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Research of Temperature Influence on the Contact Pressure in Radial Seals

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Abstract: For the operation in hostile environments and a wide range of temperature changes (in cryogenic environments) the Samara State Aerospace University developed radial contact seals of reciprocative transportation, including thin-walled open-loop casing made of fluoroplastic-4 over the cross section and an annular elastic element made of the wire material MR. The technique of the contact pressure calculation is presented. This contact pressure is generated by O-ring within its contact with the cylinder surfaces (along the outer diameter) and a rod (along the inner diameter) based on the mechanical characteristic temperature dependency of the applied materials and the thermal deformation of the parts. The calculations are performed to determine the nature of the contact pressure change in the seal when the temperature changes from normal, corresponding to the state of the seal assembly in the assembly, to low, corresponding to the conditions of the seal assembly in cryogenic units. It is shown that the variation nature of the contact pressure depending on the temperature is most strongly influenced by the polymer shell and the distribution of contact pressure across the seal width depends on the rigidity of the shell butts. The recommendations on the selection of the seal design parameters for the designed structure are proposed. These seals are operated within the wide range of temperature change conditions.

Key words: Contact radial seals, polymer casing, elastic element, contact pressure, temperature

INTRODUCTION

The problem of plug connection sealing is still relevant. It becomes the acutest one when it manifests itself in the situations of extreme operating conditions which are typical for aircraft engines (Belousov and Parovay, 2001).

One of the promising ways for the aircraft engine improvement is to increase the efficiency of their elements. For example, the 1% reduction of air leakage in a compressor or gas leakage in a turbine increases the efficiency at 2-3 and 1% increase of air leakage from the fan duct with the bypass ratio of 5.6 leads to the efficiency (to the specific fuel consumption increase up to 3-4%) and the flight distance decrease by 1-2% (Belousov and Parovay, 2001).

The problem of sealing in rocket engines is particularly relevant due to the use of chemically active fuel components at high pressure level (up to 60 MPa and higher) and a wide temperature range (from cryogenic up to 2000 K).

Thus, ensuring the connection tightness is one of the decisive factors for efficiency, reliability, environmental security, resource and energy saving.

With the equipment improvement there is a need for ultra-high pressure, extreme temperature (from cryogenic to high ones), corrosive environment values use. Such, operating conditions strongly affect the tightness of detachable joints, especially in aerospace engineering which is characterized by the use of light-weight (with reduced stiffness) flange joint designs.

The application of polymeric (non-metallic) gaskets is possible in a relatively narrow range of operating conditions. Therefore, the above-mentioned conditions prescribe almost the only option of metallic seal use almost without any limitations other than the strength and heat resistance of the seal and the coating material.

MR material ("Metal analog of rubber" or "Metallic rubber") (Belousov and Parovay, 2001; Belousov *et al*., 2001) allows for a high range of operating conditions natural for metals with a number of qualities favorable for use in seals such as a large elastic strain (up to 20-25%), elasticity. MR material is a composition of the stretched wire spirals, packed into a workpiece with a mutual crossing and compacted to the density of $\rho_p = (0.3\text{-}0.5)\ \rho_m$ where ρ_p: MR material density; ρ_m: wire material density.

The material resiliency is ensured by the contact between itself and the deformations of separate wires. The MR material deformation is accompanied by the energy dissipation which allows to make different vibration isolators, dampers and shock absorbers on its basis (Ulanov *et al*., 2008; Ulanov and Ponomarev, 2009; Jiang *et al*., 2004; Ao *et al*., 2003).

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Despite a number of specific problems (such as a static contraction, anisotropy of properties, porosity, characteristic stiffness), the MR material allowed to develop a number of sealing designs.

For the radial seals of reciprocating motion pairs, operating under the cryogenic environments, Samara State Aerospace University developed the O-rings, including a thin-walled non-closed shell made of a polymeric material and a resilient element made of MR material. The Teflon shell provides a tight sealing of a seal assembly and good anti-friction properties allow to minimize the friction and wear of the seal assembly moving parts. The weak dependence of the stiff and geometric characteristics of the MR material on temperature allows to compensate the thermal deformation and the casing stiffening at temperature lowering (Parovay and Borisov, 1984). The movable contact in the seal may be carried by the outer diameter of the sealing ring (the cylinder seal) or by the inner diameter (rod seal).

In this study, the contact pressure in the designed seal is explored when the temperature changes from normal (at which a seal assembly is assembled) to operational low temperature of cryogenic environments.

MATERIALS AND METHODS

The initial state seals depends on the residual stresses arising in detail during the ring manufacture. The resilient element and the casing are manufactured separately and assembled in a special fixture. The assembled ring is subjected to heat setting comprised of the heating to the temperature of 573+5 K and the curing for 30-45 min. The purpose of this operation is the final formation of the shape in particular the obtaining of butt belts at the casing edges, preventing its slide from the elastic element. The thermal fixation minimizes the stresses arising during the sealing ring assembly.

The cross-sectional dimensions of the sealing rings are shown by Fig. 1a. R_0 and R'_0 here are the radii of the outer and inner surface of the shell; h the casing thickness; H_u elastic element section height; c, c' butt belt width. In the assembled seal the ring contacts along the outside diameter with the cylinder of the radius R_1 (Fig. 1a), along the inner diameter with the Rod (R_2).

For ease of analysis, let's put down all sizes of the seal in a relative form. At that the dimensions that characterize the contact with the cylinder will be divided by R_1, the contact with a rod will be divided by R_2: $\bar{R}=R_2/R_1$ or $\bar{R}'=R_1/R_2$ relative sealing radius; $\bar{h}=h/R_1$ or $\bar{h}'=h/R_2$ relative casing width (or $\bar{H}_u=H_u/R_2$) relative height of an elastic element section; $\bar{c}=c/R_1$ (or $\bar{c}'=c/R_2$) relative width of the casing butt belt.

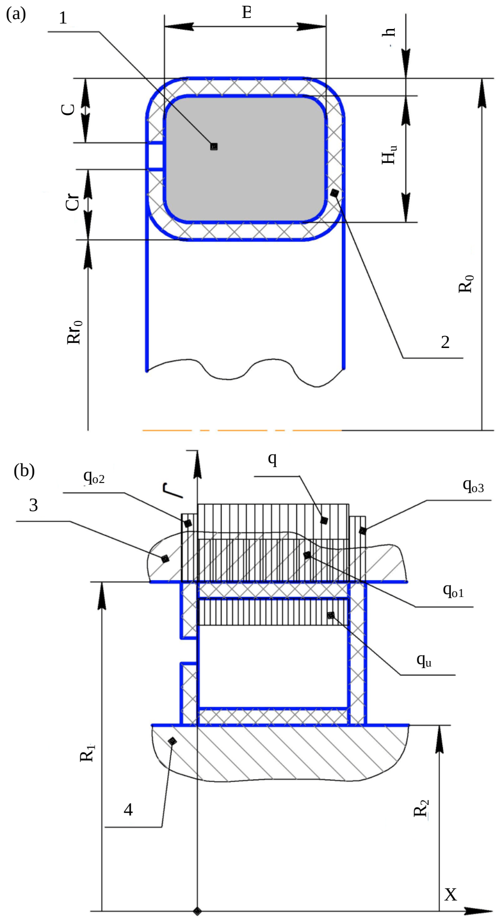

Fig. 1: Sealing scheme: a) o-ring prior to assembly and b) calculation scheme; 1: elastic element; 2: shell; 3: cylinder; 4: rod

The contact pressure in the seal largely determining by the performance of the seal assembly is provided by the elastic element and the shell deformation during the ring assembly as well as by the pressure of the sealed environment. The mounting tightness along the outer diameter $n = R_0\text{-}R_1$ and along the internal diameter $n' = R_2\text{-}R'_0$. The relative tightness during the seal installation makes $\bar{n}=n/R_1, \bar{n}'=n'/R_2$. Without taking into account the decrease of the shell thickness, we obtain the relative deformation of the elastic element.

$$\varepsilon_u = \frac{n+n'}{H_u} = \frac{\bar{n}+\bar{n}'\bar{R}}{\bar{H}_u} \tag{1}$$

Then, we will consider mainly the sealing ring contact zone with the cylinder and the parameters relating to the contact zone with the rod shall be defined by analogy.

The contact pressure calculation and the results of its change study within the temperature range of 4-293 K for a simplified sealing model without taking into account the elasticity of the butt belts at the edges of the shell are presented by Parovay and Borisov (1984). However, as the sealing tests show at the temperature of 77 K, the butt belts cause a significant sealing change. Therefore, the researchers developed the technique which allows to consider the impact of the shell end sections on the seal contact pressure.

The contact pressure without the sealing medium pressure (at the sealing medium pressure <0.1 MPa) is determined by the pressures caused by the casing and elastic element deformation:

$$q = q_u + q_o \tag{2}$$

The elastic element pressure (Belousov and Parovay, 2001; Ulanov *et al*., 2008; Parovay and Borisov, 1984) is determined by the dependence (Eq. 3):

$$q_u = \frac{\bar{A}E\varepsilon_u}{1-\alpha\varepsilon_u} \tag{3}$$

where, $\bar{A}$, α are the empirical dimensionless parameters of MR material, characterizing its rigidity and elasticity, respectively, $\bar{A}=(7\text{-}20)10^{-5}$, $\alpha = 4, 5, ..., 8$.

The pressure generated by the casing is composed of a pressure caused by the deformation of the cylindrical portion q_{o1} and the pressures from the end portions q_{o2} and q_{o3}. In general case:

$$q_o = q_{o1} + q_{o2} + q_{o3}$$

In used seals the cylindrical portions may be regarded as short thin-walled casings, so:

$$q_{o1} = \frac{\bar{n}E_o\bar{h}}{\left(1+\bar{n}-0.5\bar{h}\right)^2} \tag{4}$$

where, E_o elasticity module of the casing material. Let's consider the case 1, when the deformation of the butt and cylindrical sections may be regarded as independent ones. This calculation scheme is applicable for the mounted seal, if the stretch is a positive one (n>0 or n'>0). At that along the edges of the contact area on the butt portions the pressure increases sharply as the rigidity of these portions is higher than the cylindrical one.

Generally, the deformation of the butt portions during the mounting process of the seal is accompanied by buckling. Thus, the belts are bent and their rigidity in the radial direction is significantly reduced. The loss of stability is contributed by the smooth transitions between the cylindrical and the butt portions of the casing. The value q_{o2} in case of stability loss may be found from the expression (Birger *et al*., 1979):

$$q_{o2} = K_s \frac{\pi^2\bar{h}^2E_o}{12\left(1-\mu^2\right)\left(\bar{n}+1\right)^2} \tag{5}$$

Here:

K_s = The coefficient taking into account the conditions of edge sealing and the ring width
μ = Poisson's ratio of the shell material

The loss of stability occurs during deformation.

$$n = K_s \frac{\pi^2h^2}{12\left(1-\mu^2\right)R_o^2} \tag{6}$$

During tightness prior to buckling the butt belts develop pressure which may be determined from the Lame equation (Birger *et al*., 1979) for the ring loaded with radial load. Since, the radial displacement of points, located along the ring periphery is known and is equal to n, the pressures q_{o2} and q_{o3} make:

$$q_{o2} = \frac{E_o\bar{n}}{1+\bar{n}-\bar{h}}\left[\frac{2\left(1+\bar{n}-\bar{h}\right)^2}{\bar{c}\left(2+2\bar{n}-2\bar{h}-\bar{c}\right)}-1-\mu\right] \tag{7}$$

$$q_{o3} = \frac{E_o\bar{n}}{1+\bar{n}-\bar{h}}\left[\frac{8\left(1+\bar{n}-\bar{h}\right)^2}{\bar{H}_u\left(4+4\bar{n}-4\bar{h}-\bar{H}_u\right)}-1-\mu\right] \tag{8}$$

In case 2, the shell in a free state (without an elastic element) has $R_o<R_1$ or $R'_o>R_2$. In the presence of the elastic element, a contact pressure may be a positive one due to the action of the elastic element pressure and the sealed medium. In this case, the shell reduces the contact pressure. The linear load from the butt belts is attached to the edges of the casing cylindrical part. Let's consider the balance of loads between the sealing ring parts mounted according to the inner diameter with the interference but free along the outer diameter. The pressure of the elastic element q_u is balanced with loads caused by the shell cylindrical and the butt portion defotrmation. Its value is determined by the total interference along the outer and inner diameter of the seal.

Using the method of thin-walled short shells calculation (Birger *et al*., 1979), we define the deformations of the shell cylindrical portion under the action of an elastic element distributed load:

$$y_a = \frac{q_u R_o \left(1+\bar{n}-0.5\bar{h}\right)^2}{E_o \bar{h}} \quad (9)$$

and a concentrated load $q_{o2}h$ within the section x = 0 (Fig. 1b):

$$y_b(x) = \frac{q_{02}h}{D\xi^3}\left[\psi_1 K_0(\xi x) - \psi_2 K_1(\xi x) + K_3(\xi x)\right] \quad (10)$$

Where:

$y_b(x)$ = The radial deformation of the shell in the cross section x from the load impact q_{o2}

$D = E_o h^3/12(1-\mu)$ = Cylindrical stiffness of the casing

Ψ_1, Ψ_2 = Parameters, determined from the boundary conditions on the casing edges:

$$\psi_1 = \frac{Sh(\xi B)Ch(\xi B) - Sin(\xi B)Cos(\xi B)}{2\left[Sh^2(\xi B) - Sin^2(\xi B)\right]}$$

$$\psi_2 = \frac{Sh^2(\xi B) + Sin^2(\xi B)}{2\left[Sh^2(\xi B) - Sin^2(\xi B)\right]}$$

[Xi]$\xi = \sqrt{\frac{3\left(1-\mu^2\right)}{R_o^4 \bar{h}^2}}$

$K_0(\xi x)$, $K_1(\xi x)$, $K_3(\xi x)$ is Krylov functions:

$$K_0(\xi x) = 0.5\left(e^{\xi x} + e^{-\xi x}\right)Cos(\xi x)$$

$$K_1(\xi x) = 0.25\left[\left(e^{\xi x} + e^{-\xi x}\right)Sin(\xi x) + \left(e^{\xi x} - e^{-\xi x}\right)Cos(\xi x)\right]$$

$$K_3(\xi x) = 0.125\left[\left(e^{\xi x} + e^{-\xi x}\right)Sin(\xi x) - \left(e^{\xi x} - e^{-\xi x}\right)Cos(\xi x)\right]$$

The radial deformation of the shell under the concentrated load $q_{o3}h$ in the section x = B is equal to:

$$y_c(x) = \frac{q_{o3}h}{D\xi^3}\{\psi K_o[\xi(B-x)] - \varsigma K_1[\xi(B-x)] + K_3[\xi(B-x)]\} \quad (11)$$

Using the superposition principle, we obtain an equation describing the deformation of the shell surface:

$$y(x) = \frac{q_u R_o \left(2+2\bar{n}-\bar{h}\right)^2}{4E_o \bar{h}} - \frac{hq_{o2}}{D\xi^3}\phi(\xi x) \quad (12)$$

where, $\phi(\xi x)$ is the parameter taking into account the deformation change of the shell width. Its values are calculated for different ratios of the butt belt stiffness shown by Fig. 2.

The contact pressure during the installation the considered sealing variant in the cylinder will be maintained in the case of:

$$y(x) > |\bar{n}| R_1 \quad (13)$$

It is equal to the pressure difference of the elastic element at the values of its relative deformation Eq. 1:

$$\varepsilon_u = \frac{\bar{n} + \bar{n}'\bar{R}}{\bar{H}_u}$$

$$\varepsilon_u(x) = \frac{\bar{n}'R_2 - y(x)}{H_u} \quad (14)$$

There is no contact for the sealing portion at which the condition Eq. 13 is not satisfied and the sealing may keep its performance only due to the presence of contact at the adjacent portions of the sealing ring width.

The developed computational model allows to perform the exploration of the various factor influence on the contact pressure and the sealing performance.

The temperature change affects the mechanical properties of the sealing ring materials (in this case eflon-4 and MR material) and causes the thermal deformation of the sealing parts, including the size of the contact surfaces the cylinder and the rod. In this study, we consider the range of temperature change from the

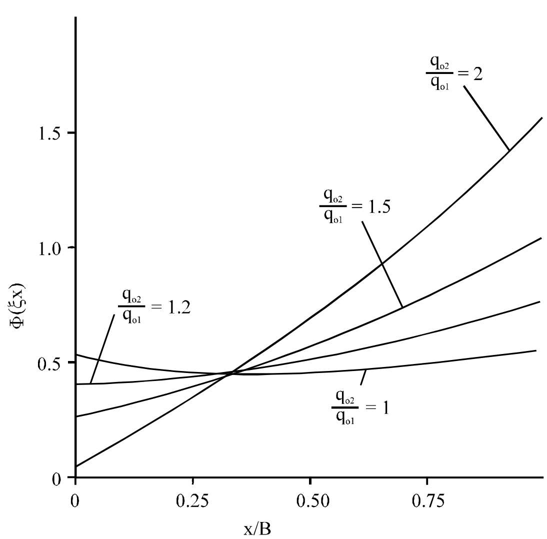

Fig. 2: $\Phi(\xi x)$ change across the contact zone width

normal one (293 K) up to 4K (the liquid helium temperature). At the elevated temperature the stress relaxation in the materials is increased (Chersky, 1981). However, the designed seals are used mainly at low temperatures when the relaxation phenomena are less significant.

The data concerning the material mechanical properties changes in this temperature range are taken from the literature (Parovay and Borisov, 1984; Wigley, 1974) and partly obtained experimentally by Parovay and Borisov (1983).

RESULTS AND DISCUSSION

The contact pressure change study was carried out for the sealing immediately after its installation without the sealed medium pressure. The relative change of the elastic element pressure at the temperature drop in the range from 297-4K is shown by Fig. 3.

The temperature decrease causes the wire material elasticity module increase and the simultaneous reduction of the elastic element relative deformation (as the shell thickness decreases faster than the other sealing dimensions from the temperature influence). At a low relative thickness of the shell (<0.02) the elastic element pressure variation on the temperature is negligible.

Figure 4 show the calculated pressure change caused by casing deformation along the outer and the inner diameter of the seal. The shell thickness reduction decreases its stiffness and makes the pressure influence on the temperature more even. The pressure in the contact place of the sealing ring with the rod (along the inner diameter) is increased during the temperature reduction. Therefore, the seals with the movable contact along the inner diameter in order to reduce the friction and wear should have a slight assembly stretch.

Figure 5 shows the contact pressure variation in the seal at the temperature of 4-293 K in the absence

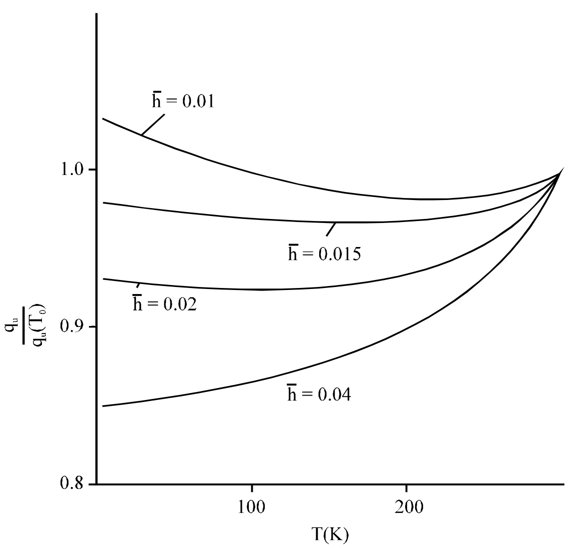

Fig. 3: Pressure variation, generated by the elastic element made of mr material, due to temperature

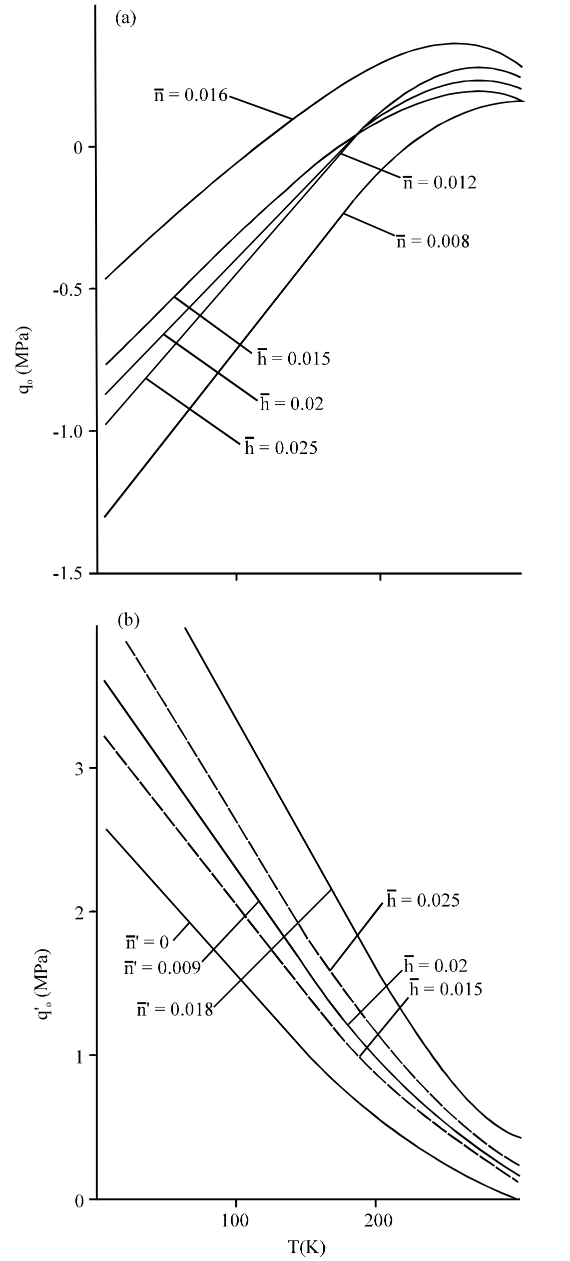

Fig. 4: Change of pressure, developed by casing, depending on the temperature along: a) the outer and b) inner diameters

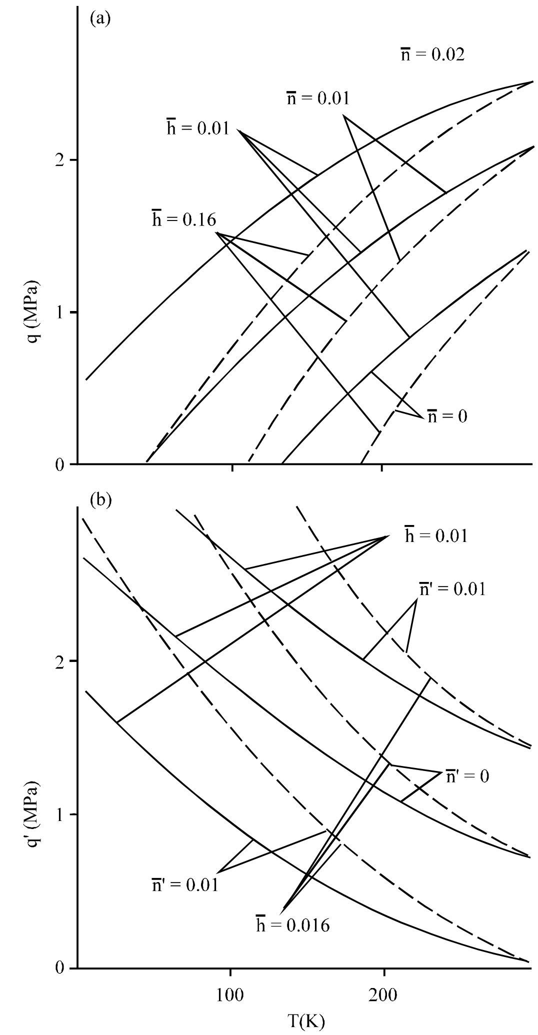

Fig. 5: Effect of temperature on the contact pressure change in the contact zone along: a) the outer diameter with a cylinder and b) the inner diameter with a rod

of sealed medium pressure. The presence of the butt belts in the shell leads to the uneven distribution of the contact pressure across the sealing ring width.

CONCLUSION

The main reason for the contact pressure change at the temperature reduction is the temperature deformation of the shell. To reduce this effect one should reduce the shell stiffness.

The considered radial movable seals the assembly of which is performed at normal temperature and the operating conditions correspond to the temperature of the sealed cryogenic medium, the mounting tightness in the movable contact area with the cylinder should be taken into account when it is reduced during cooling. In this case, the calculation of the mounting tightness should be performed during the conservation of tightness at a minimum temperature. In the case of the sealing ring movable contact with a rod along the inner diameter during cooling the tightness is increased. Therefore, the mounting tightness shall be minimal to limit the friction.

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