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Algorithmic Approach To The Assessment Automation of The Pipeline Shut-Off Valves Tightness

Aslan A. Tatarkanov[1], Islam A. Alexandrov[2], Maxim S. Mikhailov[3], Alexander N. Muranov[4]

[1,2,3,4] *Institute of Design and Technology Informatics of RAS, Vadkovsky lane 18/1a, Moscow 127055, Russian Federation*

[1]as.tatarkanov@yandex.ru, [2]islam.alexandrov@rambler.ru, [3]maximmihailov@inbox.ru

Abstract - *The article analyses the prospects for applied mathematical and algorithmic support usage to study the sealing capability of contact shut-off valve sealing joints. For ensuring the equipment's operability, it is necessary to determine the required level of sealing forces (contact pressures), on which, among other things, the product weight and size characteristics depend. The research relevance stems from reducing time and material costs at the design phase and the experimental development of pipeline valves. Accordingly, this research aims to develop a methodology for the automated assessment of quality parameters, firstly, the tightness of contact sealing joints to develop proposals for reducing the required level of contact pressures and weight-size characteristics of valves. The paper provides an overview and analytical study of methods for determining surface roughness parameters based on the theory of random functions, discrete, fractal, and discrete-fractal models. In addition, the analysis of the existing theoretical and empirical methods for determining the tightness of contact sealing joints, namely, the reduced (average) gap, porous body permeability, a set of capillaries, percolation, and finite element models, is presented. It is also made clear that the experimental methods. However, unambiguously confirming the operability of certain structures has a limited application in the early stages of the new sealing joint design process since the results depend on specific methods for monitoring tightness used in the products' experimental development. A mathematical apparatus for modeling asperities and determining the sealing characteristics of sealing joints of shut-off valves is further introduced. The main scientific result of the study is the developed algorithm for evaluating these parameters, particular modules of which can be further implemented as software for automating the assessment of the tightness of contact sealing joints at various phases of valves design. In particular, it is planned to create software for the optimal design of the sealing joint of valves with homogeneous probing in the parameter space through LPτ-sequences (Sobol sequence). The developed algorithm allows screening out irrelevant sets of design parameters at the early design phases without the need for their experimental verification, which will result in a decrease in the total time and material resources spent on the development of pipeline valves.*

Keywords - *sealing capability, contact sealing joint, shut-off valve, ball valves, pipe system, mathematical modeling, surface roughness models.*

I. INTRODUCTION

Ensuring the specified indicators of tightness and durability, which in turn is the ability of the structure to maintain tightness for a long time during the active life with established maintenance, is one of the main requirements in the design of pipeline shut-off valves, regardless of the particular industry of their application [1]-[3].

Normally, the formation of leakage through sealing joints depends on many factors, including materials properties of contacting surfaces, modes, and types of technological processing, presence, and properties of coatings, the sealed medium performance, the size and extent of the microchannel, contact forces at the joint, etc. [4]. Thus, the problem of determining the leakage mechanism requires careful consideration of the physical chemistry of surface phenomena and flow through channels with an arbitrary cross-section. In this case, the design process can be considered as defining the basic concept arising from the requirements of the technical specifications and further optimization of the design parameters applying the chosen method of blocking the operating medium flow, the method of sealing, the type of drive, etc.

In this regard, it should also be noted that when designing, often contradictory requirements are imposed on the shut-off valves design, such as: ensuring the minimum weight and size characteristics at a given level of strength and reliability, ensuring the minimum response speed with limited drive power and minimum dynamic loads, the minimum possible cost at providing the desired functionality, durability, aesthetics, etc. [1].

In addition, it is often quite difficult to pose a correct mathematical problem considering the above set of contradictory requirements in practice. Therefore, in a production environment, the design engineer is usually responsible for product development that analyses existing solutions and determines the number of alternative options for which the corresponding calculations are carried out. Moreover, based on the results, a preliminary design of the product is created, which is then considered by the experts' team at the enterprise. Further, it is worked out in more detail at subsequent design stages. Moreover, significant time and material resources are required to verify the operability of the product and its dependence on various technical parameters, including emergency conditions.

In order to reduce these costs, it is worth developing a methodology for the automated assessment of the tightness of contact seals and making a list of proposals to decrease the required level of contact pressure and, accordingly, the product's weight and size characteristics. The research team at the Institute of Design and Technology Informatics of RAS has a significant groundwork in solving quality control automation of products and technological processes in mechanical engineering [5]-[7]. This research aims to study these issues, alongside forming an algorithm for assessing the tightness and contact pressures in the shut-off valves based on the study results.

II. METHODS AND MATERIALS

A. Analysis of methods for assessing the tightness of contact seals of shut-off valves

The microgeometry of the contact surfaces is one of the determining factors affecting such performance indicators of various machine parts: tightness, corrosion resistance, wear and fatigue, contact stiffness, etc. [8], [9]. The deviation of products' entire surfaces from ideally smooth ones is caused by several factors independent of each other. Plastic deformations as a result of the technical processing of rough pieces are one of the most significant.

As a rule, the entire surface roughness of products can be divided according to the size factor into the following levels: macro deviations, waviness, roughness, and sub-micro-roughness. At the same time, to facilitate the task of determining the parameters of the joint tightness, the nanoscale irregularities can be neglected due to their insignificant contribution to the overall leakage rates. Nevertheless, considering only macro deviations and waviness when determining the sealing forces is not enough. The magnitude of the contact stresses required for their complete deformation is much less than the stresses required for complete deformation of asperities at the roughness level. Thus, to determine the sealing ability of the joint, the values of the surface roughness and parameters of the microchannels formed in the joint and the joint density and the relative contact area are required. Consequently, an effective solution to the problem of modeling the topography of rough surfaces of the real products is relevant since it is necessary for the preliminary determination of the products' operational characteristics by describing the process of their contact interaction.

All the variety of existing methods for estimation of the contact sealing joints tightness can be divided into theoretical, which are based on the construction of physical and mathematical models grounded on several assumptions; empirical, respectively based on the experimental identification of the characteristics of the tightness of specific sealing joints designs; and theoretical-empirical, using theoretically grounded dependences, for which the number of coefficients is determined experimentally.

The basis of theoretical methods is the construction of a rough surface model. From this point of view, the following basic models can be distinguished: based on the theory of random functions, discrete, fractal, and discrete-fractal.

The area of model application in a normal random field is the description of a surface after abrasion or finishing using razor tools. Among the model shortcomings, one can single out errors due to the lack of considering the mutual influence of asperities, which is especially important in the case of a high density of contact spots.

A discrete model describing the roughness with the parameters of the initial part of the bearing area curve has become widespread in tribotechnics [10]. However, the application of this model in the case of heavily loaded joints is also limited due to the errors associated with the use of only the initial part of the bearing area. Also, the disadvantages include the method's dependence on the data obtained using measuring instruments, which can vary with a change in the sample length and resolution of the equipment used.

A fractal model based on the use of the Weierstrass-Mandelbrot function, Fourier filtering, and median displacement for modeling surface microgeometry has been extensively used in recent decades [11], [12]. The indisputable advantage of this method is the invariability of the parameters at different scaling levels. However, in this case, some difficulty lies in the non-triviality of the apparatus for determining the fractal dimension. It is necessary to use the method of the normalized range, spectral power, the wavelet transform modulus maxima, or the geometric method. Accordingly, the results obtained may differ in the accuracy of determining the parameters by the methods used.

The fractal-discrete model, in turn, was developed to improve the accuracy of the classical discrete model. As in the classical discrete model, microroughness is approximated in the form of identical spherical segments, which are distributed along with the height depending on the bearing area curve, which, in turn, is described by the fractal model apparatus. This model shows fairly high accuracy and can be applied in design to generate surfaces with specified parameters and study their influence on the tightness characteristics.

In general, the disadvantages of theoretical methods include the complexity of the direct relationship between the initial design parameters and the required contact pressure and a limitation of the accuracy of assessing the product’s real parameters due to the assumptions used. Nevertheless, based on theoretical methods, it is possible to obtain dependences of leakage on the parameters of roughness and micro-gaps of the contact surfaces [13].

On the other hand, even though the results of the empirical methods unambiguously confirm the operability of the products for which the corresponding tests were carried out, their application for other types of structures and the use of the results obtained in the design of new sealing joints structures is very limited. In addition, it is worth considering the dependence of the results on specific methods of tightness control used in the experimental development of products, and the complexity of obtaining the dependences of the influence of various factors on the leakage parameters and contact pressure due to the high cost of the series of tests required for this, as well as, as a consequence of the previous factor, a change in actual results under conditions different from those tested, and, accordingly, the difficulty of predicting the operation of devices in abnormal conditions.

Theoretical-empirical methods for determining the tightness of contact sealing joints include reduced gap, a porous body, a set of capillaries, percolation models, and finite element models of a sealing joint.

The average clearance method represents the sealing surface joint as an equivalent gap. It calculates the leakage using Poiseuille's law, which relates the fluid volume to the pressure drop for the laminar fluid flow in a thin cylindrical pipe. In this case, the problem is reduced to determining the average gap parameters using the values of the contact surfaces, macro-deviations, and waviness. This method leads to errors in assessing the tightness in sealing joints with high contact pressure since, in this case, the tortuosity of the microchannels increases and, accordingly, the introduction of the flow in channels other than purely radial ones into the model is required.

Another approach to determining the leakage depending on the contact pressure value is based on the representation of real rough surfaces by a set of figures of the regular geometric shape. In this case, the task is reduced to determining the shape and geometric dimensions of these figures based on information about the parameters of the contacting surfaces with the further calculation of the joint permeability. Contact pressure is similarly determined with reasonable accuracy. Nevertheless, the method application area is limited to the structures with a high level of contact pressure, since, unlike the average clearance method, the simulation does not take into account the influence of macro-deviations and waviness and, as a rule, it is assumed that the bases of the figures are at the same level. In addition, since the shape of real microroughness can differ considerably from the correct geometric shapes, significant errors occur when determining the geometric parameters of microchannels.

When assessing the joint tightness by determining the porous body's permeability, the Kozeny-Carman equation is used, which connects the parameters of permeability, porosity, tortuosity, and specific surface area. In this case, several assumptions corresponding to a simple model of an ideal porous medium in the form of a bundle of capillaries are used. An additional parameter is introduced, which depends on the properties of the medium and is called the Kozeny-Carman constant. The assumptions made in this model limit the scope of its application, so, for instance, significant errors occur when there are structural cracks. Also, it is worth noting that additional experiments are often required to determine the tortuosity parameter [14]-[16].

The use of percolation models for determining the tightness is considered in the works such as [17] and others. The principle of the method is to determine the contact pressure value, which ensures guaranteed tightness due to the overlap of all microchannels and the formation of a closed-loop at the joint. However, based on this approach, it is impossible to quantify the tightness at lower contact pressures. Thus, except for particularly critical units, the application of the method is very limited. In addition, to carry out calculations, data on the dependence of the contact pressure on the relative contact area are required [18].

One of the first Russian works in determining the tightness of seals in high-pressure fittings is the monograph [19]. The leakage through the clearance between the plug and the seat was defined as the medium flow rate through the radially located capillary tubes. The technique of bringing the surface joint to a set of capillaries when assessing the tightness is also discussed in the works of [20]-[22]. However, when determining the dependence of leakage on contact pressure, these methods may have errors since the calculations do not consider the fraction of dead-end capillaries, isolated volumes, and the distribution of capillary sizes [23].

In the finite element model, the roughness profile is obtained from real products by scanning the surface with profile graphs and profilometers and further mathematical processing of the results applying special software on a data processing machine. In addition, for these purposes, other specialized devices can be used to assess the microtopography of the workpiece's representative surface elements, such as scanning probe microscopes (SPM). The data obtained in this way on the roughness profiles can be used later to calculate the characteristics of the contacting surfaces. Thus, this method can give fairly accurate results with a database of samples and operating environment data. The advantages of this method include the ability to consider both waviness and surface roughness. However, to obtain an evidence-based result, knowledge of the mating surface's topography is required, obtained by experimental measurements [24, 25].

B. Mathematical and algorithmic support for surface roughness modeling

In general, the following functions can be used to describe a rough surface:

$$\varphi_n (u)=n_u/n_c \text{ or } \eta_u (\varepsilon)=A_u/A_c \qquad (1)$$

Where n_c and n_u are, respectively, the total amount of asperities and the number of those whose vertices are higher than u; A_c and A_u are the contour and cross-sectional areas, the latter at a relative level $\varepsilon=h/R_{max}$.

In the discrete roughness model, the initial part of the bearing area curve is described by a parabola. To do this, the following type of functions $\varphi_n(u)=C_n u^x$ are set. However, as practice shows, this approach is not entirely correct.

The bearing area curve is used as a base in the model, since when determining the roughness parameters based on profile-grams, the functions used to describe the distribution for the profile t_p and the surface η_u (ε), coincide, in contrast to the distribution functions for the profile φ_{ni} (u,v) and the surface φ_n (u,v). Approximating a rough surface by a set of spherical segments with a radius r (Fig. 1) and using the assumptions about monotony and twice differentiability of the function $\eta_u(\varepsilon)$, the problem of finding such function $\varphi_n(u)$ can be formulated, so that in the rough layer the distribution of the material matches the bearing area curve.

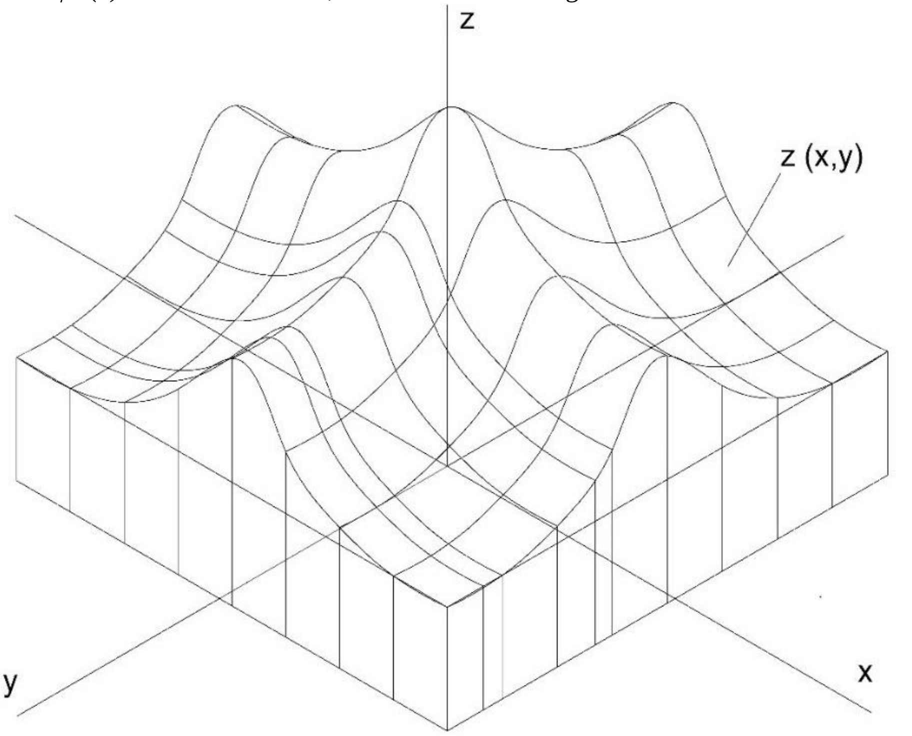

Fig 1: Individual roughness model

The cross-section of the *i-th* roughness at the level ε can be defined as $A_{ri}=2\pi rR_{max}(\varepsilon-u)$, where u is the relative length between the vertices level to the vertex of the *i-th* roughness, and the number of vertices in the layer du and at length u equals $dn_r=n_c\varphi_n'(u)du$.

In this case $A_u=A_r=2\pi rR_{max}n_c\int_0^{\varepsilon}\Phi_n'(u)(\varepsilon-u)du$; $\eta_u(\varepsilon)=(A_r(\varepsilon))/A_c=C\int_0^{\varepsilon}(\varepsilon-u)\varphi_n'(u)du$, where $C=((2\pi rR_{max}n_c)/A_c)$, A_r is the actual contact area.

Further, there is $\eta(\varepsilon)=C(\varepsilon\int_0^{\varepsilon}\Phi_n'(u)du-\int_0^{\varepsilon}u\varphi_n'(u)du)=C(\varepsilon\varphi_n(\varepsilon)-u\varphi^n(u)\,/\,\varepsilon\,¦0++\int_0^{\varepsilon}\Phi_n(u)du)=C\int_0^{\varepsilon}\Phi_n(u)du$.

Differentiating twice the left and the right sides of the resulting expression by ε, there is $\eta'(\varepsilon)=C\varphi_n(\varepsilon)$; $\eta''(\varepsilon)=C\varphi_n'(\varepsilon)$ and correspondingly $\varphi_n(\varepsilon)=(\eta'(\varepsilon)/C$; $\varphi_n'(\varepsilon)=(\eta''(\varepsilon)/C$.

The function of roughnesses distribution will be power-law if the bearing curve is described by the formula:

$t_p=\eta(u)=bu^v$ (2)

and correspondingly $\eta'(u)=bvu^{(v-1)}$, $\varphi_{1n}(u)=(A_cbvu^{(v-1)})/(2\pi rR_{max}n_c)$.

Bearing curve parameters can be related to roughness parameters as follows $v=2t_m(R_p/R_a)-1$, $b=t_m(R_{max}/R_p)^v$.

Designating by the symbol ε_s relative approach at which is $\varphi_n(\varepsilon_s)=1$, and thus $n_u=n_c$, there is:

$n_c=(A_cbv\varepsilon_s^{(v-1)})/(2\pi rR_{max})$, $C=bv\varepsilon_s^{(v-1)}$, $\varphi_{1n}(u)=(u/\varepsilon_s)^{(v-1)}$, $\varphi_{1n}'(u)=((v-1)u^{(v-2)})/(\varepsilon_s^{(v-1)})$ (3)

For the entire bearing curve, in turn, it is rational to apply the incomplete beta function distribution:

$\eta_u(\varepsilon)=I_\varepsilon(\alpha,\beta)=(B_\varepsilon(\alpha,\beta))/B(\alpha,\beta)$, $\eta(\xi)=(B_\xi(\alpha,\beta))/B(\alpha,\beta)$ (4)

where $\alpha=(R_p/R_q)^2((R_{max}-R_p)/R_{max})-R_p/R_{max}$, $\beta=\alpha(R_{max}/R_p-1)$.

Provided that the surface area, the curve radius, and the height of a particular roughness are unchanged, i.e., $\omega=const$, it is possible, with use of the expression for the bearing area curve as a power function, to obtain the following: $\eta'(u)=1/B(\alpha,\beta)u^{(\alpha-1)}(1-u)^{(\beta-1)}$; $\eta(u)=1/B(\alpha,\beta)[u^{(\alpha-2)}(1--u)^{(\beta-2)}((\alpha-1)(1-u)-(\beta-1)u)]$, $\varphi_{2n}(u)=(A_cu^{(\alpha-1)}(1-u)^{(\beta-1)})/(2\pi rR_{max}n_cB(\alpha,\beta))$.

Given that $\varphi_n(\varepsilon_s)=1$, there is:

$n_c=(A_c\varepsilon_s^{(\alpha-1)}(1-\varepsilon_s)^{(\beta-1)})/(2\pi rR_{max}B(\alpha,\beta))$,
$\varphi_{2n}(u)=(\eta_u'(u))/C=(\varepsilon^{(\alpha-1)}(1-\varepsilon)^{(\beta-1)})/(\varepsilon_s^{(\alpha-1)}(1-\varepsilon_s)^{(\beta-1)})$,
$\varphi_{2n}'(u)=(\eta_u''(u))/C=(u^{(\alpha-2)}(1-u)^{(\beta-2)}[(\alpha-1)(1-u)-(\beta-1)u])/(\varepsilon_s^{(\alpha-1)}(1-\varepsilon_s)^{(\beta-1)})$ (5)

In this connection, $\omega=1-\varepsilon_s$.

To link the standard roughness parameters with the parameters of the rough surface model, a single roughness model is used. For this, the function $z(x,y)$ is introduced describing a single roughness mathematically as follows:

$z(x,y)=v+\omega(f(x)+f(y))$ (6)

where $x=2X/S_{xi}$, $y=2Y/S_{xy}$, $z=Z/R_{max}$, $S_{xi}=\sqrt{(A_{ci}K_s)}$, $S_{yi}=\sqrt{(A_{ci}/K_s)}$, $K_s=S_{xi}/S_{yi}$, $f(t)=\{(0.5-t,|t|\leqslant 0.5;\ @(1-|t|)^2,\ 0.5\leqslant|t|\leqslant 1)$.

The above model of a single roughness has a linear distribution of material and height, similar to a spherical segment. With appropriate mathematical apparatus, it is possible to determine the relationship of the parameters of the rough surface model with the standard parameters of the surface roughness.

a) Individual roughness contact To describe the rough surfaces contact, it is necessary to consider the contact model of a separate rigid spherical shape asperity with radius r_i with an elastoplastic half-space in the cylindrical coordinates z, ρ, φ system (Fig. 2) firstly. The asperity vertex is located at a distance uR_{max} from the vertex line of the rough surface and the concave, respectively, at a vR_{max} from the concaves line.

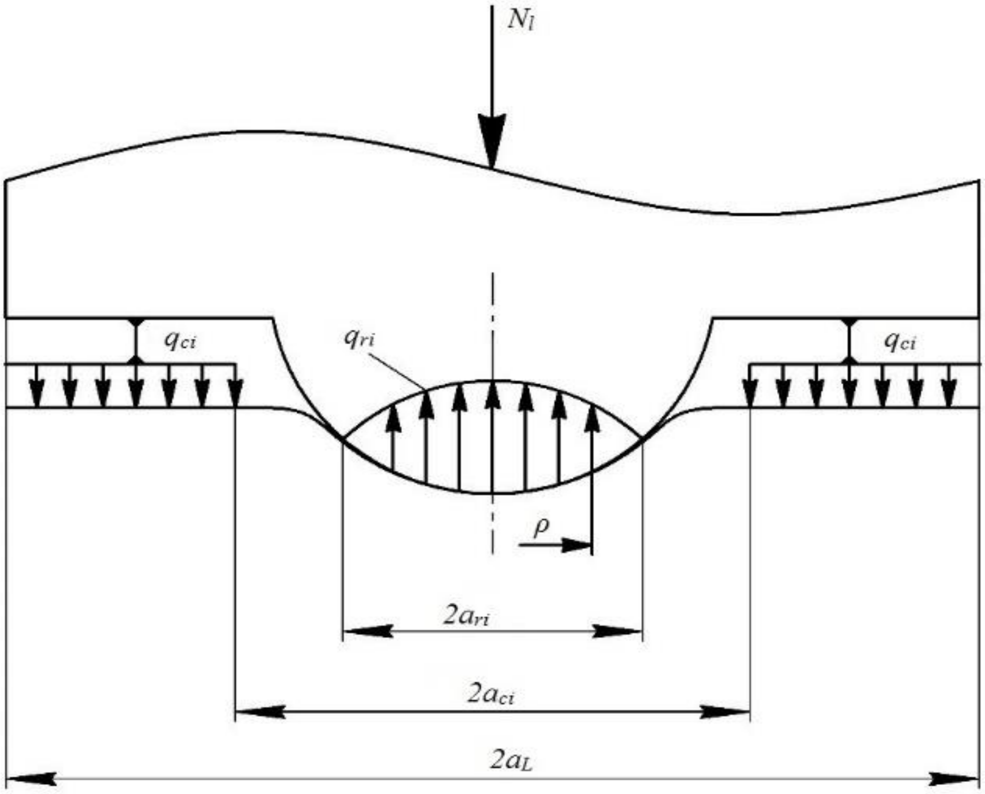

Fig 2: Individual roughness contact diagram

Moreover, assumptions from the mechanics of deformable solids and contact mechanics are used, which have an experimental justification:

1) The radius of the circular contact area of an individual roughness is much less than the radius of the roughness itself (a_{ri} r_i);

2) The displacements of the points of the half-space surface in the radial direction are negligible relative to the displacement in the axial direction;

3) Lack of friction in the contact zone of the half-space roughnesses;

4) Ideally reversible nature of elastic displacements;

5) Stresses and displacements of the half-space surface, which fall on A_{ci}, are described by the relations of the theory of elasticity until the appearance of a plastic imprint at $q_{ci}=(0,q_{cPi})$. In this case, Saint-Venant's principle is also used, according to which stress-strain state at a point far enough from the load application area does not depend on the load shape. Hence it follows that the influence of the stress-strain state of the rest of roughnesses on the size of the contact patch of an individual roughness and the distribution of stresses on the surface is equivalent to a uniformly distributed load q_c, operating in some annular region $a_{ci} \leqslant \rho \leqslant a_L$.

Thus, considering the above assumptions, it is possible to obtain expressions describing the contact of an individual roughness with a smooth half-space, providing for the mutual influence of the rest of the contact roughnesses:

$q_{ri}(\rho_i)=((4\eta_i^{0.5}\omega R_{max})/\pi\Theta a_c)$ ✓ $(1-(\rho_i^2)/(a_{ri}^2))+q_c/\pi$ arccos $(1-\eta_i(2-\rho_i/(a_{ri}^2)))/(1-(\eta_i\rho_i^2)/(a_r^2))$;
$q_{ci}=(8\eta_i$^$1.5\omega R_{max})/3\pi\Theta a_{ci}+q_c\Psi_\eta(\eta_i)$ (7)

Where $\Psi_\eta(\eta_i)=2/\pi[\arcsin \eta_i^{0.5}-\sqrt{}(\eta_i(1-\eta_i))]$; q_{ri} is the pressure distribution within a circular contact area of the *i-th* roughness; $\eta_i=(a_{ri}^2)/(a_{ci}^2)$ is the relative contact area of the *i-th* roughness; ω is the convergence of bodies; Θ is elastic property; $a_c=\sqrt{}(A_c/(\pi n_c))$ is a microgeometry parameter; A_c is the contour area.

Relative contact area of the *i-th* roughness can be expressed as follows:

$\eta_i=(a_{ri}^2)/(a_{ci}^2)=(\varepsilon-u)/2\omega-F_q[(1+1/2F_q)(1-\sqrt{}(1-(\varepsilon-u)/(2\omega(1+0.5F_q)^2)))]$ (8)

where F_q is a complex force elastic geometric parameter.

It is worth noting that the first additive component in expressions (7) and (8) corresponds to the Hertz theory (at $q_c=0$), and the second one gives a correction for the mutual influence of the remaining contact roughnesses. The importance of considering the mutual influence of contact roughnesses is clearly shown in Fig. 3.

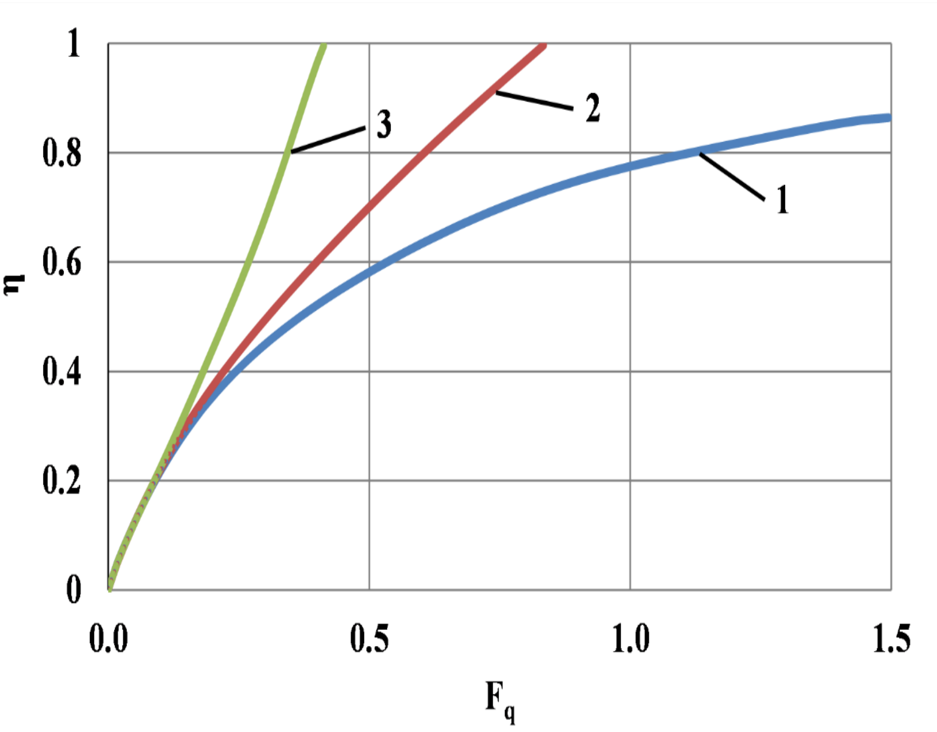

Fig 3: Dependence of the relative area of the contact zone η on the complex parameter F_q: 1 – considering the mutual influence of roughnesses; 2 – excluding mutual influence; 3 – when describing the profile curve by the parabola

Curve 1 in Fig. 3 was obtained using expression (5) considering the mutual influence of roughnesses, curve 2 using (3) and (5) without reference to the mutual influence of roughnesses (graphically, two curves merge into one), and curve 3 using the expression $\eta(\varepsilon)=0.5b\varepsilon^{\nu}$ without reference to the mutual influence of roughnesses. In this case, the following parameters of surface roughness were used $R_{max}=1\mu m$, $R_a=0.2\mu m$, $R_q=0.25\mu m$, $R_p=0.5\ \mu m$, $S=100\mu m$, $S_m=120\mu m$. When describing the roughness by the beta function, it was assumed that $\alpha=\beta=3.5$. When describing the bearing curve by a parabola, it was assumed that $b=2.8989$, $\nu=2.5355$. Thus, for models that do not consider the mutual influence of roughnesses, there is a significant overestimation of the values of the relative contact area.

b) Rough surfaces contact. Further, it is necessary to go from the contact of a particular roughness to the rough surface contact model. In this case, a roughness model is used, for which expressions (3) are valid. The contour pressure in the contact zone between the rough surface and the elastic half-space, as well as the relative contact area, will be determined by the formulas:

$$q_c = \frac{N}{A_c} = \frac{1}{A_c}\sum_{i=1}^{n_r} q_{ci}A_{ci}\,;\ \eta = \frac{A_r}{A_c} = \frac{1}{A_c}\sum_{i=1}^{n_r} A_{ci}\eta_i, \qquad (9)$$

where N is an operating force.

Moreover, for the used roughness model, it is also true that $A_{ci}=const$, $A_c=A_{ci}n_c$, $dn_r=n_{c\ n}'(u)du$.

Thus, expression (9) can be modified as follows:

$$q_c(\varepsilon) = \int_0^{min(\varepsilon,\varepsilon_s)} q_{ci}\varphi_n'(u)du;\ \eta(\varepsilon) = \int_0^{min(\varepsilon,\varepsilon_s)} \eta_i\varphi_n'(u)du \qquad (10)$$

$$F_q(\varepsilon) = \frac{\theta q_c a_c}{\omega R_{max}} = \frac{\frac{8}{3\pi}\int_0^{min(\varepsilon,\varepsilon_s)} \eta_i^{1.5}\varphi_n'(u)du}{1-\int_0^{min(\varepsilon,\varepsilon_s)} \psi_2(\eta_i)\varphi_n'(u)du} \qquad (11)$$

Finally contact of two rough surfaces $Z_i(x,y)$, $i=1,2$ can be considered as a contact of a smooth surface with an equivalent rough surface $Z(x,y)=\Sigma^{2}_{(i=1)} = (x,y)$. In order to determine the parameters of an equivalent rough surface, the initial data are the microgeometry parameters of each of the contacting surfaces $R_{max}i$, R_{qi}, R_{pi}, S_i, S_{mi}, t_{pi}, Δ_{ai}, Δ_{qi}.

When using the model of a rough surface as a stationary random field, according to the theorem on the sum of mean and variance of independent random processes, it is possible to determine the parameters of an equivalent rough surface as the sum of the parameters of the microgeometry of individual contacting surfaces $R_{max}=\sum^{2}_{(i=1)}R_{max}{}_i$, $R_q^2=\sum^{2}_{(i=1)}R_{qi}^2$, $R_p=\sum^{2}_{(i=1)}R_{pi}$, $\Delta_a=\sum^{2}_{(i=1)}\Delta_{ai}$, $\Delta_q^2=\sum^{2}_{(i=1)}\Delta_{qi}^2$.

In turn, to determine equivalent values S and S_m, an expression for the equivalent number of zeros and maxima is used $1/S^2=\sum^{2}_{(i=1)}(R_{qi}/(R_qS_i))^2$, $1/S_m=\sum^{2}_{(i=1)}(R_{qi}/(R_qS_{mi}))^2$.

c) Contacting a rigid rough surface through an elastic layer In many cases in modern pipeline fittings, various coatings are applied to the working surfaces, particularly polymer ones, to improve seals performance. In this case, the sealing ability will depend, among other things, on the coating thickness and material. Most of the existing recommendations for determining the thickness of the coating have an experimental study basis, thus obtained results often can be conflicting. So, the separate consideration of the rough surfaces' contact through the sealing layer is needed.

The works of the Bratsk State University researchers present an engineering technique for estimating the elastic characteristic following the coating thickness based on the stiffness model, the Hertz theory, and the two-point Padé approximation for the extreme values of the coating thickness $\delta_1=0$ and $\delta_1=\infty$.

So for a layered elastic half-space made of the main material with elastic characteristics μ_0 and E_0, coating with thickness δ_1 respectively with elastic characteristics μ_1 and E_1, the elastic feature θ_{01} is determined as:

$$\theta_{01}=\theta_1 F_1 \quad (12)$$

Where:

$\theta_1=(1-\mu_1^2)/E_1$;

$$\theta_1=\frac{1-\mu_1^2}{E_1};\ F_1=\frac{1}{K_1(0)}\left[\frac{\left(K_1(0)-K_1(\bar{\delta}_1)\right)^2}{K_{01}(0)-K_{01}(\bar{\delta}_1)}+K_1(\bar{\delta}_1)\frac{K_0(\bar{\delta}_1)}{K_{01}(\bar{\delta}_1)}\frac{\theta_0}{\theta_{01}}\right];\ \bar{\delta}_1=\frac{\delta_1}{a};\ K_i(\bar{\delta}_1)=K_i(\bar{\delta}_1,\mu_i)$$

is a relative coordinate; a is a radius of the loading area of a half-space with a Hertz load (at $a=a_r$ is the radius of the contact area when a spherical indenter is introduced).

Considering that for the values μ_i ranging from 0.3 to 0.5, function values $K(\bar{\delta}_1,\mu_i)$ do not modify significantly, one can use the following expression:

$$\mu_{01}=\mu_1-(\mu_1-\mu_0)/|1-\theta_0/\theta_1|F_1 \quad (13)$$

When a smooth rigid sphere contacts a layered half-space, the parameters of convergence of bodies, contact radius, and highest pressure value can be determined, respectively, as follows:

$$\omega_{01}=\omega_1 F_1^{\frac{2}{3}},\ a_{01}=a_1 F_1^{\frac{1}{3}},\ p_{01}=p_0 F_1^{\frac{2}{3}} \quad (14)$$

Accordingly, when a rough surface contacts an elastic layered half-space, the parameter $\bar{\delta}_{1i}$ for a separate roughness can be determined as $\bar{\delta}_{1i}=\delta_1/a_r=(\delta_1/a_c)(a_c/a^r=\gamma\eta_i^{(-0.5)}$, where $\gamma=\delta_1/a_c$ is the relative thickness of the coating; $\eta_i=(a_r^2)/(a_c^2)$ is the relative contact area for an individual roughness; a_c is the radius of the site per roughness.

Then, for each contacting asperity, formula (12) has the form:

$$\theta_{01i}(\gamma,\eta_i)=\theta_1 F_{1i}(\gamma,\eta_i) \quad (15)$$

where $F_{1i}(\gamma,\eta_i)$ is determined considering (12) and (13).

Also, taking into account (14) and (15), there is $\eta_i(\varepsilon,u)=\eta_{1i}(\varepsilon,u)F_{1i}^{\frac{2}{3}}(\gamma,\varepsilon,u)$.

As a result, expressions similar to (10) and (11) can be obtained:

$$F_{q1}(\gamma,\varepsilon)=\frac{\frac{8}{3\pi}\int_0^{min(\varepsilon,\varepsilon_s)}\eta_{1i}^{1.5}(\varepsilon,u)\varphi_n'(u)du}{1-\int_0^{min(\varepsilon,\varepsilon_s)}\varphi_\eta(\gamma,\varepsilon,u)\varphi_n'(u)du} \quad (16)$$

$$\eta(\gamma,\varepsilon)=\int_0^{min(\varepsilon,\varepsilon_s)}\eta_{1i}(\varepsilon,u)F_{1i}^{\frac{2}{3}}(\gamma,\varepsilon,u)\varphi_n'(u)du; \quad (17)$$

$$\varphi_\eta(\gamma,\varepsilon,u)=\frac{2}{\pi}\left[arcsin\left(\eta_{1i}^{0.5}(\varepsilon,u)F_{1i}^{\frac{1}{3}}(\gamma,\varepsilon,u)\right)-\sqrt{\eta_{1i}(\varepsilon,u)F_{1i}^{\frac{2}{3}}(\gamma,\varepsilon,u)\cdot\left(1-\eta_{1i}(\varepsilon,u)F_{1i}^{\frac{2}{3}}(\gamma,\varepsilon,u)\right)}\right] \quad (18)$$

For the values of γ and ε, expressions (5), (8), (16), (17), and (18) form a closed system of transcendental equations that make it possible to determine the correlation of the relative contact area η with the value of the complex elastic-geometric parameter F_{q1}. It should also be noted that the relative coating thickness significantly affects the relative contact area when the parameter F_{q1} is constant.

C. Measuring the sealing capacity of sealing joints

a) Joint density when rough surfaces are in contact The task of determining the intercontact space volume of the sealing joint surfaces is reduced to calculating the volume of the gaps between the individual contact and non-contact roughnesses:

$$V_g=\begin{cases}V_{ri}=2\pi\int_{a_{ri}}^{a_c}[z_{10}(\rho)-z_{20}(\rho)]\rho d\rho\\ V_{0i}=2\pi\int_0^{a_{ci}}[z_{ir}(\rho)-z_{2r}(\rho)]\rho d\rho\end{cases} \quad (19)$$

where z_{10} and z_{20} are functions that describe surfaces of the non-contact roughnesses and half-space, z_{ir} and z_{2r} – of the contact roughnesses and half-space, respectively.

Therefore, the total value of the intercontact space volume in the sealing joint can be determined as $V_g=\sum_{i=1}^{n_r}V_{ri}+\sum_{i=1}^{n_c-n_r}V_{0i}$, and the gap density corresponding to a given volume is calculated as follows:

$$\Lambda(\varepsilon)=\frac{V_g}{A_c R_{max}}=\frac{n_c}{A_c R_{max}}=\frac{1}{A_{ci}R_{max}}\left[\int_0^{min(\varepsilon,\varepsilon_s)}V_{ri}\varphi_n'(u)du+\int_{min(\varepsilon,\varepsilon_s)}^{\varepsilon_s}V_{0i}\varphi_n'(u)du\right] \quad (20)$$

Further considering that $V_{ri}=A_{ci}R_{max}\Lambda_{ri}$ and $V_{0i}=A_{ci}R_{max}\Lambda_{0i}$, there is:

$$\Lambda(\varepsilon) = \int_0^{min(\varepsilon,\varepsilon_s)} \Lambda_{ri}\varphi_n'(u)du + \int_{min(\varepsilon,\varepsilon_s)}^{\varepsilon_s} \Lambda_{0i}\varphi_n'(u)du \quad (21)$$

By integrating equations in (19), one can obtain:

$$\Lambda_{0i} = \omega\left[\frac{1}{2} - \frac{\varepsilon-u}{\omega} - 2F_q\left[(k-1) - k_2F_1\left(-\frac{1}{2},\frac{1}{2};2;\frac{1}{k^2}\right) + \frac{8}{3\pi} - 1\right]\right] \quad (22)$$

$$\begin{aligned}\Lambda_{ri} = \omega\Big\{&(1-\eta_i)\left[\frac{1+\eta_i}{2} - \frac{\varepsilon-u}{\omega} - 2F_q(k-1)\right] + 2F_qk_2\left[F_1\left(-\frac{1}{2},\frac{1}{2};2;\frac{1}{k^2}\right) - \eta_iF_1\left(-\frac{1}{2},\frac{1}{2};2;\frac{\eta_i}{k^2}\right)\right]\\ &- 2F_q\left[\frac{8}{3\pi} - \eta_iF_1\left(-\frac{1}{2},\frac{1}{2};2;\eta_i\right)\right] + F_{qi}\left[k_2F_1\left(-\frac{1}{2},\frac{1}{2};\frac{5}{2};\eta_i\right) - \frac{9\pi}{16}\eta_i^{0.5}\right]\Big\}\end{aligned} \quad (23)$$

Formulas (22) and (23) are used to determine the joint density from (20).

When using the equation of roughness and the equation of the half-space deformation from pressure $q_r(\rho)$, there is:

$\Lambda_{0i}=\omega(1/2-(\varepsilon-u)/\omega)$ (24)

$$\Lambda_{ri} = \omega\left[(1-\eta_i)\left(\frac{1+\eta_i}{2} - \frac{\varepsilon-u}{\omega}\right) + \frac{8\eta_i^{1.5}}{3\pi}\left(2_2F_1\left(-\frac{1}{2},\frac{1}{2};\frac{5}{2};\eta_i\right) - \frac{9\pi}{16}\eta_i^{0.5}\right)\right] \quad (25)$$

Excluding the parameter ε from formulas for $\Lambda(\varepsilon)$ and $F_q(\varepsilon)$ relationship $\Lambda(F_q)$ is available.

Fig. 4 shows the change in the joint density with a load increase, considering and excluding the mutual influence of roughnesses. In this case, the relationships obtained using formulas (3) and (5) practically coincide.

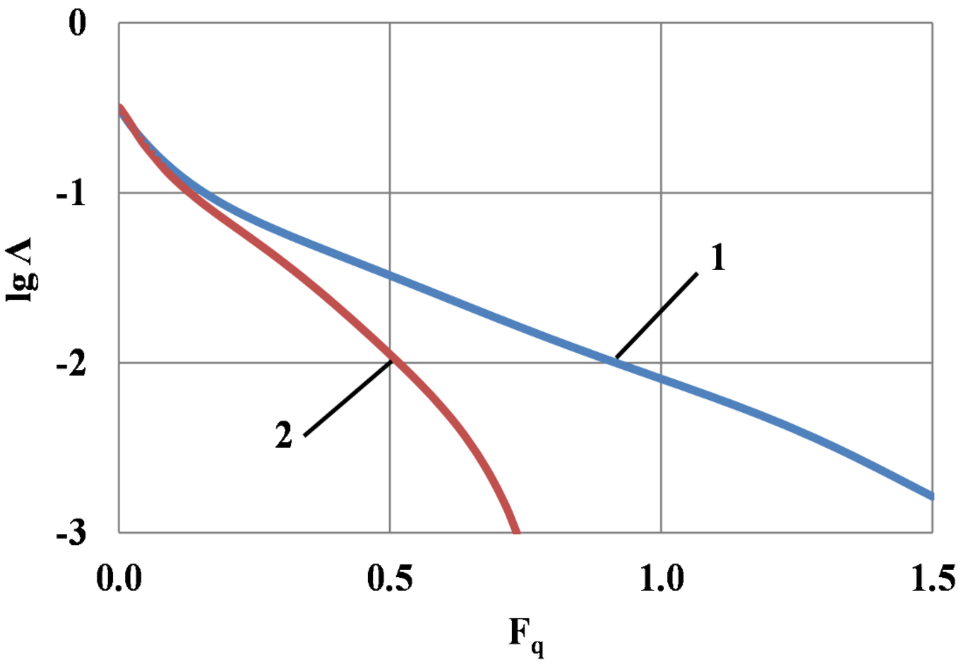

Fig 4: Logarithmic dependence of the joint density with (solid line) and without (dashed line) the mutual influence of roughnesses

b) Mass transfer mechanism for leakage through the sealing joint As shown in the works [26], [27] leakage as mass transfer in capillary-porous media can include the following components:

- mass transfer through microchannels proceeding in molecular, viscous, intermediate, and turbulent modes;
- mass transfer due to migration over the surface of viscous flow in the adsorbed phase;
- mass transfer of a capillary-condensed substance influenced by capillary forces;
- mass transfer due to the penetration of the medium through the elements of the seals or gas release from their surfaces.

The mass transfer mode can be determined depending on the ratio of the characteristic size of microchannels formed in the sealing joint of rough surfaces and the free-path length of molecules in the medium. In the case of pipeline fittings, the viscous transfer mode is most typical. Thus, to estimate the mass rate of the medium flow, it is possible to apply Poiseuille's law mentioned in the first chapter:

$Q=(\pi r^4)/8\eta l\cdot(p_1^2-p_2^2)/2,\ G=(\pi r^4\rho\Delta p)/8\eta l$ (26)

Where r is microchannel section radius, η,ρ is a medium viscosity and density, l is microchannel length, $\Delta p=p_1-p_2$ is the differential pressure.

Also, since the microchannels can have an arbitrary cross-section in the sealing joint, it is possible to introduce the concept of the hydrodynamic radius defined as:

$$r_h^4 = \frac{4F^3}{\pi P^2}, \qquad (27)$$

where F and P are microchannel cross-sectional area and perimeter.

With a discrete roughness model, it is possible to estimate the average area and the microchannel section perimeter per unit length of the profile using data on the relative contact area and the volume of the intercontact space gaps:

$F=V_g/A=R_{max}\Lambda,\ P=(1-\eta)(1+\cos\beta/\cos\beta)$ (28)

Where β is an average angle of inclination of the surface profile. For real angles, the following can be considered:

$P=2(1-\eta)$ (29)

c) Measurement of the fraction of effective microchannels in case of leakage It should be noted that the microchannels have many intersections, and their parameters are determined by the features of the contacting surfaces' roughness, relative contact area, joint density. Accordingly, to determine the fraction of effective microchannels, the joint as a flat percolation model is presented:

$$x_i^* = \begin{cases} 0, \eta_i < \eta_i^* \\ 1, \eta_i > \eta_i^* \end{cases} x^* = \int_0^{\varepsilon_s} x_i^* \varphi_n'(u)\,du, \qquad (30)$$

where x_i* is an auxiliary function, η_i* is the critical value η_i, x* is a fraction of effective microchannels.

Next, there is a need to consider options according to the ratio of the size of an individual roughness S to the percolation model size $l=nS$.

If $l>>S$, the probability of leakage through the lattice of a plane percolation model exposed to the pressure difference is expressed as:

$$v_i = 1 - 2x^{*2} \qquad (31)$$

If l matches with S, a technique is used according to which the probability of leakage is determined from the following expression:

$$v_n = \begin{cases} (1-x^*), x^* < \frac{1}{3} \\ 0.5(1-x^*)(1+3^{n-1}(1-x^*)^{n-1}), \frac{1}{3} \le x^* < \frac{1}{2} \\ 0.5(1-x^*)^n(2^{n-1}+3^{n-1}), x^* \ge \frac{1}{2} \end{cases} \qquad (32)$$

d) Measurement of the leakage rate of contact sealing surfaces The leakage rate through the sealing joint, as mentioned above, depends on the mass transfer mechanism, the microchannel cross-section size, the active medium type and state, the physical chemistry of surface phenomena, and the physics of flow-through channels with an arbitrary cross-section. In this case, the leak can often be viewed simplistically as a gas outflow from a vessel with the pressure p_1 to the medium with pressure p_2 through the resistance channel.

Consider a flat sealing joint with a uniform distribution of contact pressure $(q_c=const)$, the average diameter of the seal area $d_c=r_1+r_2$, and the width of the seal area $l=r_2-r_1$, provided that $d_c>>l$, the diagram of which is shown in Fig. 5.

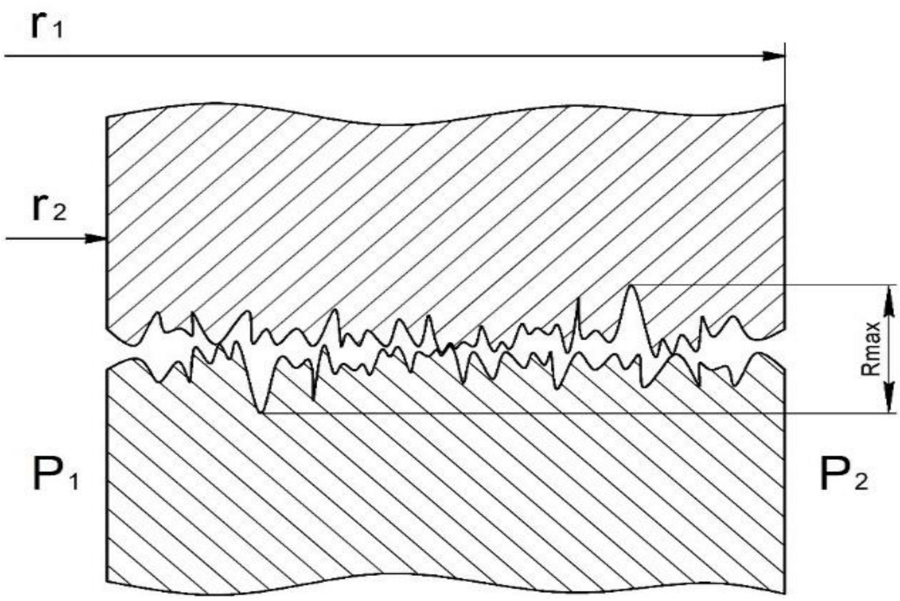

Fig 5: Sealing joint diagram

When modeling a sealing joint of two rough surfaces in the form of a porous body, its height is determined as the sum of each mating surface microroughness heights $R_{max}=R_{max1}+R_{max2}$. The medium flow is constant with a steady flow. The pressure distribution in every plane surface collateral to the pressure gradient is the same, and every plane surface perpendicular to it is inherently ergodic. For obtaining a margin of tightness, the pressure losses at the inlet and outlet can be neglected. Thus, the leakage rate through the porous body is defined as:

$$Q=\sum_{i=1}^{N} Q_i = N\bar{Q}K, \qquad G=\sum_{i=1}^{N} G_i = N\bar{G}K \tag{33}$$

where $N=\pi d_c n \upsilon_i$ is the total amount of effective microchannels; n is a microchannels number per unit length; Q_i, G_i is the leakage value for the i-th microchannel; Q, G is, respectively, mass and volumetric flow rate determined using the functional F and P average values; $K=K_f K_F$ is the coefficient; K_f is the coefficient considering losses caused by structural features and local obstacles (constriction, widening, etc.), K_F is the distribution factor of F and P; in the first approximation, these coefficients can be taken equal to one.

In the case of filtration of a viscous liquid or gas, which is often encountered in practice, for which the assumption of the continuity of the medium $(K_n \rightarrow 0)$ is valid, the transition from a viscous flow regime to a turbulent one is possible only under conditions of unsealing, that is, an emergency state that is not considered at the initial design stage. Therefore, even for high-pressure seals in standard operation, it is advisable to consider the viscous flow of the sealed medium. In this case, the critical value of the Reynolds number corresponding to the transition of flow regimes is determined as $Re=\rho\upsilon d/\mu$ where ρ is the medium density, υ is the filtration rate, d is a characteristic size for which an equivalent hydraulic radius, an average size of microchannels or roughnesses, etc. are suitable.

As a result of substituting the hydraulic diameter as the usual size, $Re=(2G_l)/\mu$ is obtained, where $G_l=G/\pi\, d_c$ is the leakage rate.

Thus, since the viscous regime is considered using Poiseuille's law (26) and the expression for the hydrodynamic radius (27), the expression for the mass flow rate can be obtained:

$$\bar{G}=\frac{\pi\rho\Delta p}{8\eta l}\cdot\frac{4F^3}{\pi P^2 n} \tag{34}$$

where the second multiplier defines the hydraulic radius of the microchannel of an arbitrary cross-section.

Subsequently, the formula (34) for the mass flow rate, together with the previously obtained formula for the number of effective microchannels, is placed into (33), and the leakage is defined as:

$$G=\frac{\pi}{2}\cdot\frac{d_c}{l}\cdot\frac{\rho\Delta p}{\mu}\cdot\frac{F^3\upsilon_i}{P^2} \tag{35}$$

Using formulas for the average values of the area (28) and perimeter (29) of the microchannel cross-section, there is:

$$G_l=\frac{G}{\pi d_c}=\frac{R_{max}^3\rho\Delta p}{2l\mu}\cdot\frac{\Lambda^3\upsilon_i}{4(1-\eta)^2}=C_f C_u \tag{36}$$

Where $C_u=\frac{\Lambda^3\upsilon_i}{4(1-\eta)^2}$ is the dimensionless viscous permeability coefficient, determined by the contact pressure value and parameters of the microgeometry of the joint surface and used in the estimation of the joint sealing ability?

Using the ideal gas model, that is, taking $\rho=p/RT$, $Q_l=G_l(p/\rho)=G_l RT$, where R is gas constant, you can get the following formula:

$$Q_l=C_g C_u \tag{37}$$

Where

$$C_g=\frac{R_{max}^3(p_1^2-p_2^2)}{4l\mu} \tag{38}$$

With increasing pressure, the gas density alters, and the difference between the real and ideal gas models increases. At pressures reaching 10 MPa and above, the presence of internal friction forces in natural gas should be viewed.

Consider the equation of state for natural gas in the varial form:

$$p=\rho RT\left(1+\sum_{i=1}^{r}\sum_{j=0}^{S} B_{ij}\left(\frac{\rho}{\rho_{kp}}\right)^i\left(\frac{T_{kp}}{T}\right)^j\right) \tag{39}$$

Ad hoc, given that $p_1=\Delta p=p$, equation (38) can be written as:

$$C_g=\frac{I_p R_{max}^3}{4l} \tag{40}$$

Where $I_p=\int_{p_0}^{p}\frac{\bar{\rho}(p)pdp}{\mu(p)}$ p_0 is atmospheric pressure, $\bar{\rho}$ is the ratio of the densities of real and ideal gases.

When introduced by analogy with $\bar{\rho}$ the following values $\bar{p}=p/p_0$, $\bar{\mu}=\mu(p)/\mu_0$, where μ_0 is the dynamic viscosity at atmospheric pressure, the equation for I_p is converted as follows:

$$I_p=\frac{p_0^2 I_{\bar{p}}}{\mu_0} \tag{41}$$

Where

$$I_{\bar{p}}=\int_1^{\bar{p}}\frac{\bar{\rho}\bar{p}d\bar{p}}{\bar{\mu}} \tag{42}$$

Then equation (40) can be written as:

$$C_g=\frac{p_0^2 R_{max}^3 I_{\bar{p}}}{4l\mu_0} \tag{43}$$

The relation lg(C_u/C_{u0}) – F_q for the joint of two rough surfaces with similar parameters as in the previously described tasks is shown in Fig. 6. In this case, $\alpha=\beta=3.5$ or $Rqi=0.25$ μm. At $F_q>0.427$, the curve bifurcation is associated with a change υ_n with different widths of the contact zone. With an increase of n parameter value, F_q decreases, and the dependence $lg(C_u/C_{u0}) – F_q$ straightens up. Dependencies $lgC_u – F_q$ at different values R_{qi}:0.233; 0.25 and 0.288 μ mare shown in Fig. 7.

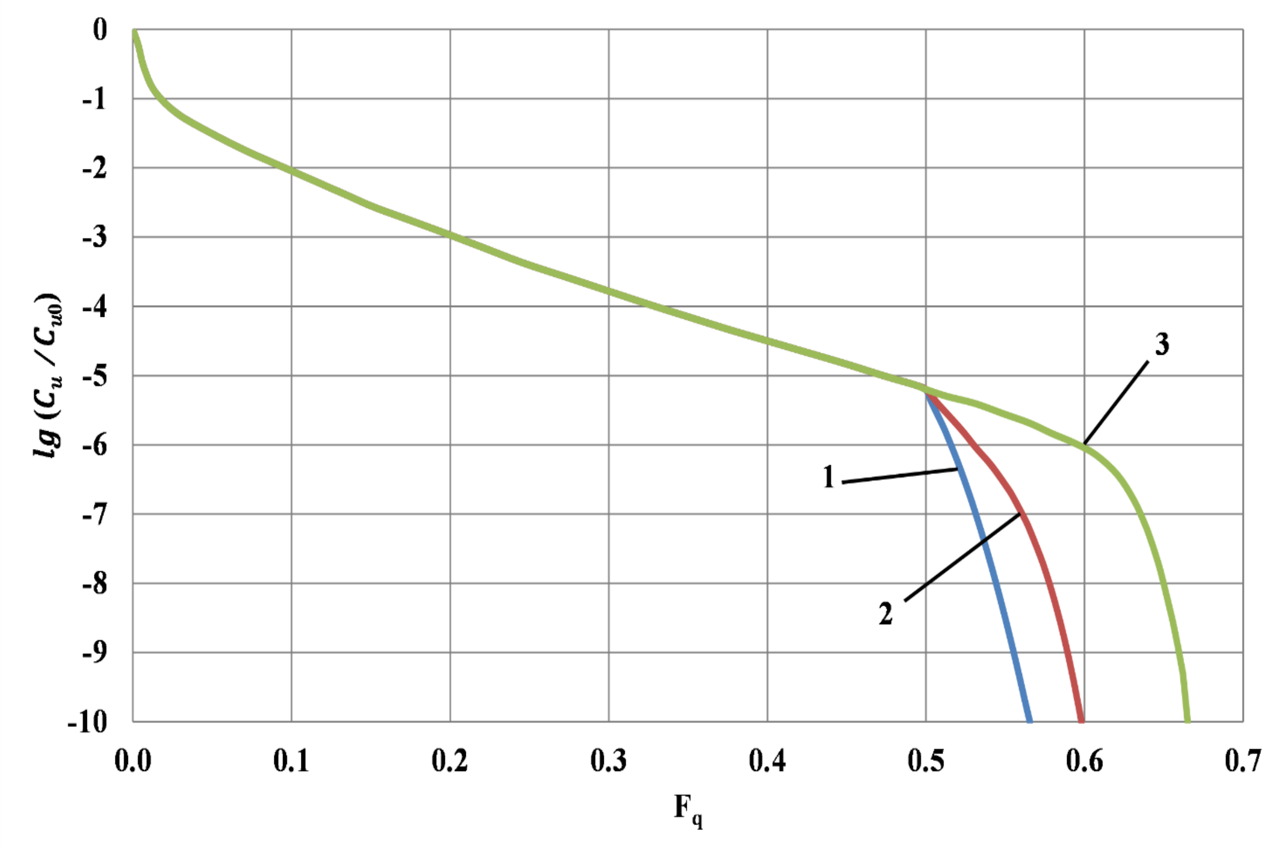

Fig. 6: Dependence of relative permeability *lg(C_u/C_{u0})* from the complex parameter F_q

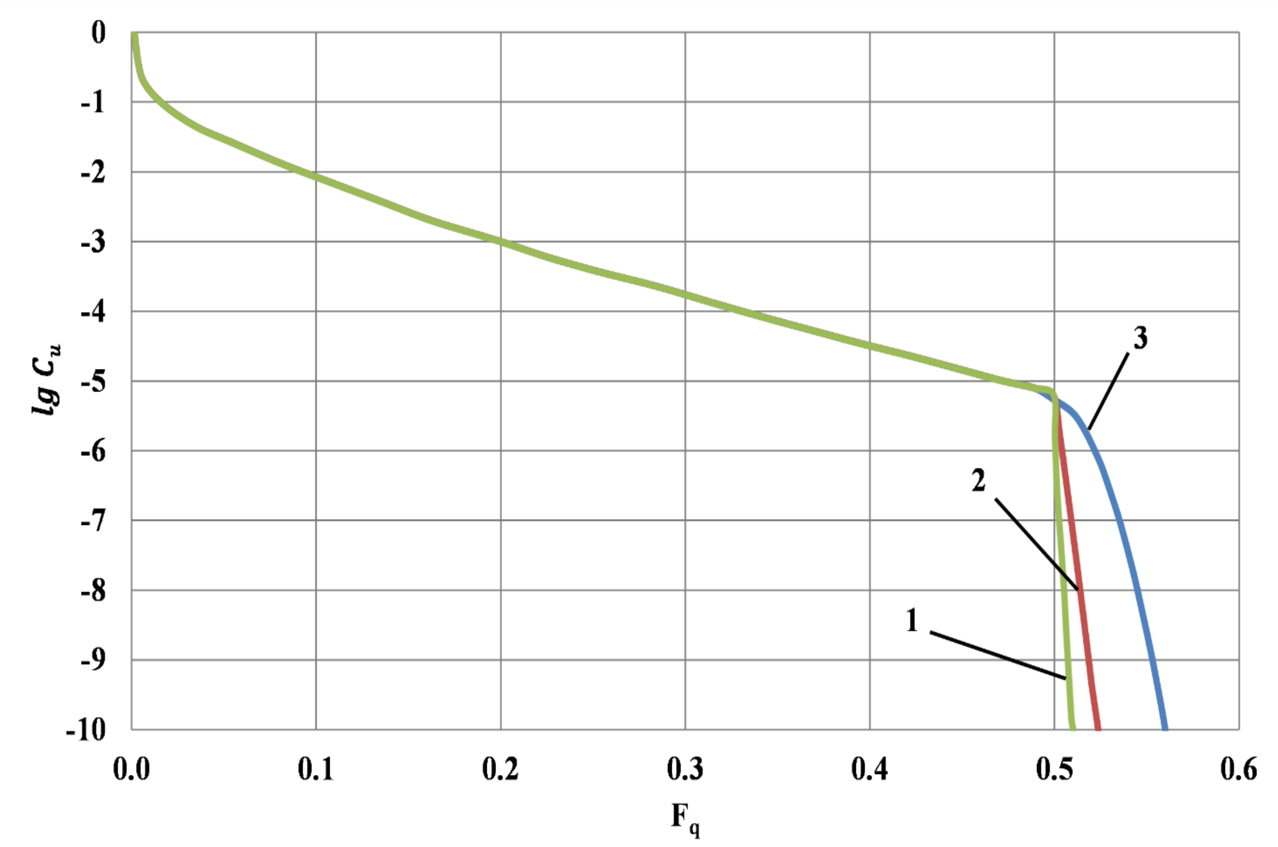

Fig. 7: Dependence of permeability coefficient *lgC_u* from the complex parameter F_q

At n=10, it is possible to approximate dependence lg (C_u/C_{u0}) – F_q for F_q>0.05 using linear dependencies:

$c(F_q)=b_{0i}-k_iF_q$, $i=1;2$ (44)

where $c=lg(C_u/C_{u0})$; $b_{0i}=b(0)$.

Also, analyzing the given dependencies, it can be concluded that there are three sealing zones: initial sealing, when the parameter values F_q range from 0 to 0.06, at which, respectively, the value C_u changes by one order; stable sealing at values F_q from 0.06 to 0.427, C_u changes by two orders of magnitude; and effective sealing at the values of the parameter F_q>0.427, at which a slight change of F_q can lead to a change C_u by several orders of magnitude.

The accuracy in calculations can be increased by using an integral indicator of the following form:

$$K_\lambda = \int_0^{E_s} \frac{\Lambda_i^3(u)\upsilon_{0i}(u)}{4\left(1-\eta_i(u)\right)^2}\varphi_n'(u)du \quad (45)$$

where Λ_i is clearance density, and η_i is relative contact area of an individual roughness; $\varphi_n'(u)$ is the density of roughnesses distribution in height; $\upsilon_{0i}(u)=1$ *if* $\eta_i(u)<\eta_c$, $\upsilon_{0i}(u)=0$ if $\eta_i(u)\geq\eta_c$, where η_c is a critical value $\eta_i(u)$.

By introducing an auxiliary function $\upsilon_{0i}(u)$, it is possible to exclude from the calculation roughnesses, through the small areas A_{ci} of which there is no medium leakage.

The dependence of the integral indicator values K_λ on the complex parameter F_q is shown in Fig. 8. Similar dependencies, determined from the average values Λ and η, give underestimated results by determining F_q, necessary to ensure the specified C_u.

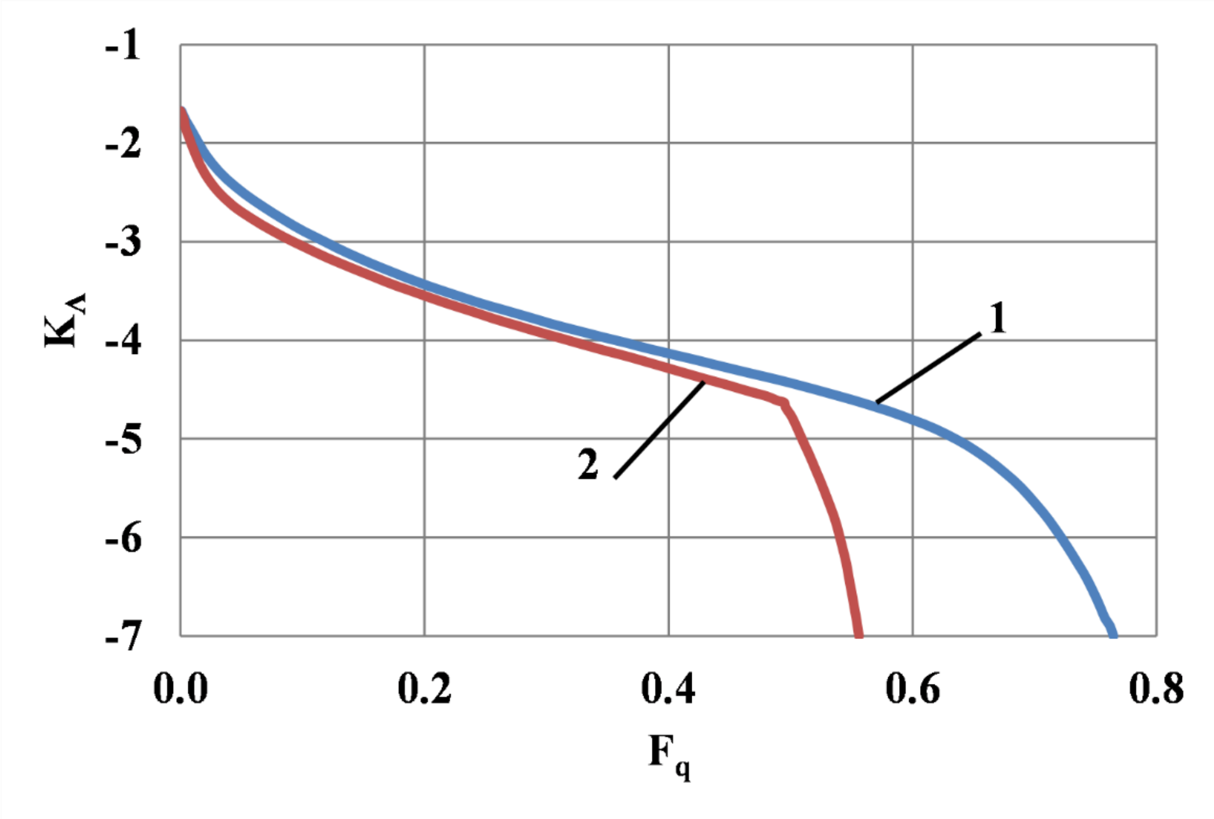

Fig 8: Dependence of the integral indicator value K_λ on the complex parameter F_q

e) Influence of uneven distribution of contact pressures

Thus, the joint sealing ability can be estimated by determining the dimensionless permeability coefficient C_u. At the same time, it is difficult to determine this coefficient's dependence on contact stresses explicitly; it is necessary to use the approximation of the coefficient depending on the complex parameter F_q. For this, it is advisable to use the formula (44).

Considering the case when the distribution of contact stresses over the width of the seal area is described by the function $q=f(x)$, where $x=l/\Lambda$, $l=L_1-L_2$, then $F_q=F_q(x)$, It is assumed that for L_1 the pressure $p_1=p$, for L_2 the pressure p_2=0. Then the equation for estimation of the mass flow rate of the medium will take the form:

$$G = \frac{\pi d_c}{l}\cdot\frac{\rho p}{\mu}C_u R_{max}^3 \quad (46)$$

Considering that $Q=G(p/\rho)$ and the average pressure at the joint $p=\bar{p}/2$, then the following formula can be used to determine the flow rate of the amount of gas:

$$Q = \frac{\pi d_c R_{max}^3 p^2}{4l\mu} \quad (47)$$

Further, given that the permeability $U=Q/p$, there is:

$$U = \frac{\pi d_c R_{max}^3 C_u}{2l\mu}\cdot\frac{p}{2} = \frac{\pi d_c R_{max}^3 C_u}{2l\mu}\bar{p} \quad (48)$$

Then the permeability of the area dl is:

$$U_i = \frac{\pi d_c R_{max}^3 C_u}{2dl\mu} \quad (49)$$

Where $\bar{p_i} = \frac{p_i+(p_i+dp)}{2} \approx$

Medium flow through the area dl:

$Q= dp/dW$ (50)

As $W = \frac{1}{U_i} = \frac{\mu dl}{\pi d_c R_{max}^3 C_u p}$, then, passing to relative coordinates $x=Ll$, there is:

$$\frac{Qldx}{C_u(x)} = \frac{\pi d_c R_{max}^3 pdp}{2\mu} \quad (51)$$

Integrating this expression term by term, considering (44), a formula for determining the leak rate is formed:

$$Q_l = \frac{R_{max}^3 p^2 C_{u0}}{4\mu l \int_{-1}^{1} exp\left(2.3(k_{0l} k_r \theta q_n(X) - c_{0i})\right)} \quad (52)$$

This equation can be applied as a condition for ensuring a given leak rate in the optimal design of valves for pipeline fittings.

III. RESULTS

A. Development of an algorithm for assessing the forces required to ensure a given level of tightness of shut-off valve seals

Based on the analysis of the tightness estimation methods of shut-off valves contact seals and the above mathematical apparatus, an algorithm was developed for assessing the forces required to ensure a given level of tightness of shut-off valve seals (Fig. 9), certain modules of which can be further implemented as computer software. The proposed algorithm includes the following stages. Firstly, analysis of the available technical documentation for the product (technical specifications, design, and technical documentation, product passport, etc.) with the formation of the required product parameters list based on the available data: tightness class, operating conditions, including parameters of the working conditions and environment, design parameters, etc. Further, having a prototype of a structure or an actual product, an experimental measurement of the roughness parameters of the shut-off valves' contact interface is carried out applying a profilograph/profilometer, which is subsequently used to calculate the required tightness and contact pressure parameters. In case there is no prototype, the available data on the geometric parameters of the structure and the planned technological process of product manufacturing, including structural materials, as well as the type and parameters of the tool processing of the working surfaces, that results in the calculation of the required parameters of roughness and the desired parameters of tightness and contact pressures, are used. In the absence of this data, which may be at the early stages of design, the theoretical models discussed in Chapter 1 can be used to calculate the roughness parameters, or reference data can be used on the available analogs of the developed design. For the definition of the contact pressure value required to ensure the specified tightness standards, the dimensionless permeability ratio and the complex force elastic geometric parameter F_q are preliminarily calculated based on the mathematical model presented in Chapters 2 and 3. Then a list of recommendations is formed to reduce the required value of contact pressures by changing the technological process, for instance, the type or parameters of tool processing, the use of coatings, including polymeric, determining the required coating thickness, changes in structural materials or structure parameters with an assessment of the design features values, carried out either by an expert commission of an enterprise or by software with a choice of points in the multidimensional space of design parameters using homogeneously distributed LPτ-sequences (Sobol sequence) providing sufficient uniformity of the study of the parameter space with a relatively small number of tests.

The last algorithm module should be considered in more detail in this context. For defining the optimal design parameters, it is necessary to carry out mathematical modeling of the rough surfaces contacting, its stress-strain state, mass transfer through the joint, etc. The initial design parameters are the required nominal diameter, leakage rates, the working medium parameters (pressure, viscosity, temperature), environmental parameters (pressure and temperature), design requirements, and other demands for the product features. The task is established as the definition of a set of product design parameters. The required values of tightness, strength, and durability are provided due to the minimum contact pressures, which reduces the product's overall weight and size characteristics. For this, a set of values of design parameters is selected in a multidimensional space based on homogeneously distributed LPτ-sequences. When choosing N sample points, each parameter is assigned N different values quasi-uniformly located in the entire range of possible parameter values. Naturally, before this, it is essential to impose restrictions on the possible values of the parameters, determined based on an analysis of the manufacturer's technical requirements and production capabilities. In this case, the problem solvability considering the accepted restrictions is needed to be checked and, if necessary, adjusted the problem statement. The following features are determined for each set of initial parameters: stress-strain state, dimensionless permeability coefficient C_u, and the complex force elastic geometric parameter F_q, etc. The target function is the specific sealing force, which must be minimized. Based on the analysis performed, tables of parameter values are compiled according to the basic requirements. In this case, the ranking of the parameter sets in the tables of values is carried out in terms of the load intensity. Further, for these parameters, quality criteria are calculated in terms of durability and minimum cost. The final decision on the design choice is made by the enterprise employees responsible for the project based on the presented analysis.

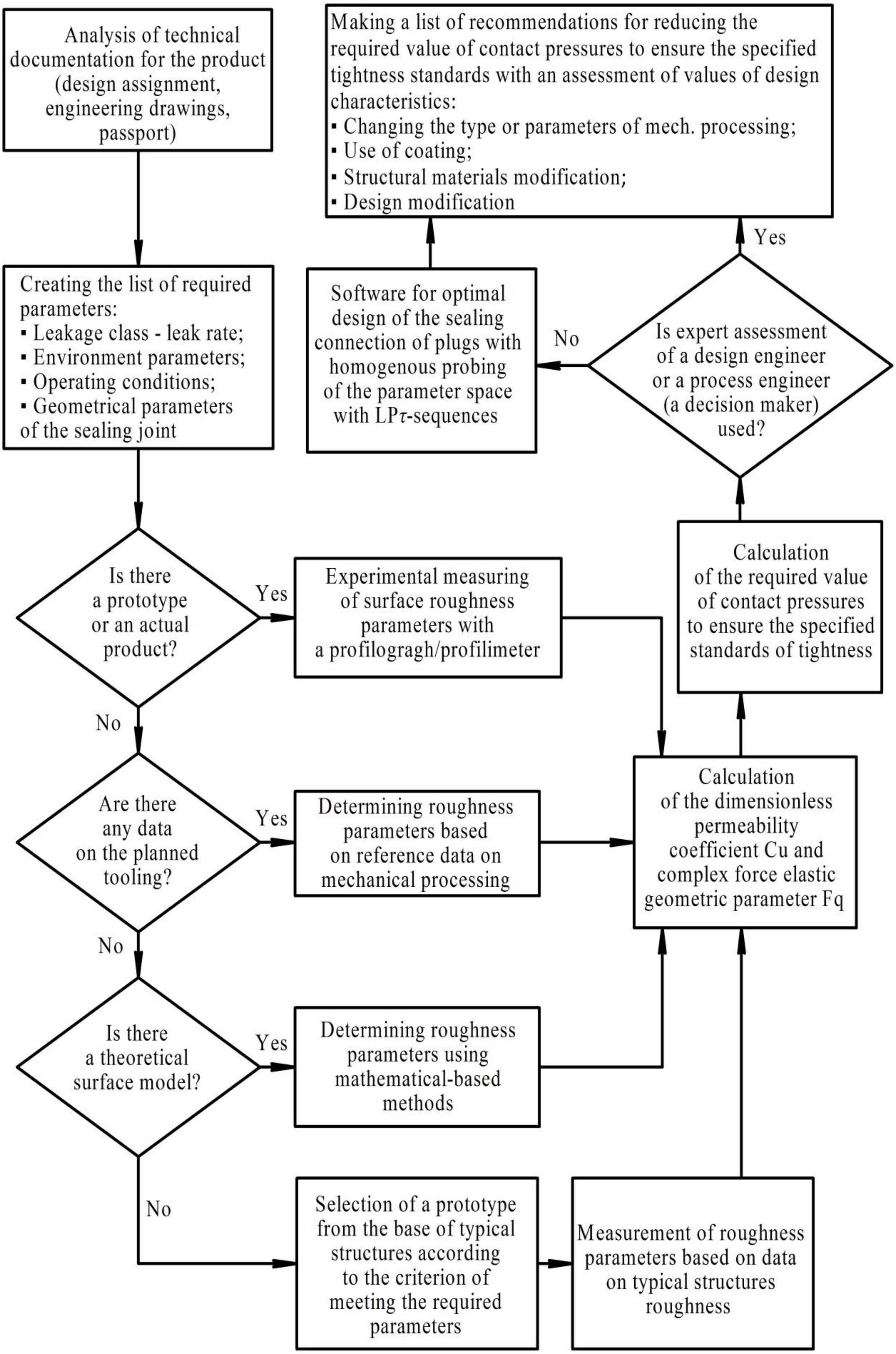

Fig 9: Diagram of the algorithm for assessing the tightness of contact seals of shut-off valves

IV. DISCUSSION

The problem of ensuring the quality of pipeline valves must be considered at the design stage, when there is a possibility of adjusting the design based on multi-criteria analysis, considering the competing demands for the product [28, 29]. At the same time, the priority is the reliability criterion and the set of functional parameters, including the nominal diameter, working pressure, the sealed medium parameters, leakage rates, climatic performance, external influencing factors, and special requirements due to a specific application. Also, it is necessary to introduce safety margins due to loads both in nominal and emergency operating modes [30]. At the same time, in order to minimize the product's weight and size features, it is essential to determine the required level of contact pressures with sufficient accuracy at the design stage to ensure the specified tightness and reliability indicators. Thus, to study the sealing ability of contact seals of shut-off valves, it is advisable to apply for exceptional mathematical and algorithmic support. Due to the development and implementation of specialized software based on the existing groundwork in the field of mathematical modeling of surface roughnesses and finding out the sealing characteristics of shut-off valve seals, it is possible to reduce the time significantly, and material resources cost at the design stage and experimental development of pipeline valves. The algorithm proposed in this study is a systematization of the operations necessary to determine the dimensionless permeability ratio and the corresponding complex elastic geometric parameter at various design stages according to the assumed design parameters, followed by the calculation of the required contact pressures to ensure the required leakage rates and the formation of a list of recommendations for reducing the required contact pressure to ensure the specified tightness standards with an assessment of the values of design features. Later, it is planned to implement certain stages of the algorithm in the form of an automated software package, in particular, to create software for the optimal design of the sealing connection of plugs with homogenous probing of the parameter space with LPτ-sequences. The developed algorithm allows screening out irrelevant sets of design parameters at the early design stages without their experimental verification, which will lead to a decrease in the total cost of time and material resources for the development cycle of pipeline valves.

V. CONCLUSIONS

This work reviews and carries out an analytical study of existing methods for assessing the tightness of contact seals of pipeline shut-off valves. It is stated that the experimental methods, although unambiguously confirming the specific design efficiency, have limited application in the early stages of developing new designs of sealing joints since there is a dependence of results on specific methods of tightness control used in the experimental product adjustment. It is also difficult to obtain dependencies of the various factors that affect the leakage parameters and the required contact pressure due to the high cost of testing and, as a consequence of the previous factor, the change in actual results under conditions different from those tested, and, accordingly, the complexity of predicting the device performance in abnormal conditions. At the same time, theoretical methods focused on modeling a rough surface are limited in the accuracy of estimating the actual parameters of the product due to the assumptions used. In this case, it is difficult to establish a relationship between the initial design parameters and the required contact pressure. However, due to these methods, it is possible to determine the dependence of the leakage value on the roughness parameters and the contact surface micro-gaps. The analysis of models based on the theory of random functions, discrete, fractal, and fractal-discrete, indicating the advantages and disadvantages of these methods, is carried out. The analysis of the existing theoretical-empirical methods, including the adjusted clearance, a porous body, a set of capillaries, percolation models, and finite element models of the sealing joint, is carried out similarly. These methods are applied in mathematical and algorithmic support for modeling surface roughnesses and determining the sealing ability to seal joints. An algorithm for assessing these parameters has been developed, which can be further implemented as computer software to automate the assessment of the tightness of contact seals at various stages of valves design and develop a list of proposals to reduce the required level of contact pressure and, accordingly, the product's weight and size features.

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