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DETERMINATION OF CHARACTERISTICS OF CIRCULAR DISTRIBUTOR VALVE PISTON PULP PUMP

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Summary. Prospective constractions for piston pumps are suggested and installation of distributive joint valve as well. Number of examples of parameters and characteristics experimentally have got, approximation dependences have found for their use in math modeling of the working process. It gives the possibility to lead research and completing on the stage of projecting on the way of numeral experiment on personal computer.

Key words: piston pulp pump, data of circular valve.

INTRODUCTION

Valve dependent is one of many joints in a hydraulic part of piston pumps which in many aspects define their power and exploitation indexes.

Among the all well known different dependents in pulp pumps which pulp over abrasive and other media here are used disk conic valves with a spring load and elastic pack which are situated either on anticline or on the plate of a valve (in this case removable anticline set on the body in conic manner).

With inlarging pulping up and pressure, the size and mass of the valve increasing (pump Y8-6M has the plate with diameter 222 mm and mass 9,5 kg) the high of lifting, because of unsuficient capacity and large persistence has diapason 30-40 mm for the powerful pumps while their regular work. All that leads to increasing of closing valve angle lagging up to 28^0,increasing power of pressure up to 15 kN and percussion attachment of these strengh at the moment of fitting on anticline, that not allow in some cases increasing pump high-speed and is main trouble of breakage as the valves and pressure-operated box as well [Braslavskiy B.I. and oth.; 1984].

Valves are the most vulnerable from the point of view of longevity joint of the pump. So, their construction foresee simple accessibility to the valves and approximately quick replacement of them. But with the same circumstances in the produced pumps deliver the large quantity of harmful volume while exceed useful volume in several times. IT leads how it will be showed further to essential decreasing of the feeding coefficient. Especially with the pressure growing up and gas containing in pumping up media, and conditions of loading of driving part.

Thus, the quantity of approximately harmful volume must be among the most important criterions pump pulp quality, while the valve joint and it setting in hydroblock provide as less it quantity as possible while access and replacement simplicity.

The second problem is in that that important to find hydraulic and other characteristics of new technical decision of valves and presentation them in acceptable for math models appearance.

OBJECTS AND PROBLEMS

The most radical direction of the first objective, as well as reducing the valve lift while increasing its capacity, is the use of annular valves whose flow with large values of the boundary of knocking is much higher than the plate [VI Pogorelov, 1967; Baryshnikov G. A. et al 1984]] In addition, under the same outer diameter and an annular poppet valve pressure forces the latter to the saddle will be 1,4 - 1,7 times smaller because of smaller active area of the locking body.

Prospective design of distribution sites of assembly for the suction and discharge valve unit [. AS 821741 (USSR), 1981; AS 1629597 (USSR, 1990], are shown in fig.1 and fig.2.

Distributor (fig.1) consists of the saddle 1, the interchangeable annular valve 2, helical springs 3, guide 4 and the elastic sealing rings 7-8, installed on the inside perimeter of cracks.

The upper, intermediate and lower parts of the saddle, forming channels in the suction and pumping H, are interconnected with ribs.

The interior of the valve, providing them with the direction of motion, is connected to the outer ring, too, with the ribs, which, not being force elements, virtually not create resistance to movement of the liquid.

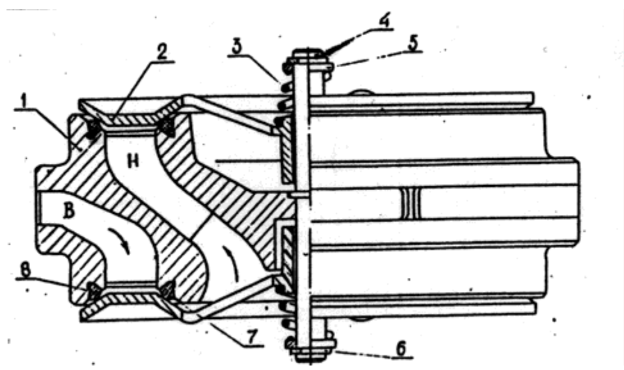

Fig. 1. The distribution node of the pump with annular valves and conical compression springs

The valve location on fig. 2, established in gidrobloke pump, two compression springs replaced by a spring tension, common to the suction and discharge valves. However, valve guides, each of which is attached to the outer side of one of the valves, and on the other side of the shank with a helical groove - one of the ends of the springs, they are placed in the inner boring of the saddle. Spring has a preload, so that both valves with non-operating pump compressed to the saddle.

Installation of a valve unit 1 (fig. 2) in the case gidrobloka 2 is a coaxial cylindrical chamber with a working window in its front wall having a hard bead, with the fixation in the axial direction with the help of hard 3 and the locking ring 4. Window in the shell closes the lid 5, fastening with two semirings-sections 6 and the locking ring 7. In this case the lid 5 is constantly underwent by a permanent mark force of the pressure and the need to install gidrozazhimov as on the covers of the suction poppet valves operated by a pump, is no longer, making it easier and cheaper design.

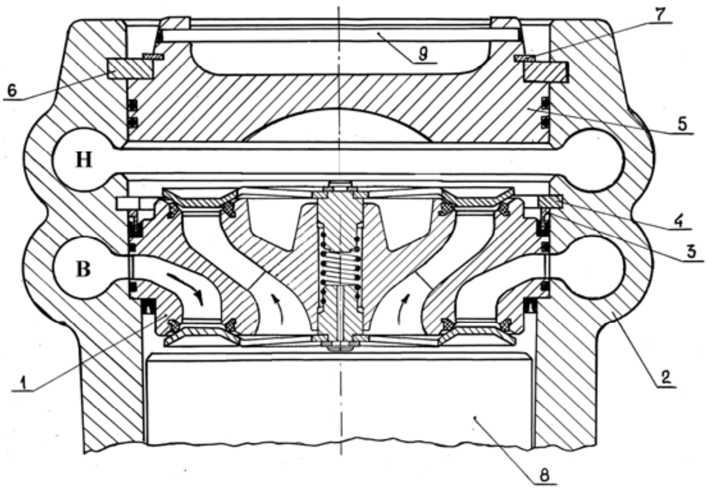

Fig. 2. Installing the distribution node with ring valve
in gidrobloke of the pump

Replacement of the developed valve unit is much faster and simpler than existing valves pulpovyh pumps, without the special puller for Extrusion saddles. In this case there is no damage to the mating surfaces shell gidrobloka and saddles.

Calculations and design study of the proposed valve for various pumps, including pump NBT-600 (boring three-cylinder pump power of 600 kW). The mass of the pump decreased in 1,2 times, and the quantity of harmful volume - 6 times and amounted to 2 liters. If the pump with serial valves relative magnitude of the harmful volume $\overline{W}$ at the maximum diameter of cylinder sleeves was equal to 1,886, and at a minimum - 4,244, the annular valve is equal to 0.3144 and 0.71, respectively. To compare the effectiveness of reducing the $\overline{W}$ expression of:

$$\eta_n = 1 - \beta_o p_o \left(1 - \overline{W}\right)\left(\frac{1}{p_b} - \frac{1}{p_н} \right), \quad (1)$$

was calculated coefficient of filing ηп pump at his work in the mode of self-suction, various volumetric gas content β_o fluid and the following conditions:

1. $d_п$ = 180 mm, p_o = 0,1 MPa, $p_в$ = 0.08 MPa, $p_н$ = 10 MPa,
2. $d_п$ = 120 mm, $p_н$ = 25 MPa at the same p_o and $p_в$.

Here: $d_п$ - diameter of the piston, $p_н$ - discharge pressure, $p_в$ - minimum pressure in the cylinder, p_o - atmospheric pressure.

The results of these calculations are shown in fig. 3, where the solid lines shows the $\eta_п$ pump with ring (1 - $\overline{W}$ = 0.3144, $p_н$ = 10 MPa; 3 - $\overline{W}$ = 0.71, $p_н$ = 25 MPa), and dotted - with serial poppet valves (2 - $\overline{W}$ = 1,886 , $p_н$ = 10 MPa, 4 - $\overline{W}$ = 4.244, $p_н$ = 25 MPa). It is easy to see that the coefficient of a pump with annular valve is much greater than that of the pump with serial valves

Deeper and more comprehensive data on performance and working process of the pump with annular valves can be obtained through a numerical experiment on a mathematical model which, for example, to install three-cylinder pump NBT-600 in supporting the pump is a system of 19 nonlinear differential equations of first order [Study, 1984].

A mathematical model of the pumping unit includes a number of dependencies, the analytical determination of which are virtually impossible. These include the hydraulic characteristics of valves, flow rates of force on the valve, the apparent mass of the valve and others that have found only experimentally

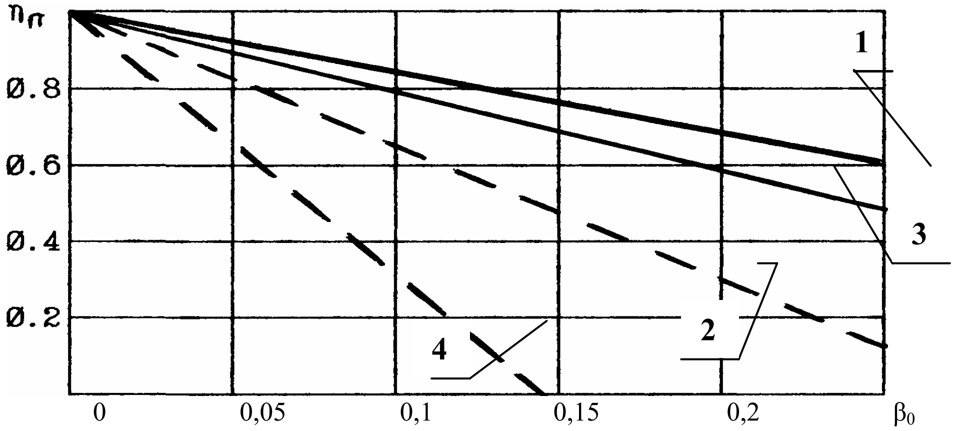

Fig. 3. Coefficient of a pump NBT-600 on the gas content of the pumped medium and the harmful volume of the chambers

For this purpose a special stand was designed, manufactured and installed, at which was installed simplified gidroblok model with the size of the ring valve and flow areas of channels, designed to pump the NBT-600, fig 4, the housing 1 of the gidroblok installed valve unit 3 with a suction ring 2 and the positive 4 valves. The last rigidly fixed bracket 7 with the rod 6 of the induction valve lift sensor 5, which is associated with the booster and sweep oscilloscope, not shown in the figure.

Determination of the coefficient of hydraulic resistance of the injection valve, for example, by direct water flow supplied by the centrifugal pump through the lower vertical pipe (this side of the piston of the pump is located) was carried out on the stand. Then the water passes through the channels of the saddle, opens the discharge valve on necessary height and exits through the upper pipe. With that, for each fixed flow valve lift height h and the pressure drop on it were measured . By the results of the experiments the coefficient of hydraulic resistance was determined from the formula:

$$\varsigma = \frac{\pi^2 d_c^4 g\left(\gamma_p - \gamma\right)h_p}{8\gamma Q^2} - 1 , \qquad (2)$$

where Q - water flow; h_p - pressure drop between input and output section of the valve (in meters of mercury); d_c - the average diameter of flow area valve seat; γ_p и γ - the proportion of mercury and water.

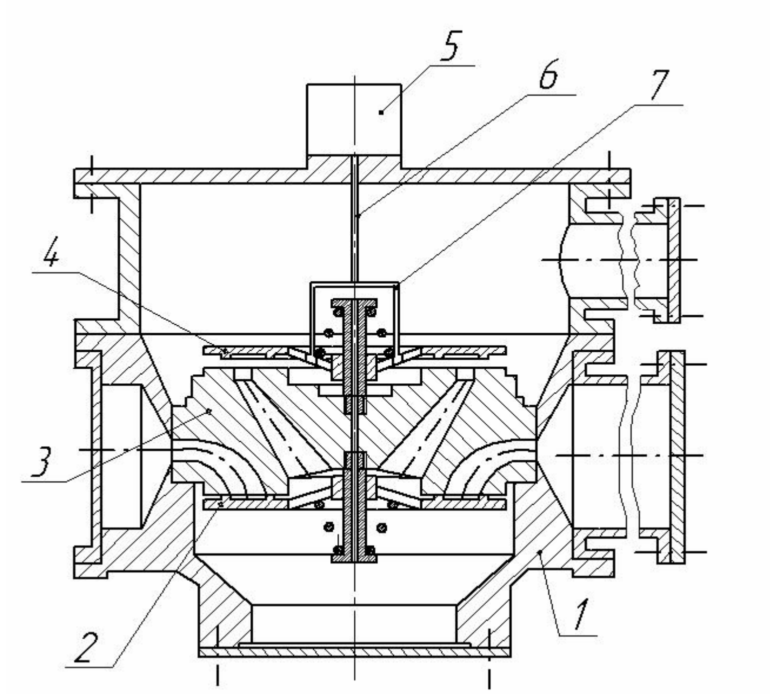

Fig.4. Gidroblok of the test stand with ring valve

For reverse current of liquid through the discharge valve, always having a place in running the pump under its guide pins are set alternately rings of different thicknesses, providing a variety of lift height of the valve. Water is supplied from the pump through the upper lateral pipe and then through the valve gap and channels of the saddle goes out through the lower vertical pipe. Measurement and calculations were performed as in the previous case.

For the flowings suction valve clamp 7 set up at him, and the rod 6 sensor was mounted on the inside and held up to the sensor through the holes in the valve guides. In other cases, these openings insulate gasket, installed between them.

According to the results of preliminary experiments, the coefficients of hydraulic resistance of the suction and discharge valves in direct current and the same height of their lift are very close to each other. So in the future was theirs one total dependence. The same applies to the reverse current

The determining factor in the coefficient of hydraulic resistance is the lift height of the valve relative to the seat [Kosenko, Belinsky Yu.A., 1982 - Research 1987]. Therefore our hydraulic characteristics of valves in the forward and reverse followings (fig.5) allowed approximating each of them with an expression depending on the relative valve height $\bar{h}$:

for direct current $$\varsigma = \frac{0{,}644}{\left(\bar{h}\right)^{1{,}61}} + 10{,}44, \quad (3)$$

reverse current $$\varsigma_o = \frac{4{,}5635}{\left(\bar{h}\right)^{1{,}21}} + 1{,}2, \quad (4)$$

The relative height of the valve lift

$$\bar{h} = \frac{h}{d_c}, \quad (5)$$

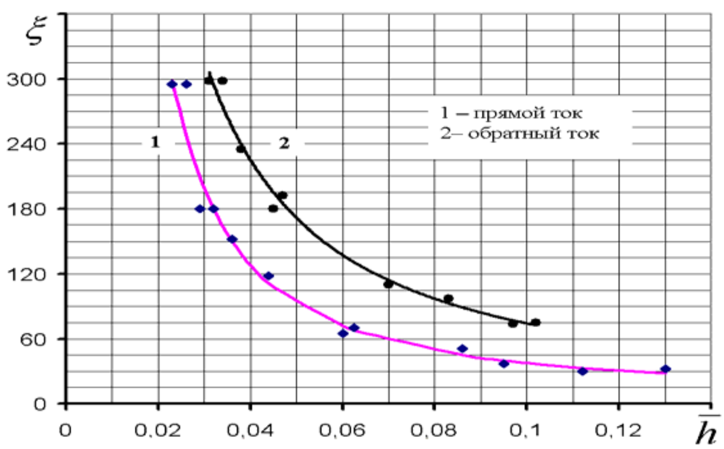

Fig. 5. Characteristics of the coefficient of hydraulic resistance of the annular valve

One of the most important issues in mathematical modeling of the work of the valves and the entire pump is to find the dependence of force with which the fluid act onto the valve in its motion and fluid flow in the forward or reverse directions. Analytical determination of that force is extremely difficult and not much convincing. As the works previously performed by the author of [Kosenko, Belinsky Yu.A., 1982 - Study of 1987] has shown, it should be presented with relation, which has the form:

$$F_{жс} = k\Delta p S_k .$$

Here k- being sought factor of the force interaction of fluid flow with the valve, taking into account the distribution of pressure on both its sides;

Δp - Pressure drop in the dimensional cross-sections before and after the valve; S_k - projected area of the valve on the plane normal to the direction of its motion.

Since the force acting on the valve from the fluid, equal to the force on a spring, ie , $F_{жс} = F_o + ch$, Where F_o and- c - known preload and stiffness of the springs, the coefficient of force of fluid flow on the valve is determined from the expression

$$k = \frac{4F_{жс}}{\Delta p \pi \left(d_{1k}^2 - d_{2k}^2\right)}, \tag{6}$$

where: d_{1k} and d_{2k} - outer and inner diameter of the ring valve.

In determining the k for the reverse flow there constructed and installed under the valve at its lower guide special spring c known characteristics.

Graphs dependencies $k = f\left(\bar{h}\right)$ and $k_o = f_1\left(\bar{h}\right)$ (in reverse flow) are presented in Fig .. 6, and their approximating expressions have the form:

$$k = 0{,}511\exp\left(-38{,}5\bar{h}\right) + 0{,}1, \tag{7}$$

$$k_o = 0{,}624\exp\left(-45{,}6\bar{h}\right) + 0{,}25. \tag{8}$$

When the body moves in a fluid, the part of the fluid is captivating by its body and has the same suspended speed and acceleration. That part of the liquid as if increases the mass of the body and is called the joint mass. With an unsteady motion of a body that takes place at the work of the valve, the phenomen leads to an increase of the operating forces of inertia on it .

For the experimental determination of the virtual mass $m_{пр}$ (or its coefficient $k_{пр}$) into the upper part of the saddle 3 (fig. 4) elongated guide screwed with the attached valve 4, a bracket 7 with the core of 6 of the sensor 5 and a special shortened spring with known stiffness c, which ends fixed to the valve and to the hard collar guide. At that the valve was in limbo at some distance from the saddle. In deriving it from the equilibrium valve performed their own damped oscillations with a period T in gidrobloke filled with water.

Then the total oscillating mass m is determined from the expression:

$$m = \left(T/2\pi\right)^2 c. \tag{9}$$

On the other hand

$$m = m_{пр} + m_л = m_л\,(k_{пр}+1), \tag{10}$$

$$m_k = m_{кл} + m_{сл} + m_c + 0.33\ m_{spring}. \tag{11}$$

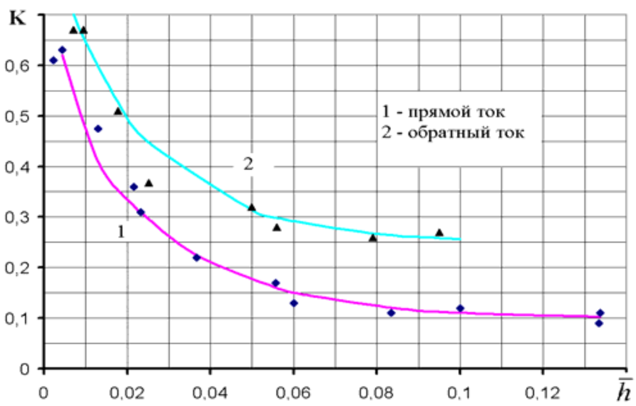

Fig. 6. Dependence of the force effect of flow onto the valve

Here $m_{кл}$, $m_{ск}$,, m_c and accordingly m_{spring} - mass of the valve, bracket, , rod and spring determined by weighing.

Using the data obtained and the expressions (9) - (11) the finding of $m_{пр}$ and $k_{пр}$ is not difficult.

It should be noted that in general, added mass depends on the geometry of the hydroblock and the valve position relative to its walls and the saddle. So, experiments were carried out at different equilibrium states of the valve relative to its walls and seat and the initial amplitude oscillations. Processing waveforms did not identify clear dependence $k_{пр}$ on the lift height of the valve. in the studied range (the distance between the saddle and the valve were varied from 6 mm to 22 mm) After processing the results of the experiments the factor associated mass annular valve was found $k_{пр}$ = 1,109, that 1,3 times as less than that of the serial poppet. The ratio reduced masses of the poppet and annular valve is 2.31, which also indicates the possibility in the need to improve operating speed of the pump with annular valves ..

Oscillograms of oscillations of the valve in hydroblocke filled with water represents graphical solution of the equation of its own vibration. This allowed to determine by the parametric identification methods the coefficient $k_т$ = 72 kg/s of viscous friction and dry friction forces in guideways $F_{тс}$ = 8.5 N. Their values were selected so as to ensure a minimum value of optimization criterion, as which it was chosen mean-squared deviation between experimental h_{ex} (taken from the waveform) and calculated h_{cal} move by the vibrations of the valve .

CONCLUSION

For the construction of a valve distributor which is suggested and which emerge itself erection unit for intake and plenum valves model with such valve joints have developed for one cylinder pump NBT-600. Possible in this stage conclusions have made and their result have provided which confirm the effectiveness of it using in the pump. Possible on this stage calculation is made and their result have provided which confirm the effectiveness of it using in the pump.

The stend have projected, made and assembled where the physical experiment was conducted to get experimental dependences of hydraulic resistance coefficient of circlular valves and influence of a stream on them while direct and indirect movement of fluid stream through the valves.

Numerical means of coefficients have found of the attached mass of valves and tough frictions in directions and power of dry frictions which are necessary movements equations.

Empiric expressions have got for pointed characteristics which are very comfortable for the use in math modeling of working process of piston pulp pump. Completion of numerical experiment on computer gives it a way to refused from large and expensive physical experiment as early as at the stage of projecting and to make research and completion of piston pulp pump that leads to cutting expenses on their development and industrial assimilation.

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ОПРЕДЕЛЕНИЕ ХАРАКТЕРИСТИК КОЛЬЦЕВОГО КЛАПАННОГО РАСПРЕДЕЛИТЕЛЯ ПОРШНЕВОГО ПУЛЬПОВОГО НАСОСА

Косенко-Белинский Ю.А.

Аннотация. Предложены перспективные для поршневых насосов конструкции и установка кольцевых клапанов в гидроблок. Экспериментально получены ряд их параметров и характеристик. Найдены аппроксимационные зависимости для последних в удобной форме при их использовании в математических моделях рабочего процесса насоса. Численный эксперимент на математических моделях с помощью ЭВМ позволит проводить исследования и доводку насосов на стадии проектирования, что приведет к сокращению сроков и расходов на их разработку и производство.

Ключевые слова: поршневой пульповый насос, клапанный распределитель, характеристики кольцевого клапана.