International Journal on
"Technical and Physical Problems of Engineering"
(IJTPE)
Published by International Organization of IOTPE

ISSN 2077-3528
IJTPE Journal
www.iotpe.com
ijtpe@iotpe.com

September 2022 | Issue 52 | Volume 14 | Number 3 | Pages 8-13

DURABILITY STUDY OF SPECIALIZED SEALING ELEMENTS

J.N. Aslanov Z.S. Huseynli N.M. Abbasov A.V. Sharifova

Industrial Machines Department, Azerbaijan State Oil and Industry University, Baku, Azerbaijan
tribo72@mail.ru, huseynli_z@rambler.ru, natig.abbasov@asoiu.edu.az, aynur.sh84@mail.ru

Abstract- The significant success has been achieved in the development of the theoretical and applied directions of the elastic object's mechanics. At the same time, the arising problems with the development of modern technology in the oil and chemical industries, requires further important expansion of research, raise a number of new important and difficult problems. Herewith, on the one hand, the improvement and further development of applied methods for solving the main seal problems and, on the other hand, a noticeable expansion of the considered problems class, complete taking into account the specific nature of the various geometries (shapes) of sealing rings and interactions having an importance value for practice, there are still many undeveloped problems, important both in scientific and applied terms. These include: development of practical applied methods for calculating sealing rings, weakened by holes of the oscillation sealing rings under the influence of external pressure, further development of the oscillation's theory and dynamic stability of the sealing rings.

Keywords: Downhole Seal, Round Ring, Oscillation, Elastic Deformation, Elastic-Plastic Deformation, Frequency.

1. INTRODUCTION

In the oil and gas industry, an elastically fixed sealing element (rubber) is often used in oil and gas well equipment for sealing, so the study of the dynamic behavior of this element is of practical interest. Sealants are widely used to connect various types of X-mas tree assemblies and ensure the tightness of assemblies in parts that ensure tightness. Since the main components of the X-mas tree are plug valves, direct-flow valves, adjustable throttles and their flanged connections between metal surfaces by means of sealing joints, the performance of the seals will be included in his performance criterion.

The durability and effective wear resistance of the sealing elements used in the flanged connections of the X-mas tree and its parts and assemblies directly depend on its correct installation in the socket. That is, the geometrical dimensions of the seat of the seal must be designed in accordance with its own geometrical dimensions. One of the important issues here is to ensure the absence of "leaks" in the area of the hermetic connection in accordance with the requirements for seals, ensuring operational reliability and durability. By choosing the seal design in accordance with the design, operating parameters and required properties of the housing in which it is installed, its performance can be increased many times over. To select the structural parameters of the seal in accordance with the required properties, it is necessary to study its deformation state and adapt the results obtained to the required properties within the criteria.

Making the necessary design decisions to improve the efficiency of X-mas' trees and connecting structures used during their installation. This will increase their visibility and make them easier to use. Valves in the process of designing in this direction and one of the important issues is to ensure that the cranes are properly supplied with materials in accordance with their working conditions. Behind the metal-metal surface in the installed grooves of the key parts of the connecting structures, between the joints and at the ends sealing is carried out directly with elastomer sealants. The seals are mainly made of rubber material. Has various designs with preparation. Connecting structures are mainly round, rectangular, trapezoidal, triangular and Y-shaped sealing rings are used. Round cone and Y seals are the most commonly used seals (Figures 1 and 2).

These seals are used to create a complete tightness between metal surfaces, as well as behind and in the groove of threaded connections developed. The most popular sealant, widely used in the sealing of assemblies and the construction of machines and equipment, is a round section rings. API O-Ring Materials: Nitrile (NBR or Bugun-N), Silicone (VMQ), Fluorocarbon (Viton, FKM), Perfluoro elastomer (Kalrez, FFKM), Fluor silicon (FVMQ), Ethylene Propylene (EPM, EPDM, EP, EPR), Neoprene (CR, Chloroprene), Polyurethane (AU, EU). Round cone seals are standardized according to specifications.

Figure 1. Wear marks on the rubber material

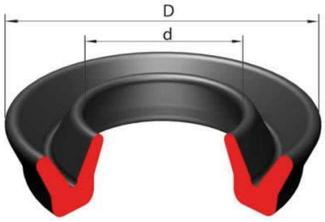

Figure 2. Seal of rubber material

The main task in sealing systems is to determine the criteria, which is the selection of the parameters of sealing parts that perform the required technical functions. Two or more contact zones of the equipment, sealed with elastic materials (seals), must be provided with such contact stresses or deformations to compensate for the tightness of the connection. To create the required tightness according to the required characteristics (pressure, temperature, speed, etc.), such a level of deformation must be created in the elastic sealing material in this contact zone so that the contact stress is maintained at the required level at this moment and during the working process. Here, the first task is elastic reporting, and the second is the study of the stress relaxation process. To this end, the main direction of research is to study the functionality of the structural potentials of the seals that meet the requirements, and to determine the exact stress values in the seals, which play the role of intermediate seals.

The significant success has been achieved in the development of the theoretical and applied directions of the elastic object's mechanics. At the same time, the arising problems with the development of modern technology in the oil and chemical industries, requires further important expansion of research, raise a number of new important and difficult problems [16, 17]. Herewith, on the one hand, the improvement and further development of applied methods for solving the main seal problems and, on the other hand, a noticeable expansion of the considered problems class, complete taking into account the specific nature of the various geometries (shapes) of sealing rings and interactions having an importance value for practice, there are still many undeveloped problems, important both in scientific and applied terms. These include: development of practical applied methods for calculating sealing rings, weakened by holes of the oscillation sealing rings under the influence of external pressure, further development of the oscillation's theory and dynamic stability of the sealing rings.

One of the issues is to ensure maximum tightness of elastic elements in the nest. The most general case of elastic fastening can be reflected with help of boundary conditions:

$$-M = \alpha_2 W + \alpha_{22}\frac{\partial W}{\partial r} \qquad (1)$$

where, F and M are accordingly the shearing force and the bending moment at the edge of the sealing element; W and $-\frac{\partial W}{\partial r}$ are deflection (deformation) and angle of the edge rotation; and $a_{i,j}$ is coefficient of fastening elasticity.

In the axisymmetric case, which we will research for F_2 great concreteness of the obtained results, we have [1]:

$$F = -D\frac{\partial}{\partial r}\cdot\nabla^2 W;\ M = -D\left(\frac{\partial^2}{\partial r^2}+\frac{\partial}{r}\cdot\frac{\partial}{\partial r}\right) \qquad (2)$$

the equation of the sealing element motion under the action of an external pressure $P_0\cdot l^{-i\omega t}$ in the form [1]:

$$D\nabla^2\nabla^2\cdot W - Ph\omega^2\cdot w = P_0 \qquad (3)$$

2. METHODS FOR SOLVING THE STATED PROBLEM

We will write in the sum form of a particular solution of the inhomogeneous equation and the general solution of the homogeneous equation [1, 5]:

$$W = -\frac{P_0}{\rho h\omega^2 W} = P_0 \qquad (4)$$

where, k is the wavenumber of bending waves, defined by [2, 4, 6].

$$k = \sqrt[4]{\rho h\omega^2 / D} \qquad (5)$$

Taking into account that the Bessel functions $J_0(kr)$ and $I_0(kr)$ are the eigen functions of the operators:

$$\left(\nabla_r^2 + k^2\right)\cdot J_0(kr) = 0,\ \left(\nabla^2 - k^2\right)\cdot I_0(kr) = 0 \qquad (6)$$

We write the values of F and M in the form:

$$\begin{aligned} &F = ADk^3 I_0(kr) - BDk^3\left(I_0(kr)\right)\\ &M = D\,k^2 A\left[J_0(kr) + \frac{1-\nu}{kr}\cdot J_0(kr)\right] +\\ &+Dk^2 B\left[-I_0(kr) + \frac{1-\nu}{kr}\cdot I_0(kr)\right] \end{aligned} \qquad (7)$$

Using the boundary conditions (1), we obtain the following system of algebraic Equation:

$$\begin{aligned} &A\left\{\alpha_{11}J_0(kR) + I_0(kR)\left[Dk^3 + \alpha_{12}k\right]\right\} +\\ &+BA\left\{\alpha_{11}J_0(kR) + I(kR)\left[-Dk^3 + \alpha_{12}k\right]\right\} =\\ &= \frac{P_0\alpha_{21}}{\rho h\omega^2} \end{aligned} \qquad (8)$$

Thus, the solution of the problem is obtained in closed form without any expansions in eigen functions:

$$W(r,t) = \frac{P_0 e^{-iat}}{\rho h\omega^2}\cdot\left\{-1 + \frac{\Delta_1 J_0(kR) + \Delta_2 I_0(kR)}{\Delta}\right\}$$

where,

$$\begin{aligned} &\Delta_1 =\\ &= \left|\begin{matrix} \alpha_{11}\left\{\alpha_{11}J_0(kR) + I_0\left(kR\left[Dk^3 + \alpha_{21}k\right]\right)\right\} \\ \alpha_{22}\left\{J_0(kR)\left[\alpha_{21} - Dk\ \right] + I(kR)\left[\alpha_{22}k\right] + Dk^2\frac{1-\gamma}{kR}\right\} \end{matrix}\right|;\\ &\Delta_2 =\\ &= \left|\begin{matrix} \left\{\alpha_{11}J_0(kR)\left[Dk^3 + \alpha_{21}k\right]\right\}\alpha_{11} \\ \left\{\left[\alpha_{21} + Dk^2\right] - J_0(kR) + I_0 kR\left[\alpha_{22}k + Dk^2\frac{1-\gamma}{kR}\right]\right\}\alpha_{21} \end{matrix}\right| \end{aligned} \qquad (9)$$

Now we have the opportunity to study various special cases of elastic fastening of the circular sealing element edge, which are most often found in engineering practice

$$\alpha_{12} = \alpha_{21} = 0;\ \ \alpha_{22} = \infty;\ \ 0 < \alpha_{11} < \infty \lim_{\delta x \to 0} \tag{10}$$

It is easy to see that the compliance of the support in this case is connected with the shearing forces. Another possible case of elastic pinching, which we will call the second type of fixation, is expressed by the conditions:

$$\alpha_{12} = \alpha_{21} = 0;\ \ \alpha_{11} = \infty;\ \ 0 < \alpha_{22} < \infty \tag{11}$$

In this case, the compliance of the supports is related to the bending moment. The stiffness coefficients of the sealing element can be expressed through their mechanical and geometric parameters in a known manner [15]:

First, we consider an elastic fastening of the first kind. For $\alpha_{12} = \alpha_{21} = 0$ and $\alpha_{22} = \infty$ the solution can be represented in the form;

$$W(r,t) = \frac{P_0 e^{-i\omega t}}{\rho h \omega^2} \times \left\{ -1 + \frac{I_0(kR) J_0(kR) - I_0(kR) J_0(kR)}{I_0(kR) J_0(kR) - I_0(kR) J_0(kR) + \frac{2Dk^2}{\alpha_{11}} I_0 J_0} \right\} \tag{12}$$

For strength calculations the value of the reaction force, arising in elastic fastening introduces a special interest. In accordance with the boundary conditions, this quantity is,

$$F = -\alpha_{11} W(R,t) \tag{13}$$

After the transformations, we obtain

$$F = -\frac{P_0 \alpha_{11} e^{-i\omega t}}{\rho h \omega^2} \times \left\{ -1 + \frac{\frac{I_0(kR)}{I_0(kR)} - \frac{J_0(kR)}{I_0(kR)}}{\frac{J_0(kR)}{I_0(kR)} - \frac{J_0(kR)}{I_0(kR)} + \frac{2Dk^3}{\alpha}} \right\} \tag{14}$$

This ratio can be written in a dimensionless form

$$f(\Omega) = \frac{ED}{\beta \left[\frac{I_0(\Omega)}{J_0(\Omega)} - \frac{I_0(\Omega)}{J_0(\Omega)} \right] \cdot \Omega + \Omega^4} \tag{15}$$

where, $f(\Omega) = \frac{ED}{P_0 \alpha_{11} R^4}$ is a dimensionless quantity characterizing the strength of the reactions; $\Omega = kR$ is dimensionless excitation frequency and $\beta = \frac{\alpha_{11} R^3}{2D}$ is dimensionless stiffness parameter of the support.

3. ANALYSIS OF RESULTS

A particular dependence of the quantity modulus $f(\Omega)$ gives a resonance curve; reverse value $f(\Omega)$ has a transmitting character in strength.

We explore in detail the frequency dependence of $f(\Omega)$ for various parameters characterizing their fixation rigidity. An important feature of this type of boundary conditions is the appearance of a low-frequency resonance, caused by the compliance of the support. However, as the rigidity of the fastening increases, this resonant frequency increases, reaching, at $\beta \to \infty$ the lower frequency of the clamped disk. The physical appearance of low-frequency resonance can be explained by the existence of a special oscillation shape at which the deformations of the disk are small, and it oscillates almost like a rigid mass, and the elasticity of the system is due to the compliance of the sealing element. If the support is sufficiently soft, then for low frequencies the expansion.

$$\frac{J_0(\Omega)}{I_0(\Omega)} - \frac{J_0(\Omega)}{I_0(\Omega)} = -\frac{2}{\Omega} - \frac{2}{\Omega} = -\frac{4}{\Omega} \tag{16}$$

This implies that the dimensional record of the reaction force will have the form

$$F = \frac{2P_0 R\, e^{-i\omega t}}{\frac{2\rho h R \omega^2}{\alpha_{11}}} \tag{17}$$

This formula corresponds to the elastic oscillations of a solid disk on a sufficiently compliant spring, which is confirmed by previous arguments. Finally, we`ll analyze the overall frequency characteristics of the system. It is convenient to consider the case of small $\beta(\beta \approx 1)$ and the case of large $\beta \geq 1$ those the case of small and high stiffness of the seal. It is interesting to note here that the resonances alternate with the anti-resonances whose position does not depend on the rigidity parameter β Indeed, the denominator turns to infinity only at frequency values, when,

$$I_0(\Omega) = 0;\ \Omega = \mu_k;\ k = 1, 2, 3...$$

At these values, the frequencies $f(\Omega) = 0$ and are anti-resonances. The values of the dimensionless frequency μ_k at which $F = 0$ are given;

Table 1. The values of the dimensionless frequency μ_K [14, 15]

K	1	2	3	4	5	6	7	8	9
μ_k	3.832	7.016	10.17	13.32	16.47	19.61	22.76	25.90	29.05

Resonances of the system are determined by a complex transcendental Equation.

$$\frac{J_0(\Omega)}{I_0(\Omega)} - \frac{J_0(\Omega)}{I_0(\Omega)} = -\frac{\Omega^2}{\beta} \tag{18}$$

The solution of this Equation depends essentially on the parameter $\Omega\beta$. If β is small, then the first resonance is very low-frequency and can be found by expanding the Bessel functions into series. Thus, we obtain

$$\Omega^* = \sqrt[4]{4\beta} \tag{19}$$

The remaining roots of this Equation can also be found, since at small β the resonances are very close to the anti-resonances and can be found by the formula:

$\Omega_k = \mu_k + \delta_k$

where, $\delta_k / \mu_k << 1$; μ_k, the value of the anti-resonances given in the table. From the approximate formulas and from the correct calculation it is seen that the natural frequencies increase asymptotically approaching the corresponding frequencies of the clamped disk. Another practically possible variant of the elastic fastening of the edge is the fixation, when the boundary conditions are satisfied.

$$-M = \alpha_{22} W_r'; \ \alpha_{22} < 0 \tag{20}$$

It can be obtained from the general case by passage to the limit, taking

$$\alpha_{12} = \alpha_{21} = 0; \ T\alpha_1 = \infty; \ 0 < \alpha_{22} < \infty \tag{21}$$

and substituting in (1)

$$W\left(r,t = \frac{P_0 e^{-i\omega t}}{\rho h \omega^2}\right) \times \left\{-1 + \frac{\Delta_1 V_0(kR) + \Delta_2 J_0(kR)}{I_0(kR) J_0(kR) - I_0(kR) J_0(kR) + \frac{2Dk^2}{\alpha_{11}} \cdot I_0 J}\right\} \tag{22}$$

where,

$$\Delta_1 = I_0(kR) \cdot \left(\alpha_{22} + \frac{D(1-\eta)}{R}\right) k - Dk_2 J_0(kR)$$
$$\Delta_2 = -J_0(kR)\left(\alpha_{22} + \frac{D(1-\eta)}{R}\right) k - Dk^2 J_0(kR) \tag{23}$$
$$\Delta_3 = \left[J_0(kR) I_0(kR) - I_0(kR) - J_0(kR)\right] \times \left[\alpha_{22} + \frac{D(1-\mu)}{R}\right] k - J_0(kR) I_0(kR) \cdot 2Dk^2$$

where, the characteristic value of the determining reactive bending moment in the pinch is the angle of rotation of the section in the seal. This value is dimensionless, therefore, the dimensionless parameters introduced will be:

$$r = R; \ \frac{\partial W}{\partial r} = \phi(\Omega); \ \Omega = kR \tag{24}$$

$$\gamma = -\frac{\alpha_{22} R}{2D} + \frac{1-\gamma}{2}; \ \phi_0 = \frac{P_0 R^3}{2D}$$

Thus, we obtain

$$\phi(\Omega) = \frac{\phi_0 \left(\frac{J_0(\Omega)}{I_0(\Omega)} + \frac{I_0(\Omega)}{J_0(\Omega)}\right)}{\gamma \Omega^2 \left(\frac{I_0(\Omega)}{J_0(\Omega)} + \frac{I_0(\Omega)}{J_0(\Omega)}\right)} \tag{25}$$

A feature of this particular characteristic is that there is no low-frequency resonance even with a small value of the solid parameter.

This circumstance reflects the mechanical meaning of the boundary condition (21), related with deformation of the sealant. The fact is that the boundary conditions are formulated in such a way that no oscillations are possible without its bending, and consequently the rigidity of the fastening cannot be a source of a special self-form of oscillation, in which the sealer moves like a rigid disk.

In other words, this circumstance can be reflected as follows. For separate values of the elasticity parameter, i.e., for $\gamma = -\frac{1-i}{2}$ and for $\gamma = \infty$ are accordingly, under the condition of free support and rigid pinching conditions.

Although the first case is much low-frequency second, nevertheless the lowest frequency does not drop to zero, as in the previous case, in this case, the anti-resonances are determined by the Equations (26) and (27).

$$\frac{I_0(\Omega)}{J_0(\Omega)} + \frac{I_0(\Omega)}{J_0(\Omega)} \tag{26}$$

and resonances,

$$\frac{I_0(\Omega)}{J_0(\Omega)} - \frac{I_0(\Omega)}{J_0(\Omega)} \tag{27}$$

The correct solution of this frequency equation, in which the value of the first resonance is represented as a function of the stiffness parameter of fixation γ. It is important to note that the dimensionless elastic-fixing parameter $\alpha_{22} R / 2D$ adds additive to the parameter $(1-\gamma)/2$, which characterizes the sealant material. Consequently, the parameter

$$\gamma = -\left(\frac{\alpha_{22} R}{2D} + \frac{1-\gamma}{2}\right) \tag{28}$$

Can take both positive (for small α_{22}) and negative values. This shows that with free support, the quantity $(1-\eta)/2$ forms, as it were, an additional elasticity due to the antiplastic effect.

The most interesting, but very difficult for research because of the greater number of parameters affecting the frequency characteristics is the general case, taking into account the compliance of the support at the same time to displacement and to bending.

We consider the elastic fastening variant, which is described by the following boundary conditions:

$$F = \alpha_{11} W \tag{29}$$

$$-M = \alpha_{22} \frac{\partial W}{\partial r} \tag{30}$$

$$W(r,t) = \frac{P_0 e^{-i\omega t}}{\rho h \omega^2}\left\{-1 + \frac{\Delta_1 I_0(kR) + \Delta_2 J_0(kR)}{\Delta}\right\} \tag{31}$$

$$\Delta_1 = \begin{vmatrix} \alpha_{11} \\ k\left[\alpha_{22} + \frac{D(1-\eta)}{R}\right] \cdot I_0(kR) - Dk^2 \cdot J_0(kR) \\ 0 \end{vmatrix}$$
$$\Delta_2 = \begin{vmatrix} \alpha_{11} \\ k\left[\alpha_{22} + \frac{D(1-\eta)}{R}\right] \cdot I_0(kR) + Dk^2 \cdot J_0(kR) \\ 0 \end{vmatrix} \tag{32}$$

$$\Delta = \begin{vmatrix} \alpha_{11} J_0(kR) + Dk^3(kR); \alpha_{11} J_0(kR) - Dk^3(kR) \\ k\left[\alpha_{21} + \frac{D(1-\gamma)}{R}\right] \cdot J_0(kR) - Dk^2 J_0(kR) \\ -Dk^2 J_0(kR) + k\left[\alpha_{22} + \frac{D(1-\gamma)}{R}\right] \cdot I_0(kR) \end{vmatrix} \quad (32)$$

As can be seen, in this case the frequency response depends on three dimensionless parameters, γ, β. The problem becomes simpler if we confine ourselves to investigating only the natural frequencies, which are functions of only two parameters characterizing the compliance of the fastening. The equation of frequency is written in dimensionless form:

$$-\frac{I_0(\Omega)}{J_0(\Omega)} \cdot \frac{\Omega^2 \gamma}{\beta} + \left[\frac{\Omega^3}{\beta} - \frac{\gamma}{\Omega}\right] \cdot \left[\frac{I_0(\Omega)}{J_0(\Omega)} - \frac{I_0(\Omega)}{J_0(\Omega)}\right] \quad (33)$$

In this case, there is a characteristic low frequency, which can be low if the parameter β is small. It is for small β that the solution of the frequency equation can be represented in the form:

$$\Omega^* = \sqrt[4]{4\beta\left(\frac{\gamma+1}{\gamma+4}\right)} \quad (34)$$

4. CONCLUSIONS

1. Studies of rubber seals of fasteners have shown that sealants sometimes break down tirelessly. Studies have shown that this is due to physical resonances caused by dynamic forces.
2. With the dynamic behavior of a circular borehole seal under the influence of external pressure, a low frequency physical resonance arises due to a special form of oscillation.
3. Preliminary consideration of resonant efficiency in the design of elastic elements will increase their longevity.

ACKNOWLEDGEMENT

This work was financially supported by the Science Development Foundation - Grant No. EIF-MQM-ETS-2020-1(35)-08/04/1-M-04.

REFERENCES

[1] R.A. Aliyev, I.B. Turksen, J. Kasprzuk, "Decision and Control", Conference Dedicated to Professor Lutfizadeh, pp. 397-399, Antalya, Turkey, September 2015.

[2] V.T. Mamedov, J.N. Aslanov, "Calculation of Stress-Strain State of Sealing Arrangement from rubber Comproition Materials using in Oilfield Equipment", European Science and Technology, Materials of the XII International Research and Practice Conference, Publishing Office Web Verlag Waldkraiburg, pp. 78-81 Munich, Germany, December 2015.

[3] A.J. Nuraddin, S.A. Bahman, "Forecasting of Improved Straightforward Valves Technical Condition Using Fuzzy Inference Models", IFAC-Papers Online, Vol. 51, Issue 30, pp. 12-14, Baku, Azerbaijan, September 2018.

[4] S.P. Timoschenko, S.V. Krigers, "The Theory of Plates and Shells", M.State Publishing House Physics-Mathematics-Literature, pp. 33-37, California, USA, June 1963.

[5] J.N. Aslanov, "Implementation of a New Model of Rubber Matrix Seals on Closing Devices", Equipment. Technology. Materials. pp. 24-27, Baku, Azerbaijan, October 2020.

[6] V.L. Biderman, N.A. Sukhova, "Installation, Operation and Repair of Borehole Rod Pumping Units", Elm, pp. 208, Baku, Azerbaijan, September 1993.

[7] V.T. Mammadov, T.U. Khankishieva, "Determination of the Optimal Seal Shape of the Shaft Seal of Rod Borehole Pumps", Equipment and Technologies for the Oil and Gas Complex, pp. 31-33, Moscow, Russia, May 2015.

[8] A.J. Nuraddin, H.Z. Seidi, "Determination OG the Design Parameters of Seals with Required Properties", Process Management and Scientifik Developments, Birmingham, United Kidtom, pp. 204-208, November 2020.

[9] V.T. Mammadov, J.N. Aslanov, F.G. Bayramov, Z.S. Huseynli, "Methodology for Calculating the Sealing Parameters of the Packer Sealing Unit used in the Repair of Wells", Global Science and Innovation Proceedings of the VIII International Scientific Conference, pp. 184-196. Melburn, Australia, June 2016.

[10] S.G. Vlasova, "Fundamentals of Chemical Glass Technology", Ural Publishing House, Vol. 108, Yekaterinburg, Russia, November 2013.

[11] J.N. Aslanov, "New Model Rubber Matrix for Connectors Application of Sealers", Equipment, Technologies, Materials, Vol. 3, Issue 1, Baku, Azerbaijan, January 2020.

[12] M.B. Babanli, "Theory and Practice of Material Development Under Imperfect Information", Advances in Intelligent Systems and Computing, pp. 4-14, Munich, Germany, May 2018.

[13] M. Babanli, R. Karimov, A. Bayramov, I. Abbasov, "An Impact of the Ladle Lining on the Refining of Reinforced Steel when Blowing with Powders", Eastern-European Journal of Enterprise Technologies, Vol. 5, Issue 1, No. 101, pp. 65-71, Ukraine, Kiev, February 2019.

[14] V.T. Mamedov, S.R. Qurbanov, "The Impact of the Relatively ION Compression Rubber to Functions Sealing Elements of well Bore Packers", International Conference on Soft Computing, Computing with and Perceptions in System Analysis, ICSSW, pp. 397-399, Antalya, Turkey, September 2015.

[15] J.N. Aslanov, K.S. Mammadov, N.A. Zeynalov, "Selection of Structural Materials for Improved Liner Motion Gate Valves Based on Friction Correlation Method", International Journal of Advanced Technology and Engineering Exploration, Issue 9, Vol. 87, pp. 155-161, India, June 2022.

[16] Shruti, S. Pahadsingh, B. Appasani, N. Bizon, "Mathematical Absorbers in Microwave and Terahertz Regions", International Journal on Technical and Physical Problems of Engineering (IJTPE), Issue 47, Vol. 13, No. 2, pp. 149-159, June 2021.

[17] A.M. Hashimov, N.M. Tabatabaei, K.B. Gurbanov, L.C. Suleymanova, Z.A. Tagiyeva, "Structure Interpretations of Polymers Properties", International Journal on Technical and Physical Problems of Engineering (IJTPE), Issue 35, Vol. 10, No. 2, pp. 34-38, June 2018.

BIOGRAPHIES

Jamaladdin Nuraddin Aslanov was born in Jalilabad, Azerbaijan on August 18, 1972. He received the Ph.D. degree in the field of oil and gas (mechanical engineering) from Azerbaijan State Oil and Industry University, Baku, Azerbaijan in 2011. He is the head of industrial machinery department at the same university. His research interests are related to increase the efficiency of valves being used in industry.

Zenfira Seydi Huseynli was born in Baku, Azerbaijan on June 23, 1961. She received the Ph.D. degree in engineering science from Azerbaijan State Oil and Industry University, Baku, Azerbaijan in 2015. She has worked for 32 years in different positions at Department of Industry Machines of the same university. At the moment she is an Assistant Professor.

Natig Mayis Abbasov was born in Baku, Azerbaijan on February 13, 1966. He received the Ph.D. degree in engineering science from Azerbaijan State Oil and Industry University, Baku, Azerbaijan in 2003. He has worked for 15 years in Department of Foreign Economic Relations of Azerbaijan State Oil Company as Head of the Regime Management Department. Currently, he is an Associate Professor of Department of Industrial Machines of Azerbaijan State University of Petroleum and Industry. His scientific field of research is optimal synthesis of heat exchange systems in the petrochemical industry.

Aynur Valeh Sharifova was born in Baku, Azerbaijan on October 20, 1984. She received the Ph.D. degree in engineering science from Azerbaijan State Oil and Industry University, Baku, Azerbaijan in 2014. She has worked for 8 years in Department of Materials Science and Processing Technology of the same university as assistant. Her scientific field of research is composite materials and powder metallurgy.