UDC: 621.88/.89

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ASSEMBLY ACCURACY OF POWER CYLINDERS FOR POWERED ROOF SUPPORTS IN LONGWALLS

The article actualizes the issues of linkage precision in joints in hydraulic cylinders of powered roof supports at the stages of their design, manufacture and repair. The influence of the structural clearances on the position deviations of piston shaft under the action of external forces is shown. The linkage precision parameters for the joints being discussed are substantiated, and the methods to achieve these parameters in hydraulic props during their manufacture, assembly and repair are proposed. Based on the study of the effect exerted by the clearances in the cylinder–piston and bottom box–shaft joints on the position deviations of the shaft and on the level of the contact stresses initiated, the contact conditions of the mating parts are identified to minimize the stresses. The recommendations are developed to ensure the required sizes of clearances in the joints. For this purpose, such methods of precise linkage precision as selective assembly, assembly by the method of intergroup interchangeability and the method of adjustment using a piston as an immobile equalizer are considered. The main technological conditions for the applicability of selective assembly are established in terms of a hydraulic cylinder–piston connection with a diameter of 110H9/f9 in the serial production conditions. The calculated results and the ways to reduce the excess stock as a consequence and a main disadvantage of the selective assembly are presented.

Keywords: *powered roof supports, hydraulic props, life span, precision parameters, selective assembly, excess stock, incomplete parts, intergroup interchangeability, oversize dimensions, adjustment method, piston–equalizer*

DOI: *10.17580/em.2023.01.11*

Introduction

One of the critical objectives in the machine construction is validation of the accuracy parameters for joints to ensure the required assemblage quality at the proper performance characteristics both in terms of mechanisms and a machine as a whole. The quality of machine manufacturing has been addressed by many researchers in Russia and abroad [1–4], and these studies show how the manufacturing quality of parts can affect the service properties and the life of the items manufactured. In particular, in movable fitting, it is most critical to set and ensure the optimized surface roughness and clearance in joints [5–10].

Regarding power cylinders of powered mine roof supports in longwalls, it is especially important to meet the said objectives subject to complex geotechnical conditions of operation of powered roof support, asymmetrical external loading and roof vibrations. The latter results in unstable and discontinuous contact between the cylinder and piston and between the bottom box and shaft, and induces severe wear of the contact surfaces. The size of the joint clearances in the cylinder governs the position deviation of the shaft, the contact stresses of the parts and, eventually, their life span [10–12]. Given certain ratios of clearances and linear dimensions of parts, the actual loading patterns of hydraulic props may differ from the design loading (**Fig. 1**).

The design setting of the precision criteria for the cylinder–piston and bottom box–shaft links follows the cylinder machining technology and the economic considerations to optimize the labor content of manufacturing.

The earlier researches show that it is impossible to find the linkage precision parameters and to reveal their influence on the contact conditions, stress state and life span of joints in hydraulic cylinders from calculations of the static and contact strengths. Moreover, the static strength calculations totally neglect the linkage clearances [13], while in case of the contact strength calculation, the cylinder design is a very complex and labor-consuming engineering procedure.

The practice of such problem solving involves the approximation-based simulation modeling of contact interaction between joint parts at certain boundary conditions preset at the contact [14, 15].

It was succeeded in finding the linkage precision parameters and their influence on the life span of a hydraulic cylinder by means of optimization of clearances in these joints via the stress state analysis at the contacts of the joint surfaces (**Fig. 2**).

The study of the cross-effect of clearances in the cylinder–piston and bottom box–shaft links at Ø110*H*9/*f*9 finds out that when the

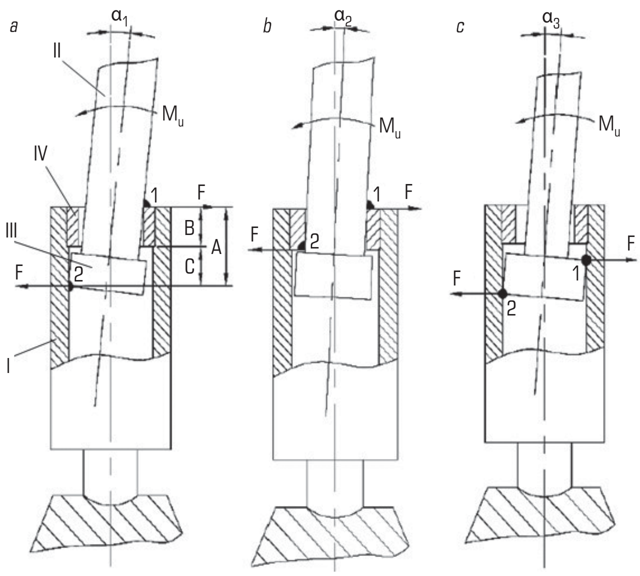

Fig. 1. Tilting of shaft in hydraulic prop:
a–the bending moment M_{bend} is taken up both by bottom box *IV* and cylinder *I*, then the force *F* is effective on the shoulder *A* (most favorable contact pattern); *b*–the bending moment is taken up exclusively and locally by bottom box *IV*, and the force *F* is effective on the shoulder $B < A$; *c*–the bending moment is exclusively taken by piston *III*, and the force *F* is effective on the shoulder $C < A$; *1*, *2*–points of contact; α_1, α_2, α_3–axial tilting angles of shaft *II* relative to the axis of cylinder *I*

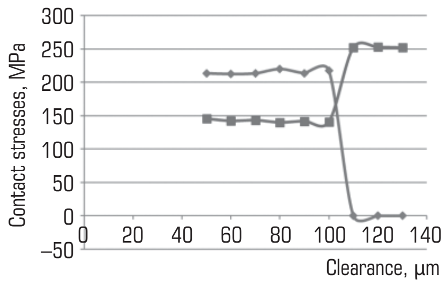

Fig. 2. Contact stresses versus clearance sizes [10]:

1–contact stresses between cylinder and piston, MPa; *2*–contact stresses between bottom box and shaft

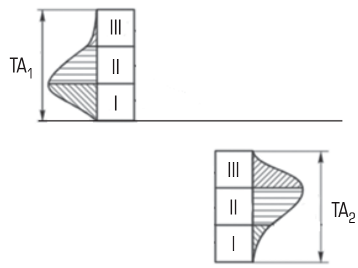

Fig. 3. Size distributions of cylinder and piston:

TA_1–range of cylinder tolerance; TA_2–range of piston tolerance; *I*, *II*, *III*–size groups

clearances between the cylinder and piston range as 50–110 µm, the position deviations of the shaft decrease to a minimum, which brings the optimal contact conditions in the joints, as per pattern *a* in Fig. 1. It is only possible to ensure such clearances in manufacture of the mating parts in accordance with N7 surface finish. The existing technology of hydraulic prop manufacturing only allows N9 surface finish, at economically validated tolerance and limit deviations. In this respect, we try to solve the set problem at the stage of assemblage of joints without essential changes to be introduced in the process of manufacture of the joint parts.

When assembling the pair of the cylinder (tolerance *H*9) and piston (tolerance *f*9), it is necessary to improve the surface finish by two points to ensure the seat *H7*/*f7*. Let us discuss the feasibility of improving the linkage precision by means of the selective assembly.

The deviations (errors) of the shape and surface in manufacture of parts arise from inaccuracy and deformation of a manufacturing machine or tool, nonuniformity of the machining allowance, inhomogeneity of materials, etc.

It is necessary to take these factors into account in sizing of parts before their linkage.

The multi-stage treatment of the inner surface of the cylinder includes such form errors as deviations of circularity, cylindricity and vertical profile. The current regulatory documents set the allowable form errors depending on the working accuracy [16].

It is recommended that a relative geometric relationship can have the normal *A*, increased *B* and high *C* degrees to define the form and size tolerance ratio subject to the final machining method.

Machining of the inner surface of a cylinder by boring and flaring ensures the relative geometric relationship of the degree *C*. The form tolerance or the location tolerance makes approximately 25% of the size tolerance.

The levels of the relative geometrical relationship in the regulatory documents allow the form and location tolerances less than 25% of the size tolerance in case of the more accurate methods of machining. Regarding cylinders, such method is the honed finishing.

Let us calculate the number of groups in the selective assembly with regard to the form deviation of the cylinder surface at the relative geometric relationship *C* and the reduced form tolerance.

It is not of interest at this time to study the form deviation of the piston surface which is in contact with the cylinder as the piston mates with the cylinder along tin guide paths pre-rolled in the special shape, or along nylon-6 split rings, and the length of the contact surface is 15 mm upon average.

We assume the group tolerances for the cylinder sizes (a_1) to be equal to the group tolerances for the piston sizes (a_2), i.e. $a_1 = a_2$, and consider the distribution laws of the size tolerances of the mating surface to be single-type.

The joint clearances should be not less than the minimal allowable clearances to prevent the shaft jamming, and not larger than the maximal allowable clearances to ensure functioning of the joint.

The surface of the cylinder having the inner diameter of 110 mm has N9 surface finish *H9* ($ESA_1 = 87$ µm; $EIA_1 = 0$ µm, $ITA_1 = 87$ µm), and the outer surface of the piston (shaft) has the tolerance *f*9 ($esA_2 = -36$ µm; $eiA_2 = -123$ µm, $ITA_2 = 87$ µm). The seat tolerance $TS = S_{max} - S_{min} = 174$ µm. The shape tolerances for the diameter in case of N9 surface finish are:

—20 µm at the relative geometric relationship *C* [16];

—8–10 µm in case of the honed finishing [17].

The cylinder and piston linkage should have the clearances *S* to feet with N7 surface finish and the seat *H7/f7* ($ESA_1 = 35$ µm; $EIA_1 = 0$ µm; cylinder tolerance $ITA_1 = 35$ µm; $esA_2 = -36$ µm; $eiA_2 = -71$ µm; piston tolerance $ITA_2 = 35$ µm). The ranges of the allowable clearances are: $Smin = 36 \le S \le S_{min} \le S_{max} = 106$ µm ($ITS = 70$ µm).

The complete interchangeability condition is: $ITA_1 + ITA_2 \le ITS$.

The limit group tolerance for N7 surface finish is:

$a_1 = a_2 \le (ITS + ITA_2 - ITA_1) / 2 = (70 + 35 - 35) / 2 = 35$ µm.

Furthermore, the group tolerances should be not more than the one/third of the clearance fit range ($a_1 = a_2 < ITS/3$). The decrease in the group tolerance reduces the probability of the mismatch parts in assembly of joints.

The values of the group tolerances should never be less than the form error tolerances.

In the flaring and honed finishing, the group tolerance a_1 is 30 and 12 µm, respectively. Considering probability of fitting of parts that conform with the limit values in the group, the group tolerance value is increased by 50%.

When it is impossible to set equal group tolerance, the different tolerance should be given to the last group for the aperture and to the first group for the shaft with regard to the asymmetry of the size distribution curve (**Fig. 3**).

This is connected with the fact that a machine tool operator usually obeys the lower limit of the tolerance for the aperture and the upper limit of the tolerance for the shaft. In this fashion, the resultant size of the aperture is closer to the minimum (nominal) and the resultant size of the shaft is closer to the maximum. The size pattern

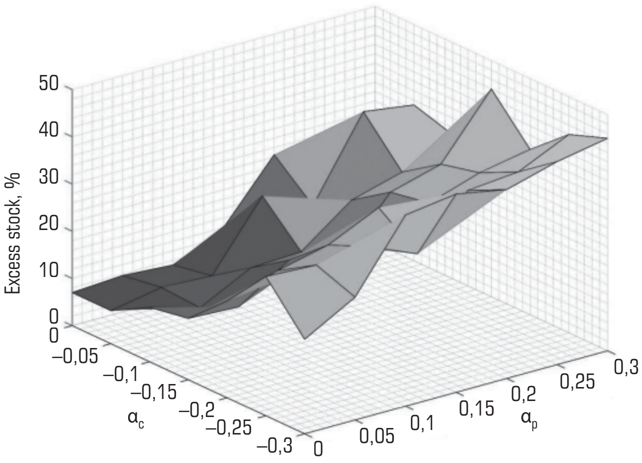

Fig. 4. Excess stock in selective assembly (three assortment groups): α_c, α_p–asymmetry coefficients of size distribution laws of cylinder and piston, respectively

asymmetry shows up also in the series production with numerically controlled machine tools, owing to the cutting tool wear.

Number of size groups *k* for the cylinder and piston is assumed to be as follows ($ITA_1 = ITA_2$):

$k_1 = ITA/a = 87/30 = 3;$

$k_2 = ITA/a = 87/12 = 7.$

Then the range of clearance fit is $ITS_1 = ITS/k = 58$ µm, $ITS_2 = 24$ µm.

Table 1 describes the resultant grouping of parts.

When cylinders can be completed with piston from any of the three groups, the maximal clearance exceed the allowable clearances for all possible alternatives of the group assembling. Thus, it is impossible to improve precision of the selective assembly by 2 points of the surface finish by means of boring and flaring (**Fig. 4**).

In the meanwhile, the intergroup interchangeability method by Professor Nabatnikov allows nullifying the size of the excess stock [9, 10, 18, 19].

The equipment procedure for the cylinders and pistons (7 assortment groups) at the maximal and minimal clearances to conform with the seat *H7/f7* is described in **Table 2.** Assembly of the parts from the other groups fails to ensure the two-points higher surface finish.

For the mismatched parts, it is possible to use one-point higher surface finish *H8/f8* (**Table 3**), or to re-finish the pistons to ensure a more precise joint.

With more groups of pistons to participate in completing a certain size group of cylinders, the probability of an excess stock lowers. On the other hand, the increase in the number of the selective groups increases the labor content of metrology operations in assortment of parts.

Precision of the cylinder–piston linkage remains a challenge during repair of hydraulic props, considering the big number of parts to be reconditioned. The main and most advisable method of repair is the cylinder face machining capable to remove any traces of wear and to recover the geometrical relationship of the working surfaces, lost during operation. In this case, the parts are transferred to an oversize and their interchangeability is violated. It is no more possible to carry out a selective assembly. In this case, it is more convenient and economic to ensure the linkage precision by means of assemblage and adjustment using an

Table 1. Size-based grouping of parts

Group tolerances $a_1 = a_2$, mm	Diametral sizes, mm		
	Cylinder apertures per groups	**Piston per groups**	**Clearance fit range *TS* (S_{min}; S_{max})**
0.030	1 group $Ø110_{0}^{0.030}$ 2 group $Ø110_{0.030}^{0.060}$ 3 group $Ø110_{0.060}^{0.087}$	$Ø110_{-0.123}^{-0.096}$ $Ø110_{-0.096}^{-0.066}$ $Ø110_{-0.066}^{-0.036}$	0.058 ($S_{max} > 0.106$) for all feasible alternatives of group assembly
0.012	1 group $Ø110_{0}^{0.012}$ 2 group $Ø110_{0.012}^{0.024}$ 3 group $Ø110_{0.024}^{0.036}$ 4 group $Ø110_{0.036}^{0.048}$ 5 group $Ø110_{0.048}^{0.060}$ 6 group $Ø110_{0.060}^{0.072}$ 7 group $Ø110_{0.072}^{0.087}$	$Ø110_{-0.123}^{-0.118}$ $Ø110_{-0.118}^{-0.096}$ $Ø110_{-0.096}^{-0.084}$ $Ø110_{-0.084}^{-0.072}$ $Ø110_{-0.072}^{-0.060}$ $Ø110_{-0.060}^{-0.048}$ $Ø110_{-0.048}^{-0.036}$	0.024
Initial precision of joint parts and clearances			
Ø110*H9*/*f9*	Ø110*H9*; $Ø110_{0}^{0.087}$	Ø110*f9*; $Ø110_{-0.123}^{-0.036}$	0.174 (0.036; 0.210)
Required precision of joint parts and clearances			
Ø110*H7*/*f7*	Ø110*H7*; $Ø110_{0}^{0.035}$	Ø110*f7*; $Ø110_{-0.071}^{-0.036}$	0.070 (0.036; 0.106)

Table 2. Piston and cylinder assembly at seat H7/f7

Group of assortment of cylinders	Group of assortment of pistons
1	3, 4, 5, 6, 7
2	4, 5, 6, 7
3	5, 6, 7
4	6, 7
5	7

Table 3. Cylinder and piston assembly at seat H8/f8

Group of assortment of cylinders Group of assortment of pistons	Group of assortment of cylinders Group of assortment of pistons
1	1, 2, 3, 4, 5, 6, 7
2	2, 3, 4, 5, 6, 7
3	3, 4, 5, 6, 7
4	4, 5, 6, 7
5	5, 6, 7
6	6, 7
7	7

immobile equalizer. The equalizer in the test joint is the piston. It is then unnecessary to stringent the geometrical relationship standards for the shapes of the mating surfaces, and the finishing machining operations remain the same as in the manufacture of the parts.

The proposed method of reaching the wanted precision of a clearance includes the machining of apertures of all cylinder at wider tolerances, first, and then, in accordance with the achieved precision of the apertures, machining of piston collars mated with the cylinder faces is carried out.

The applicability of the method is described as a case-study of the precise cylinder–piston joint Ø110*H9*/*f9* transferred to the oversize

Ø112$H9/f9$. The initial size tolerances of the cylinder and piston, to fit N9 surface finish, are assumed to be wider, without the increased standards of the geometrical relationships: Ø$112_{0}^{0.09}$ for the cylinder aperture (instead of the standard Ø$112_{0}^{0.087}$, Table 1); Ø$112_{-0.12}^{-0.03}$ for the piston (instead of the standard Ø$112_{-0.123}^{-0.036}$, Table 1). The latter is conditioned by the engineering capabilities of the repair workshops at mines, which are mostly equipped with the universal conventional machines or with the three-dimensional N/C machines.

Simultaneously, we tighten the clearance fit range Ø112$H9/f9$ from 0.174 μm to 0.12 μm subject to the conclusions [9, 10] which say that the reduction of the initial clearance of a joint, in particular, its upper limit, and, accordingly, the tolerance allows the higher quality and longer life of the friction couple (**Fig. 5**). The precision stability factor is given by:

$K_p = ITS_p/ITS$,

where ITS_p is the operating tolerance; ITS is the initial seat range.

Apart from the reconditioning of a hydraulic cylinder, it is possible to extend the life of the joint.

Thus, the dimension chain includes the clearance A_D, the cylinder aperture diameter A_1 with a wider tolerance, and the diameter A_2 of the piston which functions as an immobile equalizer:

$$A_\Delta = \overrightarrow{A}_1 - \overleftarrow{A}_2. \quad (1)$$

With regard to the adopted limit size deviations for the mating parts, the found clearance deviations are: the upper limit deviation $\Delta_\Delta^u = S_{max} = 0.15$ mm (for Ø112$H9/f9$, it is 0.21 mm), the lower limit deviation $\Delta_\Delta^l = S_{min} = 0.03$ mm (for Ø112$H9/f9$, it is 0.036 mm); then, the closing link tolerance:

$$T_\Delta = \Delta_\Delta^u - \Delta_\Delta^l = T_D = T_s = 0.15 - 0.03 = 0.12 \text{ mm},$$

where Δ_Δ^u and Δ_Δ^l are, respectively the upper and lower deviations of the closing link (clearance).

Table 4 gives the input data for the calculation subject to oversizing of the parts and setting wider tolerances for them.

The feasibility of adjustment with the immobile equalizers requires that the closing link tolerance T_Δ exceeds the equalizer size tolerance TA_2. In our case, this condition holds true.

The reduction in T_Δ to 0.12 mm at the existing tolerances of the design chain links unavoidably causes an error at the closing link, and it is necessary to compensate the error by means of adjustment using the link A_2 (piston) as an immobile equalizer.

The compensation T_k, which defines the largest required size of adjustment, is calculated from the formula [20]:

$$T_k = \sum_{i=1}^{i=m-1} T_i' - T_\Delta, \quad (2)$$

where T_Δ is the adopted tolerance for the closing link (for $IT8$); T_i' are the tolerances assumed to be widened and conformable with IT9 for the other links.

According to (2) and with regard to the data from Table 4, we have

$$T_k = \sum_{i=1}^{i=m-1} T_i' - T_\Delta = (0.09 + 0.09) - 0.12 = 0.18 - 0.12 = 0.06 \text{ mm}.$$

Then, we determine the number of the groups of the pistons–equalizers, N, necessary to compensate the error at the closing link:

$$N = T_k/(T_\Delta - T_{equal}) + 1, \quad (3)$$

where T_{equal} is the equalizer link tolerance which is:

$T_{equal} = T_2 = 0.09$ mm

in our case. Then, in accordance with (3), we have:

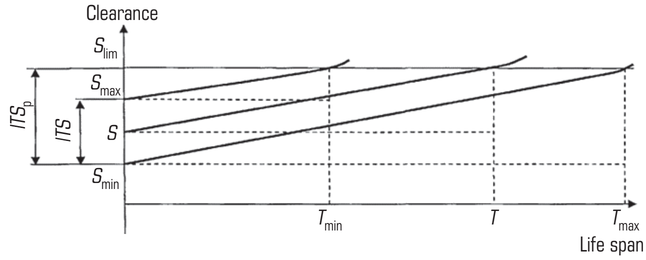

Fig. 5. Influence of initial clearance on life span of friction couple: *ITS*–clearance fit tolerance dependent on manufacture precision of mating parts and on assembly quality; S_{lim}–limit operating clearance; ITS_p–operating tolerance

Table 4. Input data

Dimension chain links	Limit upper deviation Δ_i^u, mm	Limit lower deviation Δ_i^l, mm	Size tolerance T_i, mm
A_Δ	0.15	0.03	0.12
A_1	0.09	0	0.09
A_2	–0.03	–0.12	0.09

Table 5. Limit deviations for pistons–equalizers

Limit deviations	Group I	Group II	Group III
Δ_k^u	–0.03	0	0.03
Δ_k^l	–0.12	–0.09	–0.06

$N = 0.06/(0.12 - 0.09) + 1 = 3$,

i.e., we need three groups of the pistons–equalizers.

The required equalization step P, or the difference in the dimensions of equalizers in the neighbor groups is given by:

$$P = (\Delta_1'^u - \Delta_1'^l)/N, \quad (4)$$

where $\Delta_1'^u$, $\Delta_1'^l$ are the limit deviations for the oversized apertures in the cylinder A_1.

At the present maximum allowable deviations $\Delta_1'^u = 0.09$, $\Delta_1'^l = 0$ and $N = 3$ as per (4), we obtain:

$P = (0.09 - 0)/3 = 0.03$ mm

In the problem under discussion, the equalizer of the first group is the piston A_2 which has the limit diameter deviations as follows: $A_2 = $ Ø$122_{-0.12}^{-0.03}$ (see Table 4).

The limit diameter deviations for the pistons in the rest groups are found by adding the limit deviations of the previous group piston with the compensation step P using the formulas:

$\Delta_{IIk}^u = \Delta_{Ik}^u + P = -0.03 + 0.03 = 0$ mm;

$\Delta_{IIk}^l = \Delta_{Ik}^l + P = -0.12 + 0.03 = -0.09$ mm;

$\Delta_{IIIk}^u = \Delta_{IIk}^u + P = 0 + 0.03 = 0.03$ mm;

$\Delta_{IIIk}^l = \Delta_{IIk}^l + P = -0.09 + 0.03 = -0.06$ mm.

Accordingly, for each of the three groups of the pistons–equalizers, we calculate deviations (**Table 5**). The dimension-based connection between the diameters of the machined cylinders D_1 and the diametral dimensions of the oversized pistons is described in **Table 6**.

We check the proposed method and the calculation of the oversize pistons–equalizers using the expressions:

$$\Delta_\Delta^u = \Delta_\Sigma^u + \Delta_k^u;$$
$$\Delta_\Delta^l = \Delta_\Sigma^l + \Delta_k^l. \quad (5)$$

Table 6. Connection between oversize piston and oversize cylinder aperture deviations D_1

Cylinder aperture deviation D_1, mm	$(\Delta^u - \Delta^l)_I$ 0–0.03	$(\Delta^u - \Delta^l)_{II}$ 0.03–0.06	$(\Delta^u - \Delta^l)_{III}$ 0.06–0.09
Group and diameter *D* of oversized roller	Group I $Ø112^{-0.03}_{-0.12}$	Group II $Ø112^{0}_{-0.09}$	Group III $Ø112^{0.03}_{-0.06}$

Table 7. Calculation checkout

Upper limit clearance deviation, mm	Lower limit clearance deviation, mm
Group I	
$\Delta^u_\Delta = \Delta^u_1 - \Delta^l_K = 0.03 - (-0.12) = 0.15$	$\Delta^l_\Delta = \Delta^l_1 - \Delta^u_K = 0 - (-0.03) = 0.03$
Group II	
$\Delta^u_\Delta = \Delta^u_1 - \Delta^l_K = 0.06 - (-0.09) = 0.15$	$\Delta^l_\Delta = \Delta^l_1 - \Delta^u_K = 0.03 - 0 = 0.03$
Group III	
$\Delta^u_\Delta = \Delta^u_1 - \Delta^l_K = 0.09 - (-0.06) = 0.15$	$\Delta^l_\Delta = \Delta^l_1 - \Delta^u_K = 0.06 - 0.03 = 0.03$

According to (5), in assemblage of joints using the pistons from oversize groups I–III, in the adopted range of the cylinder aperture deviations $(\Delta^u_1 - \Delta^l_1)_i$, the required deviations for the clearance are: the upper limit deviation $\Delta^u_\Delta = 0.15$ mm, the lower limit deviation $\Delta^l_\Delta = 0.03$ mm (**Table 7**).

The check has proved the correctness of the problem solution and the applicability of the proposed procedure in repair and renovation of hydraulic props.

One of the advantages of the adjustment procedure using the pistons as the immobile equalizers consists in the improvement of the linkage precision without the increase in the geometric relationship of the mating surfaces irrespective of the distribution laws of the cylinder aperture deviation and the number of parts to be reconditioned. In connection with this, even in repair of a small batch of parts, it is possible to avoid the excess stock as there is no mismatch of the parts.

Conclusions

- The authors have found the required precision for the cylinder–piston linkage (in terms of the joint Ø110*H*9/*f*9) to minimize the position deviations of the shaft and to ensure the most favorable contact conditions for the mating parts at the contact stresses within the allowable ranges. The joint clearances should range as 50–110 μm, which sets the manufacturing standard for the joint parts to conform with N7 surface finish;
- The required precision of N9 surface finish cylinder–piston linkage can be achieved in selective assembly based on the intergroup interchangeability at the more stringent requirements imposed on the form precision of the mating parts; otherwise, it is highly probable that the incomplete and mismatch parts appear;
- One of the efficient ways of having precise linkage of the cylinder and piston in the hydraulic prop repair is the use of the adjustment method with the piston to function as the immobile equalizer (finishing of the piston using the oversize cylinder). Implementation of this approach allows elimination of mismatched parts and, thereby, avoidance of incomplete repair and excess stock at any laws of deviation distribution in the recovered sizes of mating parts, and makes it possible to tighten the initial clearance and to extend thereby the life of the joint.

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