Przemysław Jaszak*

A new solution of the semi-metallic gasket increasing tightness level

https://doi.org/10.1515/eng-2019-0030
Received March 1, 2019; accepted April 8, 2019

Abstract: The paper presents new type of the semi-metallic gasket characterized with elastic core. Thanks to this solution more elastic recovery as well as more uniform contact stress distribution on the gasket’s surface were obtained which consequently caused that tightness level of the flange joint increased by 1000% in contrast to joint gasketed with traditional solution. The tightness characteristic of the traditional as well as new solution of the gaskets were determined experimentally and discussed.

1 Introduction

Semi-metallic gaskets are becoming more and more popular among designers and users of pressure vessels and piping installations. This is due to their many advantages; for instance, they can work at a much higher temperature (up to 1000°C) and in a wide pressure range, i.e. from a vacuum and over pressure up to 4 MPa. The semi-metallic term is derived from the fact that the gasket is made of metal and an associated elastic material. The metal part ensures a high mechanical strength, whereas the elastic one provides high tightness as well as resilience of the gasket.

Depending on the type of a sealed medium, a combination of various types of materials is used to ensure proper chemical and thermal resistance. The metal part is usually made of stainless steel, for instance 1.4301, 1.4306 or 1.4436. The elastic part is made from materials such as PTFE, compressed expanded graphite or expanded mica well known as vermiculite. In [1] the most commonly used (in industrial installations) semi-metallic gaskets were analyzed, specifying their basic advantages and disadvantages. The main advantage is their high resistance to damage against pressure increase and high thermal resistance. Their disadvantages are: a very high sensitivity to changes of the medium parameters (such as: pulsation of pressure and temperature or mechanical vibrations), metallic corrosion and, in some cases, a slight elastic recovery or small tightness at a low contact pressure (assembly). The ways of increasing flexibility and tightness of these gaskets were presented in [2]. The authors assumed that it is possible to increase the operational parameters of these gaskets in case of a correctly selected design feature which, most of all, is the shape of a metallic part. One of the basic design variants of a semi-metallic gasket is a camprofile gasket. The standard construction of such a gasket is presented in Fig. 1. It consists of a metal core with concentric ridges cut on the top faces. Generally, the top faces of the core are coated with an compressed expanded graphite or PTFE. In contrast to traditional soft-material gaskets in the cam-profile gasket there is a local accumulation of the contact stress generated on the ridges. This effect causes a higher density of the elastic material which results in a higher tightness of the joint. On the other hand, the metal core is characterized by a slight elastic recovery as well as excessive stiffness, which leads to a large variation of the contact pressure along the gasket’s width [3]. In this work a new design of the camprofile gasket was presented and analyzed. This solution is characterized by an increased elas-
tic recovery and an even distribution of the contact pressure, which directly influences the increased tightness.

Figure 2: Stress distribution of the flange bolted joint a) perfectly stiffen flanges b) real stress distribution [5]

2 Problem background

During the assembly process of the gasket in the flange joint connection, a gradual growth of the bolts’ tightness leads to a very unfavorable phenomenon called a flange rotation [4]. This effect causes an uneven distribution of the contact stress along the gasket’s width. In a situation where the flanges were perfectly rigid, the contact pressure would be the same over the entire gasket’s width (see Fig. 2a). In fact, when the bolts are tightened, the flanges gradually begin to deform and rotate. As a result of such a deformation, the contact stress on the gasket’s surface is uneven, as shown in Figure 2b. The more rigid the gasket and flange the greater variation of the contact pressure in the gasket’s surface. The uneven distribution of the contact stress results, above all, in a reduction of the required minimum contact stress in the area of the inner diameter of the gasket and a reduction of the effective gasket’s width [6–9]. The effect of the contact stress on a leakage level in a new solution of the metallic gasket was analyzed and discussed in [10]. In addition, due to the flanges rotation, the pressure on the outside gasket diameter may increase excessively and exceed the limit value beyond which the gasket material will be destroyed. As proved in [11], a gasket with a greater elasticity is less vulnerable to the pressure increase which influences a higher residual contact stress and, thus greater tightness. This paper presents a construction of the gasket which prevents the above-mentioned problems and, therefore, increases the tightness level of the flange-bolted joints.

3 Aim and scope of work

The purpose of the work was to analyze the influence of the cross-section shape of the camprofile gasket core on the level of leakage and contact stress distribution in a flange-bolted joint. Three constructions were analyzed, i.e. a standard solution in which the core was characterized by a full rectangular shape and two solutions in which the shape of the core in the cross-section was significantly modified in relation to the standard solution. The modification included in a suitable grooving of the side surfaces of the metal core, thus increasing its resilience. In the first part, experimental tests were carried out. Their aim was to determine the leakage level from the joint gasketed with particular gaskets. In the second part of the study, the influence of the cross-section shape of the core on the distribution of the gasket contact stress in the joint was analyzed.

4 Research object

Fig. 3 presents three variants of gaskets differing in the shape of the cross-section of the metal core. Fig. 3a presents a typical, standard shape of the core, while the solutions shown in Fig. 3b and 3c have a modified shape of the cross-section compared to the standard solution. The solution shown in Fig. 3b is characterized by symmetrical grooves located on the lateral sides of the metal core, while the solution in Fig. 3c has one groove located on the outer wall of the core. The shape and dimensions of the grooves in both modified constructions were identical (see Fig. 3d) and were: cutting depth equal 1/3 of the core width (t = 1/3b), opening angle of the groove α = 20°, diameter of the rounding edge δ = 0.5 mm respectively.

Figure 3: The solutions of the gaskets a) standard gasket, b) gasket with symmetrical grooved core, c) gasket with core grooved on external side, d) the dimension of the groove

Thickness of layers made of the compressed expanded graphite was g = 0.5 mm. In accordance with the problem
described in paragraph 2, the modified core shapes should alleviate and make even the contact stress on the gasket surface, thereby increasing the tightness level. In addition, the circumferential grooves of the core should significantly increase the elasticity of the entire gasket structure. Overall dimensions, i.e. the outer $d_2$ and inner $d_1$ diameters as well as the height $h$ of the core in all three variants based on [12] and were the same and equal $d_2 = 69\, \text{mm}$, $d_1 = 52\, \text{mm}$, $h = 4\, \text{mm}$ respectively.

5 Experimental tests

5.1 Test rig

Fig. 4 presents the test rig where the experimental tests were carried out.

Figure 4: Test rig used for leakage measurement from the flange-bolted joint; 1- helium reservoir, 2- glass cover, 3 – helium detector, 4 – flange bolted joint with inspected gasket, 5 – micrometer sensor

The main element of the test rig was a flange-bolted joint assigned as PN40 DN40 according to [12]. A vacuum helium method was used to measure the leakage level. For this reason, after mounting the gasket between the flanges and tightening the bolts, the whole joint was placed in a sealed glass cover. Measurement of the bolt tightening force was made by means of an indirect method, reading the indication of their elongation by means of a micrometer sensor according to the method presented in [13]. Beforehand, each of the four M16 bolts was subjected to a strength test from which a characteristic describing the value of tightening force as a function of the bolt extension was obtained.

5.2 Experiment program

The test procedure based on the method included in standard [13]. After mounting the gasket on the contact surface of the lower flange, the upper flange was applied and centered. Then, the bolts were placed in the flange holes and nuts were slightly tightened with fingers. The first stage of the research included a gradual increase of the bolts tension in five steps: 5, 10, 20, 30 and 40 kN per bolt respectively. After each increase of the bolt load, a glass cover was applied on the joint. Next, using a vacuum pump (coupled with a helium detector), the air was pumped out of the area between the glass cover and the flange to give a vacuum of 1 mbar. Later, helium with pressure of 9 bar was applied to the inside of the flange, as a result of the vacuum effect the flange internal absolute pressure was 10 bar. Leakage escaping from the joint was measured by means of a helium detector connected to the vacuum chamber. The results in the form of the helium leakage as a function of the bolt tightening level of the joint with given gaskets were shown in Fig. 5. In the second stage of the test, the maximum bolt’s force from 40 kN was gradually reduced up to 30, 20 and 10 kN respectively. After each decrease of the bolt’s tension the joint was placed in a sealed chamber and leakage was measured at the same helium pressure as in the first stage of the measurement. The characteristics of the leakage as a function of the bolt load an unload were shown in Figures 6 to 8.

5.3 Experimental results

Fig. 5 shows leakage of helium from the flange joint with a given variant of the gasket as a function of the bolt load.

Figure 5: Tightness characteristic obtained at loading state

From the analysis of the data presented in the chart it can be seen that at the maximum bolt load, i.e. at
40 kN, the highest tightness (as much as $4 \cdot 10^{-6}$ g/s·m) was obtained in the joint gasketed with the gasket where the core was grooved on the outer side. The unit g/s·m is the mass rate leakage from the joint (here helium) referred to the mean circumference of the gasket. In relation to the standard gasket, tightness of this solution at the bolt load 40 kN increased over one order of magnitude, that is 1000%. In the case of a gasket where the core was symmetrically grooved also a higher tightness was obtained, compared to the standard solution. At the bolt load 40 kN tightness of this solution was $1.1 \cdot 10^{-5}$ g/m·s.

Fig. 6 presents tightness characteristics during the load and unload of the bolt in a joint gasketed with the standard gasket. It is clearly visible that when the bolt was unloaded leakage changes more slowly than when the bolts were loaded in an assembly stage. This behavior of the gasket is a typical phenomenon since the elastic material (here expanded graphite) exhibits some plastic deformation (causes, among others, increase of density of the porous structure of this material), which causes that leakage during the unloading process changes slower than during the assembly load. At the bolt force to 10 kN, leakage was about $1.3 \cdot 10^{-1}$ g/s·m.

The results of the test can be summarized as follows:

- In the joint with a standard gasket at the bolt load of 40 kN leakage equals $7.4 \cdot 10^{-7}$ g/m·s
- The highest tightness among the considered gaskets variants was obtained in the case of a joint gasketed with a gasket where the metallic core was grooved in an external side.
- At the bolt load of 40 kN leakage was $4.1 \cdot 10^{-6}$ g/s·m. In relation to the standard solution this is an increase of 1000%.
- In respect of maintaining the highest leakage value during the unloading process the best solution was...
the gasket where the core was symmetrically grooved. At the bolt unload to 10 kN, leakage was stabilized at $7.3 \times 10^{-4}$ g/s·m. It is an increase of almost one order of magnitude compared to the standard gasket.

6 Analysis of contact stress distribution

In order to determine the distribution of the contact stress on the flange-gasket surface a computer simulation based on the finite element method was applied.

6.1 Computational model

The calculation model was based on DN40 PN40 flange-bolted joint with a gasket. Discretization of individual parts of the joint was performed with Hexa finite elements with a higher-order shape function. The geometric models of the joint and finite element mesh were shown in Fig. 9. The material of metal parts was represented by a bilinear isotropic model with parameters $E = 206000$ MPa, $\nu = 0.3$, $E_u = 8000$ MPa. The material of elastic layers made of a compressed expanded graphite was imitated with a GAS-KET material model [14]. It is a typical model dedicated to the simulation of gasket materials. Graphite compression characteristics obtained from the experimental test constituted the basic parameter of the above mentioned material model (see Fig. 10). On the mating surface of a particular joint element a friction effect was set. In case of a metal to metal contact the friction coefficient was 0.15, whereas on a graphite to metal contact the friction coefficient was 0.3. The simulation was carried out with a 40 kN bolt load and a 10 bar internal joint pressure. The boundary conditions of the joint model were presented in Fig 11. The simulation results were in the form of the contact stress distribution on the top gasket surface.

![Figure 9: Geometric model of the joint and its finite element mesh](image)

![Figure 10: Compression characteristics of compressed expanded graphite with a thickness of 1 mm](image)

![Figure 11: The boundary condition of the computational model](image)

6.2 The numerical results

Fig. 12 shows the distribution of the contact stress of the flange gasketed with the standard gasket. It can be seen that the contact stress along the gasket width is strongly non-uniform. The highest value was 129.4 MPa and occurred on the outer edge of the gasket. The smallest value appeared on the inside edge and was about 60.2 MPa. Fig. 13 shows a map of the contact stress in a gasket where the core was symmetrically grooved. The maximum con-
Contact pressure value in this solution was 135.3 MPa and occurred on the mean gasket diameter. An unfavorable effect in this situation is the fact that the contact stress in the area of the internal diameter dropped drastically compared to the standard solution. The area of the maximum contact stress occupies only 30% of the gasket’s width. A more even distribution of the contact stress along the gasket’s width was obtained in the case where the core was grooved on the outer side – see Fig. 14. It is clearly visible that the zone of maximum pressure (125.4 MPa) has moved to the region of the inner diameter, which is more advantageous from the point of tightness view. The maximum and even contact pressure occupied over 70% of the gasket’s width.

Based on contact stress distributions presented in Fig. 12 to Fig. 14 it was observed that the black area (represents the maximum of the contact stress) can be treated as an effective gasket’s width. The numerical results of the three gasket solution was discussed below:

- Standardized gasket causes a very high irregularity of the contact pressure distribution both in radial and circumferential directions. The minimal pressure value (measured in a radial direction was 60.2 MPa) occurred on the internal diameter of the gasket and increases in the direction of the internal diameter. The black region on the gasket contact stress (see Fig. 12) occupied only 20% of the gasket nominal width. This is a prove that the gasket with standardized metal core is a very vulnerable on flange joint rotation.

- The gasket where the core was symmetrically grooved, guarantees a maximum value of the contact pressure 135.3 MPa but the maximum value was located in mean diameter of the gasket width. The black region represents the effective gasket’s width (see Fig. 13) occupied only 30% of the gasket’s nominal width.

- The gasket where the core was grooved on outside wall guaranteed the most regular contacting pressure. The effective gasket’s width (see Fig. 14) occupied over 70% of the gasket’s nominal width. This is a simple prove why this gasket solution characterizes the increased tightness. Another important fact is that the maximum contact stress value appeared on internal gasket’s diameter. Comparing the stress distribution to
the results obtained in [15] it was found that that unevenness of the stress distribution was reduced over 50%. The mechanism explaining why this gasket provides a better tightness than other solutions was presented in paragraph 7. Explanation of the tightness improvements

7 Explanation of increasing tightness

As concluded in a paragraph 6.2 the better tightness of the gasket with the external groove was achieved thanks to an even more contact pressure distribution on the gasket surface which directly corresponds to larger effective gasket’s width. In order to explain what is an influence of the gasket’s effective width on leakage level, the mechanism of the gas flow via porous ring layer was taken to analysis. The radial flow through the porous layer in the form of ring can be described in accordance with modified Darcy’s expression presented among others in [16, 17]:

\[
Q = \frac{2\pi Kh (p^2_{02} - p^2_{01})}{\eta RT \ln \left( \frac{r_{02}}{r_{01}} \right)}
\]

where:
- \(K\) – permeability,
- \(h\) – gasket thickness,
- \(\eta\) – dynamic viscosity,
- \(R\) – universal gas constant,
- \(T\) – gas temperature,
- \(r_{01}, r_{02}\) – internal and external radius of the ring respectively,
- \(p_{01}, p_{02}\) – internal and external pressure of the ring respectively.

Based on the formula (1) it can be seen that the leakage is proportional to gasket thickness, material permeability and pressure difference, and inversely proportional to thermodynamic gas parameters as well as gasket’s radii which determine the gasket width. To investigate the influence of the gasket’s width on a leakage level only the radii of the ring will be considered. Taking above parameters to consideration, the leakage formula (1) can be reduced to the form:

\[
Q = \frac{1}{\ln \left( \frac{r_{02}}{r_{01}} \right)}
\]

Figure 15: The model of the ring gasket as a porous structure

According in the gasket’s model presented in Fig. 15 it can be assumed that the increment \(x_{01}\) will be caused enlarging of the radius \(r_{01}\), whereas increment \(x_{02}\) will be caused decreasing of radius \(r_{02}\). Depending on the gasket’s radii location, three basic cases will be taken to consideration:

Case 0: Where the radius \(r_{x1}\) and radius \(r_{x2}\) are met in one common point located in among of gasket width area. Additionally radius \(r_{x01}\) is greater than radius \(r_{01}\) and radius \(r_{x2}\) is smaller than radius \(r_{02}\). In such conditions the below relationships are:

\[
r_{x2} < r_{02}; r_{x1} > r_{01} \text{ and } r_{x2}/r_{x1} = 1; \text{ so } \ln(1) = 0 \text{ and it causes that the leakage tends to infinity } Q = \infty, \text{ because the gasket width } b = 0.
\]

Case 1: Where the radius \(r_{01}\) is constant and radius \(r_{x2}\) decreasing but is greater than \(r_{01}\). In such conditions the below relationships are:

\[
r_{01} = \text{const. } r_{x2} > r_{01} \text{ so } r_{x2}/r_{01} > 1 \text{ it means that } \ln(r_{02}/r_{x1}) > 1 \text{ it means that leakage reaches the finite value greater than zero.}
\]

\[
Q^1 = \frac{1}{\ln \left( \frac{r_{02}}{r_{01}} \right)}
\]

Case 2: Where the radius \(r_{01}\) is constant and radius \(r_{x1}\) increasing but is smaller than \(r_{02}\). In such conditions the below relationships are:

\[
r_{02} = \text{const. } r_{x1} < r_{02} \text{ so } r_{02}/r_{x1} > 1 \text{ hence } \ln(r_{02}/r_{x1}) > 1. \text{ In this case the leakage also reaches the finite value greater than zero.}
\]

\[
Q^2 = \frac{1}{\ln \left( \frac{r_{02}}{r_{01} + r_{01}} \right)}
\]

Comparing the equations (3) and (4), the basic question is which situation is better? case_1 - where external radius \(r_{x2}\) decrease or case_2 - where internal radius \(r_{x1}\) increase?
In order to respond to above question the investigation of the both cases $Q^1$ and $Q^2$ in relation to nominal leakage (where the internal and external radius was not changing) was carried out:

$$ \frac{Q_1}{Q_n} = \ln \left( \frac{r_{02}}{r_{01}} \right) \ln \left( \frac{r_{02} - x_{02}}{r_{01} + x_{01}} \right) $$ (5)

and

$$ \frac{Q_2}{Q_n} = \ln \left( \frac{r_{02}}{r_{01}} \right) \ln \left( \frac{r_{02} + x_{02}}{r_{01}} \right) $$ (6)

Putting to equation (5) and (6) the gasket’s nominal dimensions: $r_{01} = 26$ mm, $r_{02} = 34.5$ mm and considering the radius changing in range: $0 \leq x_{01} < b$ for a case 1 and $0 \leq x_{02} < b$ for a case 2, the course of relative leakages can be comparing - see Fig. 16.

Figure 16: The course of the relative leakage in case 1 and case 2

It can be seen that the smaller relative leakage was reached at constant internal radius $r_{01}$ (means in case 1) than in situation where external radius $r_{02}$ was constant. It concluded that the better leakage level was obtained if the radius $r_{02}$ is changing instead situation where the radius $r_{01}$ is changing.

Figure 17: Determination of the gasket’s effective radius, a) standard gasket, b) gasket with symmetric grooves, c) gasket with external groove

On the Fig. 17 the effective internal and external radius of gaskets were determined. Putting those values to the equation (7) the relative leakage values were determined and collected in Table 1.

$$ \frac{Q}{Q_n} = \ln \left( \frac{r_{02}}{r_{01}} \right) \ln \left( \frac{r_{EF,2}}{r_{EF,1}} \right) $$ (7)

Based on Table 1 it was proved that gasket with external groove characterizes one order of magnitude smallest relative leakage against to standard gasket which was experimentally proven. The gasket with groove on both sides revealed three times greater leakage than gasket with external groove which was also confirmed by experimental study.

In this paper only the geometrical aspect as an effective gasket width was taken to the consideration. In order to model a real leakage behavior, such parameter as material permeability (which strongly depends on the internal and surface structures as well as applied gasket stress) should be taken into consideration.

8 Summary of the work

On the basis of leakage tests, it was found that the best solution characterizing the highest tightness was the gasket where the core was grooved on an external side. This solution improved tightness by about 1000% compared to the standard gasket. The gasket solution where the core was grooved on both sides allowed to maintain the highest tightness during the unload process. At the bolt unload to 10 kN, leakage was stabilized at the level of $7.3 \times 10^{-4}$ g/s·m. This increases tightness during the unloading process by one order of magnitude compared to the joint with a standard gasket. Analyses of tightness of this solution showed that a joint can maintain high tightness even with a very large drop of the bolt tension. As a result of the computer simulation, the stress distribution of three variants of the gasket was determined. In the gasket solution where the core had a groove on the outer side, a more even contact stress was provided. In addition, the maximum contact pressure was created on the inside diameter of the gasket which influenced a greater tightness level. In the case of the gasket’s core grooved symmetrically, the contact stress was maximum but it is formed on the mean diameter of the gasket. An unfavorable effect of this solution there was the reduction of the contact stress in the inner diameter of the gasket. The mechanism of the tightness improvement of the gasket where the core was grooved on external side was analyzed and explained by means of Darcy’s flow theory through porous layer.
Table 1: Relative leakage based of effective (width) radii of the particular gasket solution

|                  | Standard gasket | Gasket with external and internal grooves | Gasket with external groove |
|------------------|-----------------|-------------------------------------------|-----------------------------|
| $r_{EF1}$, mm    | 33.8            | 29.5                                      | 26                          |
| $r_{EF2}$, mm    | 34.5            | 31.4                                      | 32.5                        |
| $Q/Q_n$, -       | 13.8            | 4.5                                       | 1.3                         |

Acknowledgement: "Calculations have been carried out using resources provided by Wroclaw Centre for Networking and Supercomputing (http://wcss.pl), grant No. 444"

References

[1] Walczak R., Zagórski A., Environmental and effectiveness based aspects of choosing static sealings, Journal of Machine Construction and Maintenance. Problemy eksploatacji, 2010 No 2. str. 139-148.

[2] Gawliński M., Jaszak P., Możliwość zwiększenia szczelności uszczelnień semimetalowych, Hydraulika i Pneumatyka Nr 4 Lipiec-Sierpień 2015 r.

[3] Estrada H., Analysis of leakage in bolted-flanged joints using contact finite element analysis, Journal of Mechanics Engineering and Automation 5 (2015) 135-142 doi: 10.17265/2159-5275/2015.03.001

[4] Bouzid A., H., Chaaban A., Bazergui A., The influence of the flange rotation on the leakage performance of bolted flanged joints. Proc. CSME Forum Montreal, 1994, Vol. 1, pp. 184-194.

[5] Nagy A., Time dependent characteristics of gaskets AT flange joint, International Journal of Pressure Vessel and Piping No 72, 1997.

[6] Bouzid A., H., Diani M., Derenne M., Determination of gasket effective width based on leakage, PVP-Vol. 478, Analysis of Bolted Joints – 2004 July 25-29, 2004, San Diego, California USA.

[7] Feng X., Gu B. Q., Liu R., Finite element analysis for the metallic gasket effective width, Computational methods in engineering and science EPMESC X, Aug. 21-23, 2006, Sanya, Hainan, China

[8] Żyliński B., Buczkowski R., Analysis of bolt joint using the finite element method, The archive of mechanical engineering, Vol. LVII No 3 2010.

[9] Bouzid A, Derenne M., Distribution of the gasket contact stress in bolted flanged connections. Journal of Pressure Vessel and Piping, 1997; 354: 185-193.

[10] Choiron M. A., Kurata Y., Haruyama S., Kaminishi K., Simulation and experimentation on the contact width of new metal gasket for asbestos substitution, International Journal of Mechanical and Mechatronics Engineering Vol.5, No.3, 2011

[11] Ryś. J., Malar P., Barski M., Analiza numeryczna wpływu sztymności uszczelki na obciążenie śrub w połączeniu kołnierzowym, Czasopismo techniczne Mechanika, Zeszyt 5, Maj 2011, Wydawnictwo Politechniki Krakowskiej,

[12] EN 1514-6: Flanges and their joints – Dimension of gasket for PN – designated flanges – Part 6: Covered serrated metal gasket for use with steel flanges, 2013.

[13] DIN EN 28090-2 - Static gaskets for flange connections - Gaskets made from sheets - Part 2: Special test procedures for quality assurance, 2014.

[14] Walczak R., Pawlicki J., Zagórski A., Tightness and material aspects of bolted flange connections with gaskets of nonlinear properties exposed to variable loads, Arch. Metall. Mater., Vol. 61 (2016), No 3, p. 1409–1416.

[15] Bouzid A., Drenne M., El-Rich M., Birembaut Y., Effect of Flange Rotation and Gasket Width on the Leakage Behavior of Bolted Flange Joints, Welding Research Council bulletin 469, ISSN 0043-2326, 2004.

[16] Zhang Q., Chen X., Huang Y., Zhang X., An experimental study of the leakage mechanism in static seals, Appl. Sci., Volume 8, Issue 8 (August 2018).

[17] Zhang Q., Chen X., Huang Y., Chen Y., Fractal modeling of fluidic leakage through metal sealing surfaces, AIP Advances 8, 045310 (2018); https://doi.org/10.1063/1.5023708