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MODELLING OF THE SEALING PROCESS IN HYDRAULIC CYLINDERS

The paper presents a model of the sealing process in kinematic pairs of hydraulic cylinders with elastic seals and an analytical form of this model based on the results obtained by the author. The prepared model distinguishes rheological parameters, allowing one to determine the criteria of a correct course of the sealing process and to forecast the operating time for the seals. Exemplary test results and their analysis are presented, too. It results from the analysis that leakage efficiency through the seal is dependent on the sealing pressure determined by the parameter δ, and it is unstable in relation to this parameter. Basing on this fact, the author determined conditions of hydrodynamic convection of the sealing and elaborated an analytical model of the sealing process including roughness of the piston rod surface as well as the seal flexibility.

1. Introduction

Sealing in kinematic pair strongly influencis its operating parameters, life and reliability.

Sealing in kinematic pair of hydraulic cylinder is realized by forcing an elastic element (seal) between the surfaces where the pressure chambers of hydraulic cylinder are separated. Phenomena occurring in the contact zone between these surfaces are called the sealing process.

A sealing kinematic pair belongs to the ageing objects. Ageing, or so-called wear, is a complex process, causing mass decrement, structural degradation of the material and changes of its properties influencing the

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sealing process. In this process, ageing appears as a progressive increase of leakage intensity. It can be understood as exhaustion of the sealing ability determined by the packing parameters and structural properties of the kinematic pair. Tests of such objects usually tend to determination of their usability time and formulation of a rheological model enabling life prediction.

One can distinguish static and dynamical leak tightness. The static leak tightness is characterized by a lack of leakages in the kinematic pair subjected to pressure under a relative rest of the sealed surfaces. The dynamical leak tightness can be observed when a relative motion takes place in the packing pressure zone.

Leakage capacity beyond the sealing chamber is a measure of the dynamical leak tightness.

Dynamical leakage intensity is strongly influenced by

- distribution and value of the packing pressure to the cooperating surfaces,
- microstructure of the cylinder surface,
- material elasticity and seal flexibility,
- wear degree.

The author has been working on the above problems for many years. This paper contains a synthesis of the results of the previous research work on life time [4], [5], [6], [7] and simulation of phenomena proceeding in the sealing process [8], [9], [10].

2. Structure and working parameters of the sealing kinematic pair

An exemplary kinematic pair is shown in Fig. 1. The essential sealing process proceeds in the area of contact of the sealing element with the surface of the relative motion (detail A). Packing pressure distribution in the sealing zone is shown in Fig. 2.

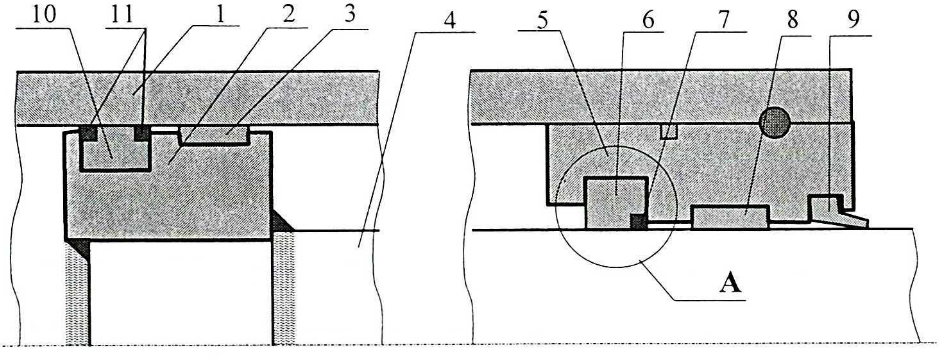

Fig. 1. Structure of sealing kinematic pair in a hyraulic cylinder 1 – cylinder, 2 – cylinder body, 3 – piston guilding ring, 4 – piston rod, 5 – gland body, 6 – gland sealing, 7 – preventive ring, 8 – gland guiding ring, 9 – striking ring, 10 – piston sealing, 11 – preventive rings

The packing is pushed into the seat (Fig. 2a) and then on the contact surface between the packing and the piston rod preliminary pressure $p_{nw}(x)$ occurs. It depends on the push value and shapes of the packing. The shapes of the packing profiles tested by the author are presented in Fig. 3 (the data and notation are given from the manufacturers' catalogues [11], [12].

Under the influence of the kinematic pair loading by pressure p (Fig. 2b), the packing undergoes a deformation, and it causes a certain distortion of the

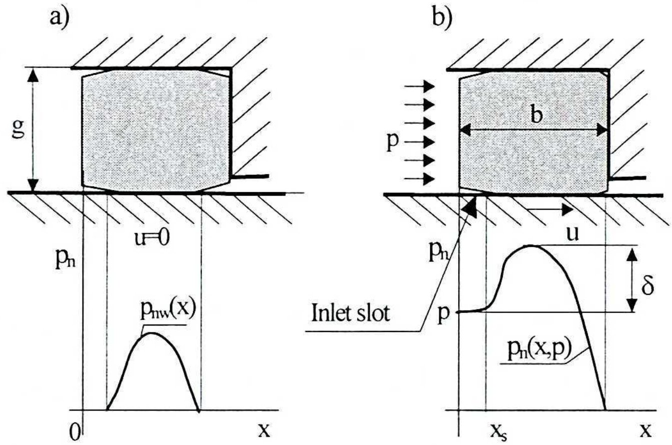

Fig. 2. Distribution of the packing pressure to the piston surface

stress field in the packing material, primarily obtained under the influence of the packing push into the seat and, at the same time, pressure p loading the kinematic pair overlaps that field and, as a result, pressure $p_n(x,p)$ occurs in

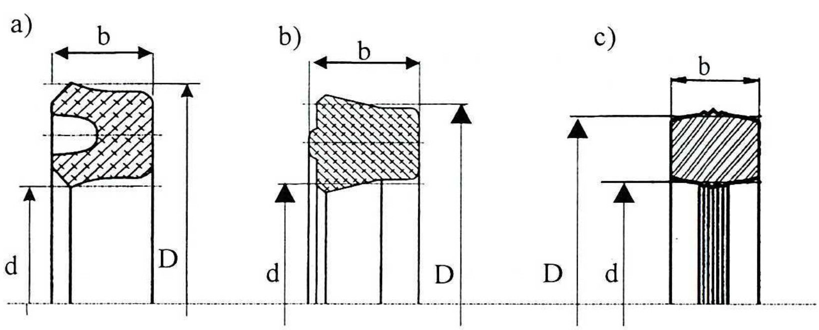

Fig. 3. The objects tested: a) packing U2 [11], b) packing U6 [11], c) packing US [12]

the contact zone. On the basis of $p_n(x,p)$, we assume a function of the sealing pressure distribution $\delta_n(x,p)$ determined by

$$\delta_n(x,p) = p_n(x,p) - p. \tag{1}$$

The extreme value of function $\delta_n(x,p)$ is assumed as a distribution parameter and it is written as δ (Fig. 2b).

Sealing pressure should ensure static leak tightness of the kinematic pair loaded by working pressure. The to-and-fro motion of the piston rod towards the sealing is conducive to the formation of a grease layer in the sealing zone and the occurrence of hydrodynamic pressure. All the phenomena occurring in the sealing zone cause leakages of the working liquid outside the sealed chamber, the so-called dynamic leakages. The intensity of dynamic leakages q is determined from

$$q = \frac{v_n}{n s \pi d_t} \tag{2}$$

where: n – number of working cycles, v_n – volume of leakages measured while n working cycles, s – piston rod stroke, d_t – piston rod diameter.

Friction and power dissipation are always connected with phenomena occurring together with the sealing process. The power dissipation intensity in the friction zone (w_T) is determined from measurements of the resisting force of the piston rod motion (F_T) according to the following relationship

$$w_T = \frac{F_T u}{\pi d_t b} = p_T u \tag{3}$$

where: u – piston rod velocity, b – sealing width, p_T – unitary friction force.

3. Tests on the sealing process and the sealing life time

3.1. A system approach to the sealing process

It is difficult to formulate a satisfactory physical model since the phenomena occurring in the sealing process are very complicated, and dynamic leakages take place. In such a case, it is convenient to consider the sealing kinematic pair as a system.

A scheme of the sealing process in a system approach is shown in Fig. 4. A set of input data (**V**) results from the given operating conditions. They can be determined from a cyclogram of the servomotor operation [5].

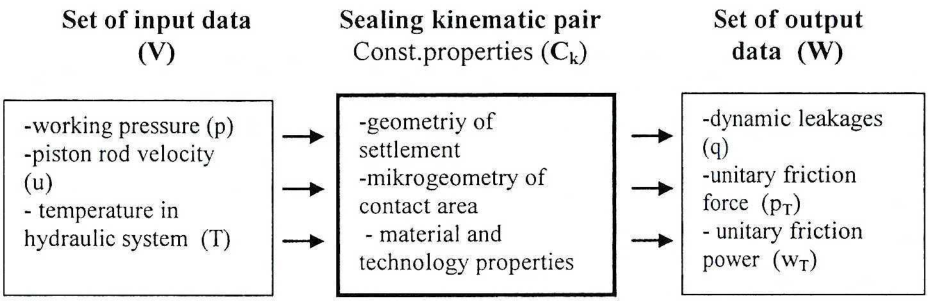

Fig. 4. Scheme of the sealing peocess in a system approach

A set of construction properties ($\mathbf{C}_k$) includes geometry of the sealing settlement in the seat, microgeometry of the contact area and material and technological properties of the kinematic pair elements, especially sealings. The total effect of construction properties selection is the function of sealing pressure distribution (1) and the parameter δ (see Fig. 2b).

A set of output data (**W**) includes the so-called kinematic pair operating parameters, i.e. dynamic leakage intensity and unitary friction force. According to Eq. (3), a product of unitary friction force and piston rod velocity determines intensity of power dissipation (w_T) in the friction zone for a unit of the friction area.

Considering the wear process occurring in the sets $\mathbf{C}_k$ and **W**, we can distinguish quantities dependent and independent on time understood as time of operation. On the assumption that the tested object is a "black box", the tests of the sealing process include a search of a relation between the properties of the set **V** (independent variables) and the set **W** (dependent variables). In this approach, the set $\mathbf{C}_k$ remains unexplicit and concerns a given object with properties resulting from the given conditions of the sealing settlement. For quantities shown in Fig. 4, the tests concern the following relationship

$$q = q\,(p, u, n);\ p_T = p_T(p, u, n);\ w_T = w_T(p, u, n), \tag{4}$$

where: n – a number of cycles assumed as a measure of operation time. Unitary friction power (the third of relationships (4)) is calculated from Eq. (3).

When the values of the set $\mathbf{C_k}$ are properly chosen, we can test influence of construction parameters on operating parameters. When the object life is tested, we can obtain interesting information about the process from the determined relationships between quantities in the sets $\mathbf{C_k}$ and $\mathbf{W}$ dependent on operation time. In the given case, we have such a quantity in the set $\mathbf{C_k}$ i.e. δ – the parameter of distribution function of sealing pressure (1), as the total effect of construction parameters, the centre loading by pressure and operating time. In the set $\mathbf{W}$, dynamic leakages q are dependent on operating time. In a consequence of the wear process, the parameter δ decreases and leakages increase. Now, the sealing process tests should include the following relationships

$$\delta = \delta(p, n) \text{ and } q = q(\delta, u). \tag{5}$$

A set of relationships (4) and (5) expresses a general model of the sealing process. An analytical description and determination of structural parameters of the model require much time and its realization needs some further simplifications. The influence of some selected variables on the tested phenomena was analysed [5,7] and it was found that a set of relationships essential for a model description of the process can be limited to an analysis of the relationship

$$\delta = \delta(w_T, p, n); \quad q = q(\delta, u); \quad w_T = p_T u = w_T(p, u). \tag{6}$$

Before determination of Eq. (6), it was assumed that in the correct sealing process the leakages were stable in relation to the parameter δ, i.e. the function relationship $q = q(\delta)$ takes place.

3.2. Analytical form of the model

The test results obtained for sealings of US type (Fig. 3c) in the range determined by Eq. (6) are presented in [6], [9], [10]. The following function relationships were assumed for the model description

$$\delta = \exp[-(a_1 n + a_2 p) + a_3], \tag{7}$$

$$q = \exp[-a_4(\delta - \delta_k) + a_5], \tag{8}$$

$$w_T = u^{a_6} p^{a_7} a_8, \tag{9}$$

where: $a_1 ... a_8$ – structural parameters of the model, δ_k – critical value of the sealing pressure related to tightness.

Among the model structural parameters, the parameter a_1 standing near the variable n (Eq. (7)) expresses the sealing wear intensity. From the previous tests of the sealing operating parameters [4], [6] it appears that the friction power as the parameter strongly influencing the wear intensity does not depend on the operating time, thus tests can be limited to a search of the relationship $a_1 = a_1(w_T)$. The results of such tests have been discussed in [7], [8], where the following power function was assumed for an analytical description of this relationship

$$a_1(w_T) = w_T^{a_9} a_{10}, \tag{10}$$

where: a_9, a_{10} – structural parameters of the model.
Exemplary results of tests of Eq.(10) are shown in Fig. 5.

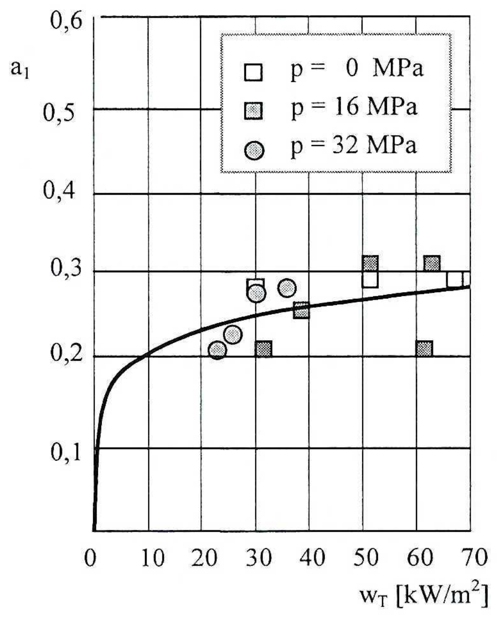

Fig. 5. Exemplary relation $a_1 = a_1(w_T)$

The parameter a_2 in Eq. (7) determines the influence of pressure loading the kinematic pair on the sealing pressure. The increase of the sealing pressure under the influence of pressure can be expected because of the material compressibility and the sealing deformation (the sealing takes the form of the seat). The parameter a_3 expresses the sealing pressure value after location of a new sealing in the kinematic pair which is not loaded by pressure. Exemplary results of tests on Eq. (7), [5] are shown in Fig. 6.

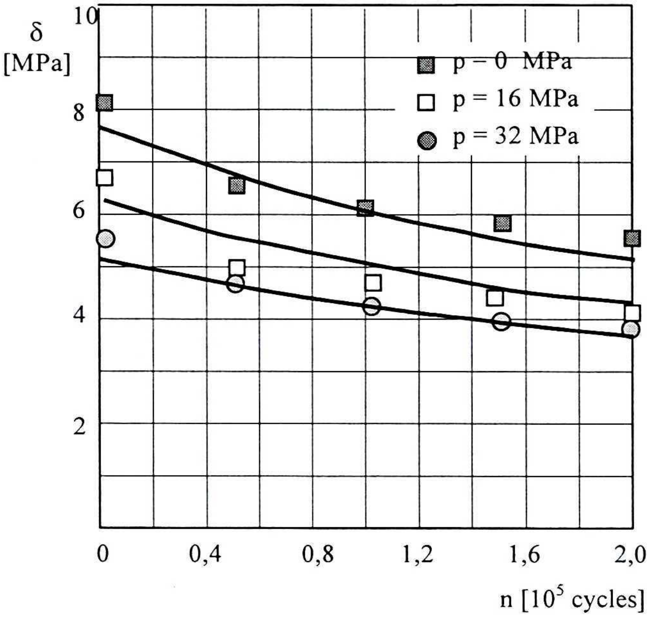

Fig. 6. Exemplary relationship $\delta = \delta(p, n)$

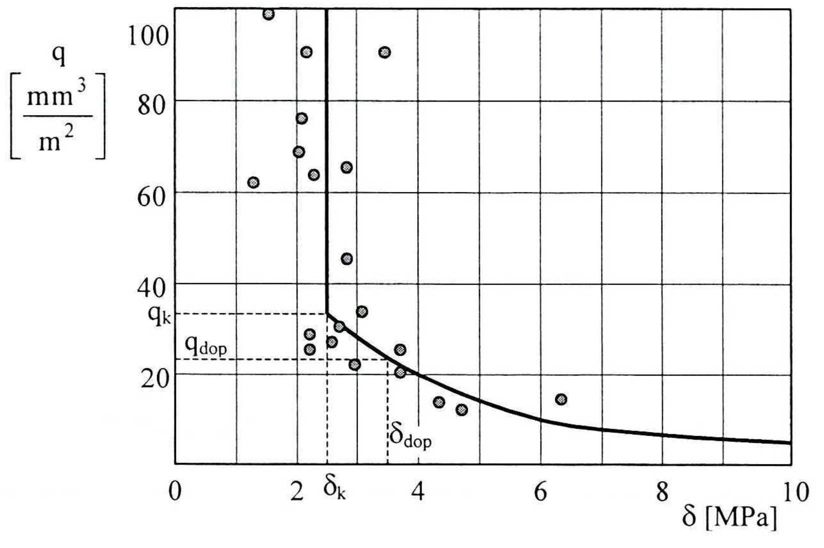

Fig. 7. Exemplary results of tests of the relationship $q = q(\delta)$

In the model (Eq. (8)), the other dependent variable is the dynamic leakage efficiency. From tests of this relationship, we can determine the character of the kinematic pair work related to leak tightness. From the analysis of the obtained test results it appears that the kinematic pair leak tightness depends on the parameter δ, and in this relationship for $\delta < \delta_k$ (see Fig. 7) we observe a high increase of leakages and their their unstability in relation to the parameter δ. It means that for a correct sealing process the sealing pressure $\delta > \delta_k$ is required. Exemplary results of tests of leakages and parameters included into Eq. (8) are shown in Fig. 7, [7].

Unitary friction power, the third dependent variable (Eq. (9)) is used for a direct determination of the sealing wear intensity coefficient expressed by Eq. (10). Exemplary test results concerning that relationship are shown in Fig. 8. Characteristics of such relationships for various sealing structures are discussed in [9].

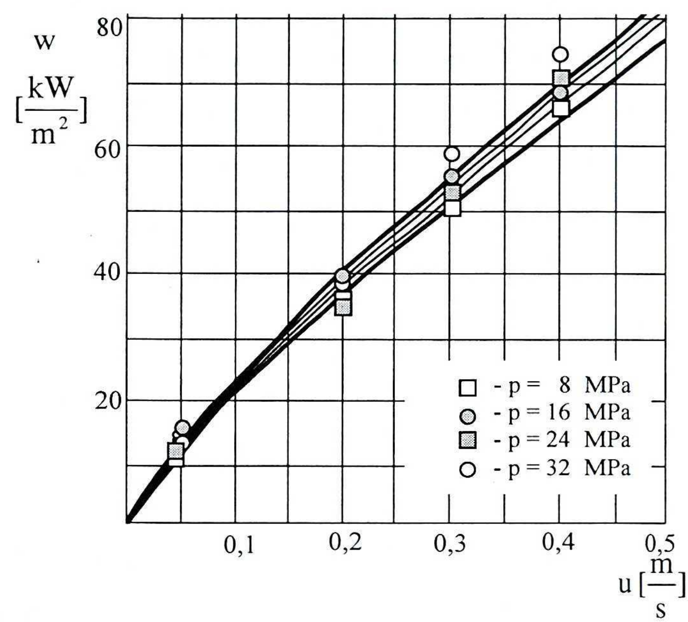

Fig. 8. Exemplary relationship $w = w(p, u)$

3.3. Life time prediction

Rheological character of the sealing process model (see Chapter 3.1) makes it possible to predict the kinematic pair behaviour under various operating conditions. In particular, it concerns the sealing life time prediction including requirements for leak tightness. Applying an example of the test result elaboration presented in 3.2 for n = 0 from Eq. (7), we obtain

$$\delta_{wp} = \exp(a_3 - a_2 p) \tag{11}$$

where δ_p – initial value of the sealing pressure in the kinematic pair loaded by pressure.
Assuming that total sealing wear takes place when the sealing pressure reaches δ (see Fig. 7), from Eq. (7) taking into account Eq. (11) we obtain

$$n_c = \frac{1}{a_1} \ln\left(\frac{\delta_{wp}}{\delta_k}\right). \tag{12}$$

Here, n_c determined by Eq. (12) means a number of cycles for which the full sealing wear occurs (uncontrolled increase of leakages).

Determination of the critical values q and δ for different sealing structures can be a base for application of a leak tightness criterion during the sealing selection. Let us assume that the leakage capacity determined with Eq. (8) reaches the critical value q at the pressure $\delta = \delta$. After transformations we obtain

$$\frac{q}{q_k} = \exp\left[a_4 \delta_k \left(1 - \frac{\delta}{\delta_k}\right)\right], \tag{13}$$

where: $\frac{q}{q_k}$ – relative value of leakages, $\frac{\delta}{\delta_k}$ – relative value of pressure.
Introducing measures of the leakage capacity and the sealing pressure in relation to their critical values, we obtain an universal characteristic of sealing in the form of Eq. (1). An example of Eq. (13) for the US sealings [14] is shown in Fig. 9.

In practice, requirements connected with leakage tightness can be different. If a value of permissible leakages q_{dop} is less than the critical value, we must predict a shorter time of operation resulting from reaching the minimum pressure δ_{min} corresponding to the permissible leakage value. Let us replace δ in Eq. (12) by a suitable value δ_{min} determined from Eq. (13) or from the graph (Fig. 9). Taking the given permissible value of the leakage capacity (q_{dop}) into account, we can determine the life including the criterion of the kinematic pair tightness.

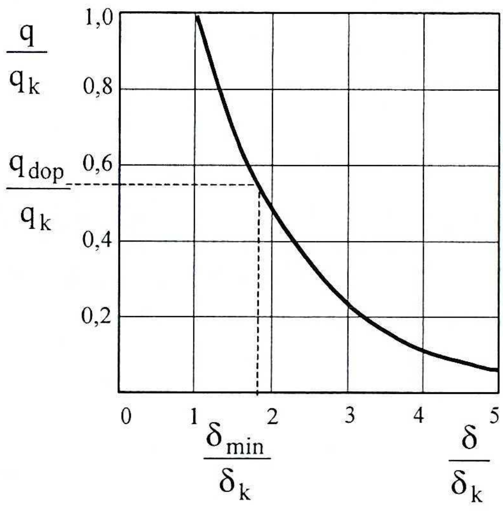

Fig. 9. Exemplary relations between relative leakages and relative sealing pressure

4. Hydrodynamics of the sealing process

4.1. Formation of the oil slot

While modelling the phenomena occurring together with the sealing proccess, the fluid friction model is widely applied (the so-called reversed theory of hydrodynamic lubrication [1]).

Fluid friction is the last phase of hydrodynamical phenomena development. As a consequence, conditions of full convection of the packing occur. These conditions need not be always fulfilled. It usually concerns transient states (starting and stoppage) typical for the servo-motor operation and limitations within the minimum thickness of the oil layer, dependent, among other things, on microstructure of surfaces forming the oil slot.

A possibility of the sealing kinematic pair operation during incomplete hydrodynamic convection of the package causes new conditions of leakage formation. That problem was mentioned in [8] after en analysis of dynamic leakage capacity (q) tests depending on the sealing pressure (parameter (δ), see Fig. 7). The observed unstability of leakages towards that parameter for $\delta < \delta$ can be understood as a result of quality changes in the sealing process when some parameters exceed suitable critical values.

Assuming that it is possible to observe the kinematic pair operation under uncomplete convection of the package, in [9] we can find the analysis of hydrodynamic phenomena development, taking into account the factors limiting formation of the oil macroslot under the package.

A scheme of the package contact with the piston rod (Fig. 2) including the surface microstrucure is shown in Fig. 10. It is assumed that in the system standstill ($u=0$) the package surface in the sealing pressure action zone represents microstructure of the piston rod surface (Fig. 10a). The interval $(0,x_s)$, where the sealing pressure does not occur or it is too small to prevent transport of pressure p between microirregularities, is an inlet slot where the hydrodynamic process is initiated. After the piston rod start, pressure increase in the inlet slot causes a process of hydrodynamic package convection above microirregularities tops (mechanical pressure is taken by pressure generated in the slot), and the oil macroslot forms and it separates surfaces of the package and the piston rod (Fig. 10b). The parameter h_m is the minimum height of the macroslot necessary for hydrostatic bearing of the package. That parameter is dependent on the piston rod surface microstructure of the piston rod surface, particularly on the height of inequality R_z ($h_m \approx R_z$). This process can be understood as hydrodynamic centre unsealing proceeding far inside the package pressure zone.

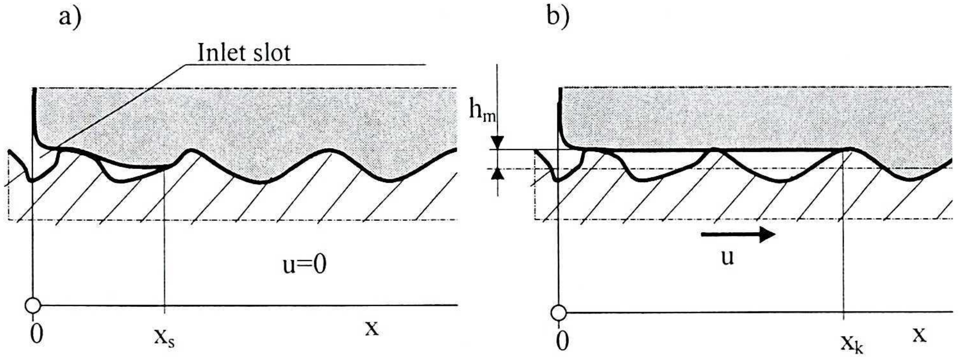

Fig. 10. Microgeometry of the contact between the package and the piston rod
a) package adhesion under the influence of pressure at rest,
b) macroslot formation after the piston rod start

From Fig. 10 it appears that unsealing of the next microirregularities is a condition for development of hydrodynamic phenomena. Approximately, we can assume that taking the real pressure δ by pressure formed in the slot includes its averaging and it causes the package surface "straighteningand filling" the space between microirregularities by a liquid. The mean height of the formed liquid layer in the slot (h_m) is the minimum height of the slot bottom conditioning development of hydrodynamic phenomena. Thus, we can assume that the bottom of the forming slot is not perfectly sharp and it strongly influences the value and the gradient of the generated pressure in the point x.

4.2. Development of hydrodynamic phenomena under the package

From the package pressure distribution (see Fig. 2) and Eq. (1) we assume the model distribution of the sealing pressure $\delta(x,p)$ presented in Fig. 11.

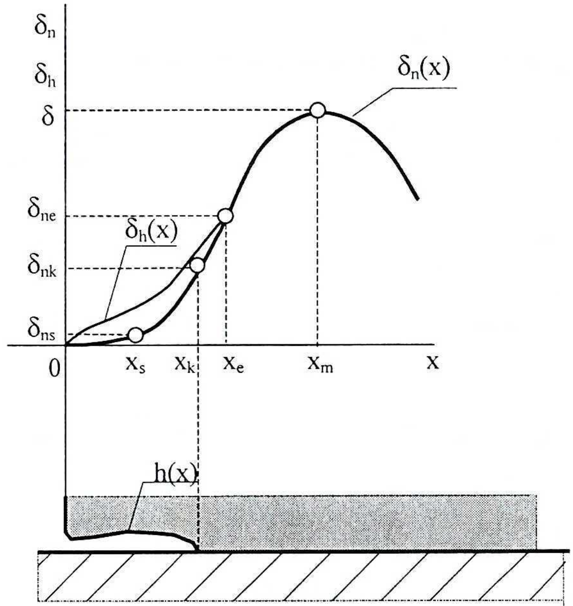

Fig. 11. Oil slot formation in the sealing zone

Taking into account the package hydrodynamic convection process being in point x at a given time, we assume that the phenomena proceeding in the slot can be described by the Reynolds equation for a unidirectional flow

$$\frac{1}{2}uh - \frac{h^3}{12\eta}\frac{dp_h}{dx} = q, \tag{14}$$

where: u – velocity of the relative motion of the rod piston, $h=h(x)$ – slot height, η – dynamic viscosity of the working liquid, $p_h(x)=p+\delta_h(x)$ – hydrodynamic pressure in the slot,. q – flow capacity through the slot. Realization of the flow capacity q (in this case, the right side of Eq. (14) means increment of the slot length. After dviding Eq. (14) by sides by h_m we obtain

$$\frac{uh}{2h_m} - \frac{h^3}{12\eta h_m}\frac{dp_h}{dx} = \frac{q}{h_m} = v_k, \tag{15}$$

where v_k – slot propagation velocity, h_m – slot height at $x=x_k$.

The given function of the sealing pressure distribution $\delta(x)$ determines the external conditions of the slot loading, and it is a base for determination of boundary values for the variable p_h in Eq. (15).

As it results from the previous assumptions for hydrodynamic package support and the minimum slot height (h_m.), the variables h and p_h in interval (0, x) must satisfy the following inequalities

$$h(x) \geq h_m; \quad p_h(x) \geq p + \delta_n(x). \tag{16}$$

Taking the package flexibility into account, we can determine the variable h with the following relationship

$$h = h_m + \frac{\delta_h - \delta_n}{c_u} \tag{17}$$

where c_u – package stiffness.

Eq. (15) expresses the hydrodynamic slot formation under the package. Depending on the external conditions, we can expect two different stable states of the centre operation:

- operation with the closed hydrodynamic slot ($v = 0$) in the point x located in the area of the sealing pressure action,
- hydrodynamic opening when the hydrodynamic process includes all the length of the sealing zone.

Assuming operation under the closed slot ($v=0$ in $x=x$), from Eq. (15) we obtain

$$\left(\frac{dp_h}{dx}\right)_k = \left(\frac{d\delta_h}{dx}\right)_k = \left(\frac{6\eta u}{h_m^2}\right). \tag{18}$$

The right side of Eq. (18) expresses the maximum pressure gradient which can occur at the bottom of the slot h_m in height. If the following inequality is satisfied

$$\left(\frac{d\delta_n}{dx}\right)_k \leq \left(\frac{6\eta u}{h_m^2}\right), \tag{19}$$

then the value of the derivative at the point x $\left(\frac{d\delta_h}{dx}\right)_k$ fits the derivative $\left(\frac{d\delta_n}{dx}\right)_k$ of the sealing pressure distribution at this point

$$\left(\frac{d\delta_h}{dx}\right)_k = \left(\frac{d\delta_n}{dx}\right)_k \tag{20}$$

and hydrodynamic unsealing of the kinematic pair takes place. Substituting the right side of Eq. (20) into Eq. (15), we can determine the slot propagation rate v. The rate v decreases as $\left(\frac{d\delta_n}{dx}\right)_k$ increases, and when it reaches a value equal to the right side of Eq. (19) it takes the value $v=0$, it means that the hydrodynamic slot growth stops. The described process is possible only when the pressure distribution function in the interval (0,x) is continuous and has a non-decreasing derivative $\frac{d\delta_n}{dx}$. When the process of hydrodynamic slot formation reaches the point x_e (the point of inflexion of the function δ), then the rate v_e determined fom Eq. (15) reaches its minimum value. It means that further course of the process, assuming (16), is disturbed because of the liquid deficiency. Thus, the function inflexion point $\delta(x)$ is a barrier for the hydrodynamic package convection $\left(\frac{d\delta_h}{dx}=0; \text{for } x > x_e\right)$. A further slot opening is dependent on the ability of the formed hydrodynamic slot to pressure increasiing at the slot bottom. Thus, in the point x_e there is a change of the hydrodynamic process of the slot opening determined with Eq. (20) into the hydrostatic process conditioned by the possibilities of pressure increase in the slot.

4.3. The threshold sealing model

The sealing pressure distribution depends on the package geometry. The assumption of distribution in form of the continuous function with the inflexion point results from classification to the pressure zone interval (0,x) (Fig. 2) determining the inlet slot length. In order to pay special attention to the hydrostatic process of the slot opening after the inflexion point of the pressure distribution function, we assume a simplified model of the pressure distribution with the pressure threshold in the inflexion point x_e (Fig. 12) determined by

$$\begin{aligned} \delta_n &= 0 \quad \text{for} \quad 0 \le x < x_e \\ \delta_n &= \delta \quad \text{for} \quad x \ge x_e \end{aligned} \tag{21}$$

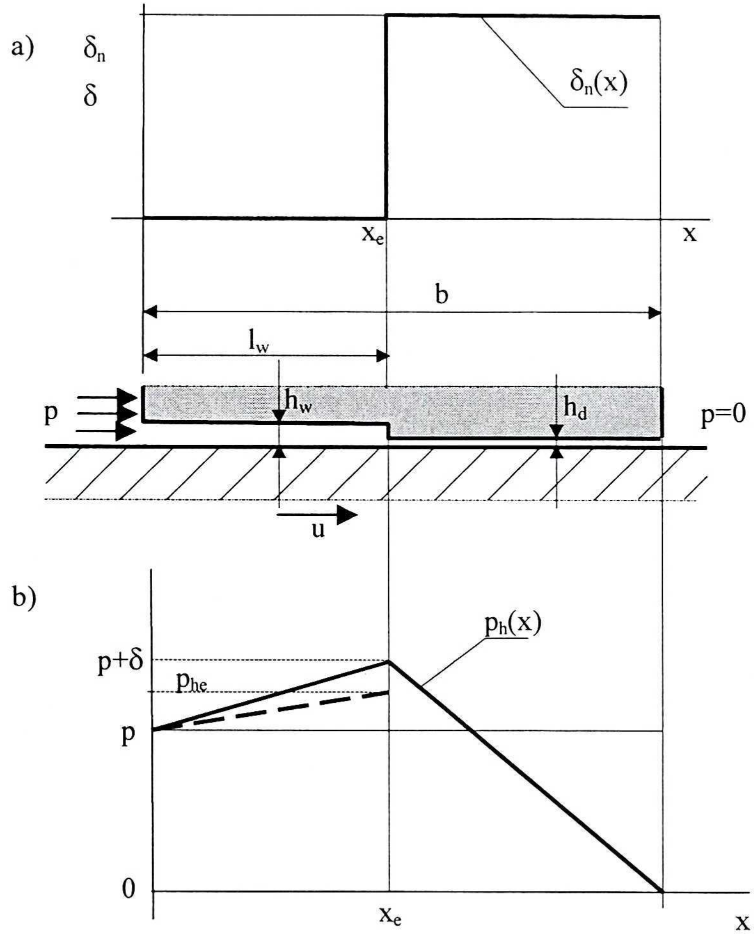

Fig. 12. Threshold model of the sealing: a) model pressure distribution, b) pressure distribution in the slot under the sealing

Under the above assumptions, the nominal slot length (b) can be divided into two parts. In the interval 0-x_e ther is a hydrodynamically active slot of thickness $h = h_{m.}$, where the pressure increases after the piston rod starting. At the same time, in the interval b – l, where pressure δ occurs, the slot is closed (h_d = 0) up to the moment when pressure δ_h at the point x_e reaches δ. Assuming a linear hydrodynamic pressure distribution in the slot and the corresponding threshold shape of the oil slot (Rayleigh model) and taking into account the package flexibility for $\delta_{he} < \delta$ (h_d = 0) we obtain

$$6\eta u - \left(h_m + \frac{\delta_{he}}{2c_u}\right)^2 \frac{\delta_{he}}{l_w} = 0 \tag{22}$$

Eq. (22) becomes invalid when $\delta_{he} > \delta$. A further pressure increase δ_{he} stops because of the hydrostatic clot opening at the interval b−1 (h_d>0) causing leakages outside. The opening height h_d can be determined from the equation of the flow through the slot, which can be expressed as

$$6\eta u\left(h_m + \frac{\delta}{2c_u}\right) - \left(h_m + \frac{\delta}{2c_u}\right)^3 \frac{\delta}{l_w} = 6\eta u h_d + h_d^3 \frac{p + \delta}{b - l_w} \tag{23}$$

The first term on the right side in Eq. (23) expresses capacity of the hydrodynamic flow through the slot at b − 1, and the other term expresses the hydrostatic flow capacity under the influence of pressure difference p + δ. However, because of the limiting conditions (16), the hydrostatic slot opening is possible when $h_d > h_m$, so we can assume that for the range $0 < h_d > h_m$ the second term of Eq. (23) is equal to zero. Then, we obtain

$$0 < \left(h_m + \frac{\delta}{2c_u}\right) - \frac{1}{6\eta u}\left(h_m + \frac{\delta}{2c_u}\right)^3 \frac{\delta}{l_w} < h_m \tag{24}$$

Eq. (24) determines the state of the sealing operation, which can be called a partial slot opening causing a stable character of dynamic leakages through the sealing. Under the opening height $h_d > h_m$ there is a full opening for which the leakage capacity increases additionally by the hydrostatic flow component (the second term of the right side of Eq. (23).

5. Conclusions

The author discussed a set of relationships, which should be taken into account while modelling the sealing process in the sealing kinematic pair of hydraulic cylinders. The performed tests were the base for formulation of the analytic model. The determined limiting values of the sealing pressure and dynamic leakage capacity made it possible to predict the operating time depending on the working conditions and requirements for tightness.

The presented simplified models of hydrodynamic phenomena development in the sealing kinematic pair allowed for determining the conditions of hydrodynamic package convection and the analysis validated the thesis that, depending on the working conditions and structural parameters, the package could work under the closed oil slot, a partial hydrodynamic opening ($0 < h_d < h_m$) and the full opening ($h_d > h_m$).

For description of the phenomena occurring in the oil slot new parameters were introduced. They vere parameters dependent on the surface microstructure (h_m) and the package stiffness (c_u).

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Modelowanie procesu uszczelniania w siłownikach hydraulicznych

Streszczenie

W pracy przedstawiono model procesu uszczelniania w węzłach uszczelniających siłowników hydraulicznych z uszczelnieniami elastycznymi oraz postać analityczną tego modelu opracowaną na podstawie wyników badań autora. W opracowanym modelu wyróżnia się parametry o charakterze reologicznym, które pozwalają na określenie kryteriów prawidłowego przebiegu procesu uszczelniania, oraz przewidywania czasu eksploatacji (trwałości) uszczelnień. Przedstawiono przykłady wyników badań oraz ich analizę, która wykazała, że wydajność przecieków przez uszczelnienie, jako wielkość zależna od nacisku uszczelniającego określonego za pomocą parametru δ ma charakter niestabilny względem tego parametru. Na tej podstawie określono warunki hydrodynamicznego unoszenia uszczelki oraz opracowano model analityczny procesu uszczelniania, w którym uwzględniono chropowatość powierzchni tłoczyska oraz podatność uszczelki.