Sealing joints with elastic edge

Yu I Belogolov¹, S K Kargapoltsev¹ and V E Gozbenko¹,²
¹Irkutsk State Transport University, 15, Chernyshevsky Street, Irkutsk, 664074, Russia
²Angarsk State Technical University, 60, Chaykovsky Street, Angarsk, 665835, Russia
E-mail: pr-mech@mail.ru

Abstract. It was cylindrical shell elements that were mainly considered, as technologically simpler ones. The angle of the cone α is usually accepted to be 10–15°, in order to avoid self-braking (jamming) of the loaded sealing joint. The long service life of the sealing joint implies the absence of plastic deformations of the shell element (excluding, of course, the deformation area of the sealed joint, i.e. the sealing line), therefore, usually hardened steel, bronze, forming a wear-resistant friction pair with the spool valve, are used as the material of the shell element.

1. Introduction

Nowadays, in the valve industry in metal-to-metal seals, there are mainly two types of sealing joints used: flat and conical. Flat surfaces are technologically easier to process, achieving accurate shape and minimal roughness, using grinding-in of sealing surfaces [1-6]. In conical sealing joints, that are more demanding on manufacturing accuracy, technological flaws can be compensated by increased contact pressures due to the wedge effect. Despite the presence of recommendations and regulatory documents, the question of the width of the seal band for both flat and conical sealing joints has not been unambiguously resolved. Thus in the works [7, 11] the question of minimizing the sealing pressure is raised by choosing the width of the sealing area and the angle of taper. In any case, even with regard to wear, the authors are talking about the contact surface, which is very difficult to maintain in the required condition during operation due to thermal cycling, installation loads, impact effects, and technological features of operation.

As a result, the ground sealing surfaces are deformed, deviations of form and waviness appear, that requires the corresponding increase in sealing force, the use of special overlay welding, etc.

A fundamentally different sealing scheme is used in sealing joints with shell elements or simply with shells. Here the sealing surface is not formed in advance; seal is not flat but is carried out along a line. Due to the fact that, during the operation of the valve, the parameters of the working medium (pressure) change, the sealing force changes (the parameters of the drive systems are also not constant), the shell is de-formed, and thus it is possible to speak about the non-constancy of the sealing line. Each time the valve is triggered or the flange connection is loaded, different parts of the sealing elements are in contact.

Therefore, grinding-in of sealing joints using deformable shell elements is meaningless.

Shell elements can be used as a spool valve or seat, as shown in Figure 1.
Figure 1. Scheme of the sealing joint with the shell element as a spool valve (a) and seat (b).

To implement self-sealing, the cone may come in contact with the shell element along the inner or outer surface, as shown in Figure 2.

Figure 2. Scheme of self-sealing when the working medium is supplied to the valve (a) and under the valve (b).

On the review of sealing joints with the use of thin-walled elements, more than 70 home-produced and foreign seals are mentioned, which indicates the interest of valve manufacturers to this type of sealing joints.

2. Results and Discussion
The first publication where sealing joints with a shell element are considered is the paper [11], the static and dynamic calculations of the shell element are given in more detail in the work [7]. It was cylindrical shell elements that were mainly considered, as technologically simpler ones. The angle of the cone \( \alpha \) is usually accepted to be 10-15°, in order to avoid self-braking (jamming) of the loaded sealing joint. The long service life of the sealing joint implies the absence of plastic deformations of the shell element (excluding, of course, the deformation area of the sealed joint, i.e. the sealing line), therefore, usually hardened steel, bronze, forming a wear-resistant friction pair with the spool valve, are used as the material of the shell element.

In contrast to a flat or conical sealing joint warped as a result of, for example, thermal cycling, the shell element easily assumes the form of a spool valve, fits tightly to it, therefore the sealing efforts for that kind of a connection are closer to the metal-polymer seal than to the flat or conical metal-to-metal seals.
For a proper fit of the shell to the mating conical surface, the shell should be made as thin-walled as possible (while reducing the required sealing force), but ensuring the work of the shell in an elastic zone. If for flat and tapered sealing joints there are recommendations for choosing the geometrical dimensions of the deformation area of the sealing joint, then for the shell element in each specific case it is necessary to perform a strength calculation. The basic computational scheme of the shell seat is shown in Figure 3. The determination of the intensity of the loads T and Q for various design variants of the sealing joint is considered in [7].

![Computation scheme of the shell seat.](image)

The differential equation of the radial deflection of the points of the middle surface of the shell is:

$$\frac{d^4 w}{dx^4} + 4\beta^4 w = -\frac{\mu T}{rD} + \frac{p}{D},$$

(1)

where \(w = w(x)\) is the radial displacement of the points of the middle surface of the shell; \(\beta = \sqrt{\frac{3(1-\mu^2)}{r_o^2 h_o^2}}\) is the shell parameter; \(D = \frac{E h_o^2}{12(1-\mu^2)}\) is the bending stiffness of the shell; \(E, \mu\) is the modulus of elasticity and Poisson ratio.

The solution of the equation (1) is sought in the form:

$$w(x) = \sum_{i=1}^{4} C_i K_i + \left( \frac{p - \mu T}{r_o} \right) \frac{r_o^2}{E h_o} ,$$

(2)

where \(C_i\) are the integration constants, \(K_i\) are the A.N. Krylov functions.

\[
K_1(\beta x) = \frac{1}{2} (e^{\beta x} + e^{-\beta x}) \cos \beta x; \\
K_2(\beta x) = \frac{1}{4} \left( e^{\beta x} + e^{-\beta x} \right) \sin \beta x + \left( e^{\beta x} - e^{-\beta x} \right) \cos \beta x; \\
K_3(\beta x) = \frac{1}{4} \left( e^{\beta x} - e^{-\beta x} \right) \sin \beta x; \\
K_4(\beta x) = \frac{1}{8} \left( e^{\beta x} + e^{-\beta x} \right) \sin \beta x - \left( e^{\beta x} - e^{-\beta x} \right) \cos \beta x.
\]

The structure of the Krylov functions as a combination of elementary functions, including the
required thickness \( h_o \) as an argument, does not allow analytically finding \( h_o \), therefore it is impossible to give immediately recommendations on the choice of the geometrical parameters of the shell element. Internal force factors arising in the shell are determined by the expressions:

\[
M_s(x) = D \frac{d^2 w}{dx^2}; M_i(x) = \mu M_s(x) T_i(x) = \mu T + \frac{E h_o}{r_o} w(x),
\]

and the stresses on the shell surface are determined by the expressions:

\[
\sigma_{s\text{max}}(x) = \frac{T}{h_o} \pm \frac{6M_s(x)}{h_o^2}; \quad \sigma_{i\text{max}}(x) = \frac{T_i(x)}{h_o} \pm \frac{6M_i(x)}{h_o^2}
\]

Further, using the accepted hypothesis of strength, it is concluded that the choice of geometrical dimensions is correct.

If the shell element works under shock loading (valve), then the adopted computational scheme of the shell is the same, however, \( T \) and \( Q \) are set taking into account the loading dynamics [8-10, 12-16]. In general, the maximum dynamic load that occurs when the valve is triggered can be determined from the expression:

\[
F_{\text{dim}} = F + \sqrt{F^2 + 2(E_o - E_{\text{dim}})(c_1 + c_2 \tan \alpha \tan(\alpha + \phi))},
\]

where \( F \) is the static sealing force; \( E \) is the kinetic energy of the moving parts of the valve at the moment of impact; \( E_{\text{dim}} \) is the energy expended on the choice of deviations of the shape of the shell element; \( c_1 \) is the drive stiffness; \( c_2 \equiv \frac{Q}{w(0)} = \frac{2\pi r_o \beta \omega D}{\Phi_i(\beta l_o)} \) is the radial stiffness of the shell element; \( \Phi_i(\beta l_o) \) is the influence function (expressed through the Krylov functions); \( \phi \) is the friction angle in the joint.

The need to reduce \( F_{\text{dim}} \) requires the minimization of stiffness \( c_2 \). \( E \) is determined (controlled) in accordance with the requirements for valve performance. The influence function \( \Phi_i(\beta l_o) \), which determines the radial stiffness of the shell, depends on its length \( l \) and increases from zero for \( \beta l_o = 0 \) to 0.5 for \( \beta l_o \approx 2 \) and above. Shells in which \( \beta l_o \leq 2...3 \) are called short, otherwise long. For a long shell, dimensions (length) grow, and the stiffness \( c_2 \) does not change, therefore usually \( \beta l_o = 2...3 \) is taken.

If the expression for the maximum dynamic load (5) is substituted into the accepted computational scheme of the shell, then assuming \( c_1 \ll c_2 \), it is possible to get an expression for the shell thickness ensuring that the permissible stress \( \sigma_{\text{adm}} \) does not exceed the given impact energy and the minimum stiffness \( c_2 \):

\[
h_o = \left[ \frac{1.64\sigma_{\text{adm}}r_o F + E_o E \tan \alpha}{\sigma_{\text{adm}}r_o^{1.5} \tan(\alpha + \phi) \Phi_i(\beta l_o)} \right]^{2/3}
\]

In the derivation of (6), \( E_{\text{dim}} = 0 \) was assumed for simplicity, \( \sigma_{\text{adm}} \), which is used in the safety margin. The definition of \( E_{\text{dim}} \) with the well-known circular plot of the end of a shell element is described in [7]. \( E_{\text{dim}} \) can be used as a criterion for the rejection of warped shells.

One of the ways to reduce the maximum dynamic load that occurs during shock loading of the shell seat in the valves is to reduce the unit stiffness of the sealing joint. This can be obtained by using the shell-plate seat shown in Figure 4.
Figure 4. Scheme of the sealing joint with the shell-plate seat.

If the stiffness of the plate element is denoted as \( c_3 \), then the maximum dynamic load during the shock landing of the spool valve will be determined by the expression:

\[
F_{\text{dyn}} = F + \sqrt{F^2 + 2(E_e - E_{of}) \left( \frac{c_1 + c_2 \tan \alpha \tan(\alpha + \phi)}{c_1 + c_2 \tan \alpha(\alpha + \phi) + c_3} \right) c_1}.
\]  

(7)

In fact, there is a sequential attachment of the stiffness \( c_1 \) to the parallel-connected \( c_1 \) and \( c_2 \), which leads to a significant reduction in unit stiffness, a decrease in maximum dynamic load, a decrease in the thickness of the shell element \( h_o \), a decrease in \( E_{of} \), and, therefore, a decrease in \( F \).

The calculation of the shell-plate seat is given in, where already unknown quantities are not only the thickness of the shell \( h_o \), but also the thickness of the plate \( h_p \) (its outer diameter is considered known for reasons of design).

The method of static, dynamic, strength and stiffness calculations of shell-plate seat shown in these works allows formulating and solving the following optimization problem: to determine the thickness of shell \( h_o \) and plate \( h_p \) elements providing the minimum value of the unit stiffness of the sealing compound (fraction in (7)) under the condition that the permissible stress in the shell and plate elements is not exceeded.

When dynamically calculating the valve's impact response, the loading of the shell element by the pressure of the working medium is not taken into account, because in a fraction of a second, while the transient process of shock loading lasts, the pressure of the working medium does not load the shell-plate element.

However, after a while the loading scheme will change. The forces \( T \) and \( Q \) will take their stationary (static) value, but there will be a pressure load of the working medium. Therefore, for selected optimal dimensions of the shell-plate seat, a verification calculation should be performed to check the strength under the new operating conditions of the sealing joint.

If the shell is loaded not dynamically, but statically (a flange connection), then its strength can be determined not by \( T \) and \( Q \), but by pressure \( p \). In this case it is more profitable to make the shell shorter than \( \beta h_o = 2...3 \).

In the works [11, 17-18] an attempt was made to link the length of the shell element with the pressure of the working medium, but the recommendations given were obtained on the basis of solving particular problems.

To experimentally verify the correctness of the developed analytical model for determining the stiffness parameters of a shell-plate seat, an experimental installation was performed, shown in Fig. 5.
Figure 5. Photograph of the installation to experimentally determine the stiffness parameters of the shell-plate seat.

In addition to experimental studies, finite element modeling was performed in APM WinMachine and MSC vN4W (Figure 6), which showed a satisfactory agreement between the analytical, numerical and experimental results.

![Displacement maps in APM WinMachine (a, b) and MSC vN4W (c, d); (a), (c) - axial; (b), (d) - radial](image)

Figure 6. Displacement maps in APM WinMachine (a, b) and MSC vN4W (c, d); (a), (c) - axial; (b), (d) - radial

3. Conclusion
A more strict statement of the choice of the geometrical dimensions of the shell seat can also be formulated as an optimization task: to choose the thickness $h_o$ and height $l_o$ of the shell element, ensuring the minimum value of the radial stiffness $c_2$ provided that the allowable stress in the shell
element is not exceeded, which can be used as a basis for developing a standard material on the dimensioning of the shell element.

References
[1] Shtayger M G, Balanovsky A E, Kondratev V V, Karlina A I, Govorkov A S 2018 Adv. Eng. Res. Conf. Proc. pp 360-364
[2] Balanovsky A E, Shtayger M G, Grechneva M V, Kondratev V V, Karlina A I 2018 IOP Conf. Ser.: Mater. Sci. and Eng. 411 012012
[3] Balanovsky A E, Shtayger M G, Kondratev V V, Huy Vu V, Karlina A I 2018 IOP Conf. Ser.: Mater. Sci. and Eng. 411 012013
[4] Balanovsky A E, Kondratev V V, Shtayger M G, Nebogin S A, Karlina A I 2018 IOP Conf. Ser.: Mater. Sci. and Eng. 411 012014
[5] Ivanchik N N, Balanovsky A E, Shtayger M G, Sysoev I A, Karlina A I 2018 IOP Conf. Ser.: Mater. Sci. and Eng. 411 012035
[6] Balanovsky A E, Shtaiger M G, Kondratev V V, Karlina A I, Govorkov A S 2018 J. of Phys.: Conf. Ser. 012006
[7] Dolotov A M, Ogar P M, Chegodaev D E 2000 Fundamentals of the theory and design of the aircraft pneumohydraulic fitting seals: A textbook (Moscow: MAI Publ.) p 296
[8] Dolotov A M 1995 Development of calculation methods and design of seals with a shell element for aircraft, Dissertation, Moscow
[9] Khomenko A P, Gozbenko V E, Kargapoltsev S K, Minaev N V, Karlina A I 2017 Int. J. of Applied Eng. Res. 12(23) 13773–13778
[10] Gozbenko V E, Kargapoltsev S K, Minaev N V, Karlina A I 2016 Int. J. of Applied Eng. Res. 11(23) 11132–11136
[11] Karmugin B V, Stratinevsky G G, Mendelson D A 1983 Valve seals of pneumohydraulic units (Moscow: Mashinostroenie Publ.) p 152
[12] Balanovskii A E, Van Huy V 2018 J. of Friction and Wear 39(4) 311–318
[13] Balanovskiy A E 2017 Welding Int. 31(6) 467–476
[14] Medvedev S I, Nezhivlyak A E, Grechneva M V, Balanovsky A E, Ivakin V L 2015 Welding Int. 29(8) 643-649
[15] Merson E D, Myagkikh P N, Klevtsov G V, Merson D L, Vinogradov A 2018 Eng. Fracture Mechanics 202
[16] Klevtsov G V, Murashkin M Y, Kushnarenko V M, Klevtsova N A, Merson E D, Belov P A, Pigaleva I N, Valiev R Z 2017 Protection of Metals and Physical Chemistry of Surfaces 53(7) 1240–1246
[17] Nazarov D V, Smirnov V M, Zemtsova E G, Valiev R Z, Yudintceva N M, Shevtsov M A 2018 ACS Biomaterial Sci. and Eng. 4(9) 3268–3281
[18] Zemtsova E G, Morozov P E, Valiev R Z, Smirnov V M, Yudintceva N M, Shevtsov M A 2018 Int. J. of Nanomedicine 13 2175–2188