr. 7, 1992, p. 414

22.PĂUNESCU, M., Rev. Chim. (Bucuresti), **56**, nr. 1, 2005, p. 24

23.PĂUNESCU, M., Construcţia de maşini, 53, nr. 4, 2001, p. 9

24.TACĂ, C. D., PĂUNESCU, M., Buletinul Institutului Politehnic Bucureşti, Seria mecanică, tom L, 1988, p. 91

25.IATAN, I. R., Buletinul Institutului Politehnic Bucureşti, Seria mecanică, tom 53, 1981, nr. 1, p. 101

26.IATAN, I. R., Buletinul Institutului Politehnic Bucureşti, Seria mecanică, tom 55, 1983, nr. 1, p. 50

27.IATAN, I. R., BEZMAN, V., Volumul Simpozionului Naţional de Utilaje pentru construcţii, SINUC 2007, Universitatea Tehnică de Construcţii Bucureşti, p. 49

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