IOP Conf. Series: Materials Science and Engineering **779** (2020) 012043 doi:10.1088/1757-899X/779/1/012043

Analysis of force factors affecting valve fitting process at the saddle

S Semenov

Bauman Moscow State Technical University

E-mail: a1e10@rambler.ru

Annotation. The analysis results of the action of various force factors that slows the hydraulic valve when fitting on the saddle are presented. Manifestations of inertia forces in a thin layer of liquid located in the gap between the valve and the seat immediately before the closing of the gap are shown. The components of inertia forces are distinguished, which differ in the mechanism of occurrence. The degree of their influence on the fitting process in relation to the viscosity forces is shown.

Introduction

Valves are ubiquitous in hydraulic equipment. The valve assembly in volumetric pumps with valve distribution is one of the most important and, as a rule, one of the most problematic. Despite the fact that the pump mechanisms can be very complex [1], it is often the valves that determine the ultimate capabilities of such pumps. Irrationally designed or faulty valves cause increased noise, reduced volumetric efficiency, and reduced pump life. Valve assemblies are widely used in other types of machines and systems [2, 3, 4, 5].

Most of the problems are somehow related to the process of seating the valve on the saddle. In volumetric pumps working with liquid at high pressure, valve parts are made of metal or high-strength hard plastic. High strength ceramics are sometimes used. At the same time, at pressures of tens of MPa, metals become practically uncontested. The tightness of the closed valve in this case is achieved due to the accuracy of the closable surfaces and the density of their fit after closure.

At the moment of approach to the saddle, the valve moves with a finite speed, and if it is not slowed down, then when solid surfaces are closed at significant speeds, shock forces inevitably arise. If the valve and saddle are made of hard materials with high density, the shock pulse may be short, but very large in amplitude. This leads to significant mechanical stresses, which, acting cyclically, destroy the valve and seat.

The design of valve assemblies used in volumetric pumps is quite diverse. In particular, there are many design methods known to reduce impact forces during fitting. However, as the pressure and operating frequency of the pump increase, the choice of design solutions narrows. At a pressure approaching 100 MPa, the shut-off element becomes either spherical or takes the form of a cone or cylinder, while the material must be strong, rigid, and the movable element should also be as light as possible. As a result, it becomes difficult to add special elements to the structure that damp the valve during fitting, as well as use materials that effectively dissipate impact energy in the structure.

To a large extent, a rational choice of the design parameters of the valve assembly, in which the damping ability of the liquid located in the gap between the joined surfaces is effectively used, allows to solve the problem. For this, it is necessary to correctly understand how various force factors appear in a thin layer of liquid between the valves at the fitting moment.

Often, if not in most cases, engineers use generalized experimental data and recommendations set forth in [6] or similar approaches when designing valves of volumetric pumps. The prevailing point of view is that at the end of the stroke, the valve is stopped due to the viscous friction forces of the fluid displaced from the closing gap under the valve. Is this always the case? What is the scale and origin of the forces caused by hydrodynamic processes in the fluid displaced from under the valve? Answers to these questions will allow a more conscious approach to valve design and improve their performance.

Analysis

To obtain the most common and visual analysis results, we will adopt an extremely simplified design scheme and accept the following assumptions:

- It will be considered that all surfaces are flattened;
- the liquid layer in the closable gap is thin (its size is small compared to the width of the sealing girdle);
- It will be accepted the parabolic law of the velocities distribution over the thickness of the extruded layer, as with a steady laminar flow in a thin gap;
- It will be considered that the sealing girdle narrow, that is, its width is much smaller than the hole radius in the valve seat, which allows us to neglect the curvature of the girdle;
- The fluid pressure flow through the closing gap is not taken into account under the influence of the differential pressure on the gap;
- The fluid flow displaced from the gap is considered to be completely symmetrical relative to the middle of the girdle;
- Fluid compressibility is not considered.

As a result, the valve during the fitting process can be represented as a rectangular prism (bar), falling to the plane and displacing the liquid from the gap between its lower surface and this plane. Due to symmetry, we will consider the half-bar with respect to the sealing girdle of width moving along the vertical plane passing through the middle of the girdle. Accordingly, there is no flow through this plane. There is also no fluid leakage at the ends of the gap. The design scheme is shown in Fig. 1.

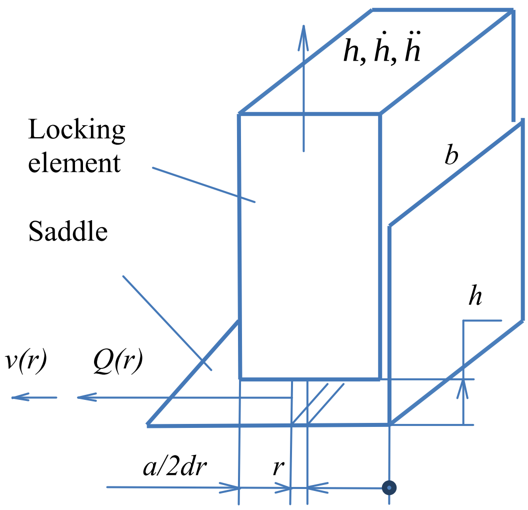


Fig. 1. The design scheme

The flow displaced from under the bar, depending on the distance from the vertical plane in the middle of the girdle is calculated as follows:

IOP Conf. Series: Materials Science and Engineering **779** (2020) 012043 doi:10.1088/1757-899X/779/1/012043

$$Q(r,\dot{h}) = -b\dot{h}r$$

Where$b$— length of the girdle;

r— Distance from the middle of the girdle;

$\dot{h}$—bar speed taking into account the sign (Fig. 1).

Then the average current velocity depending on the distance from the middle of the girdle, height and bar speed is calculated:

$$v_{cp}(r,h,\dot{h}) = -b\dot{h}r / bh = -\dot{h}r / h$$

Then the hydraulic friction losses in the gap in a section of length dr at a distance r from the middle of the girdle are calculated (having adopted, as a first approximation, the parabolic law of velocity distribution)

$$dp_{mp} = -\frac{k_{фэ}12\mu}{bh^3}Q\,dr = \frac{k_{фэ}12\mu}{h^3}\dot{h}r\,dr = \frac{k_m}{h^3}\dot{h}r\,dr$$

Where$k_{фэ}$— correction factor that allows (if necessary) to take into account the deviation of the structure of the real flow in the gap from that adopted in the first approximation (at this stage it is taken equal to unity);

$k_т$— equivalent resistance coefficient;

$$k_m = k_{фэ}12\mu$$

μ— Dynamic viscosity coefficient.

Acceleration of fluid in the gap (at average speed)

$$\frac{d}{dt}v(r,h,\dot{h}) = \frac{\partial v}{\partial r}\dot{r} + \frac{\partial v}{\partial h}\dot{h} + \frac{\partial v}{\partial \dot{h}}\ddot{h} = -\frac{\dot{h}}{h}\left(-\frac{\dot{h}r}{h}\right) + \frac{\dot{h}r}{h^2}\dot{h} - \frac{r}{h}\ddot{h}.$$

Therefore, when moving both up and down

$$\dot{v} = \frac{2r\dot{h}^2}{h^2} - \frac{r\ddot{h}}{h}$$

The pressure increment in the gap in the dr section is due to the inertia of the fluid, expressed as a function of the distance r from the middle of the girdle during accelerated movement of the valve and the fluid in the gap

$$dp_{ин} = -\rho\dot{v}dr = -\rho\left(\frac{2r\dot{h}^2}{h^2} - \frac{r\ddot{h}}{h}\right)dr = -\left(\frac{2\rho r\dot{h}^2}{h^2} - \frac{\rho r\ddot{h}}{h}\right)dr.$$

The total pressure increment in the dr section as a function of r

$$dp(r) = \left(-\frac{2\rho\dot{h}^2}{h^2} + \frac{\rho\ddot{h}}{h} + \frac{k_m\dot{h}}{h^3}\right)rdr.$$

After integration, we obtain the law of pressure distribution over the girdle width

$$p(r) = \frac{1}{2}\left(-\frac{2\rho\dot{h}^2}{h^2} + \frac{\rho\ddot{h}}{h} + \frac{k_m\dot{h}}{h^3}\right)(r^2 - \frac{a^2}{4}),$$

Where a— girdle width.

The maximum pressure in the gap (in the middle of the girdle)

$$p_{\max} = -\frac{a^2}{8}\left(-\frac{2\rho\dot{h}^2}{h^2}+\frac{\rho\ddot{h}}{h}+\frac{k_m\dot{h}}{h^3}\right)=$$
$$=\frac{\rho\dot{h}^2a^2}{4h^2}-\frac{\rho\ddot{h}a^2}{8h}-\frac{k_m\dot{h}a^2}{8h^3}.$$

The physical meaning of the terms of the last expression can be interpreted as follows:

The first term is called the convective component of pressure. It is equally determined by two phenomena: firstly, it is, in fact, a manifestation of the velocity head of a fluid moving in the gap; secondly, this is a manifestation of the inertial pressure from the acceleration that occurs when the fluid moves from a region with a lower velocity near the middle of the girdle to a region with a higher velocity near its edge. It depends squarely on the speed of lowering the valve.

The second term is called the reduced pressure component. It is determined by the inertia forces caused by the acceleration component of the fluid, which arises due to the accelerated movement of the valve (the speed of the displaced fluid is kinematically related to the speed of the valve). This term linearly depends on the acceleration of the valve.

The third term is called the viscosity component of pressure. It linearly depends on the speed of movement of the locking element and is determined by the viscosity component of the hydraulic resistance of the gap.

The vertical force (composed of the forces acting on both half-belts) is obtained by integrating the pressure over the area of the belt

$$P = 2\int_0^{a/2} p(r)b\,dr = \frac{2b\rho a^3\dot{h}^2}{12h^2}-\frac{b\rho a^3\ddot{h}}{12h}-\frac{k_m b a^3\dot{h}}{12h^3}$$

Note that the inertia forces appear in the fluid squeezed out of the thin gap, appear even with uniform movement of the valve. In this case, the second term of the last expression is zeroed, but the first remains. Moreover, even with an insignificant viscosity of the liquid and, accordingly, the smallness of the third term, a force preventing the closure of the gap will still arise, and its value will be determined by the first term.

Experiment

To confirm the above calculations, as well as to determine the degree of influence of the listed physical effects on the valve fitting process, a computational and physical experiment was conducted.

A computational experiment was carried out by numerically solving the differential equation of motion of a spring-loaded valve, obtained on the basis of the above expressions

$$m\ddot{h} = \frac{2b\rho a^3\dot{h}^2}{12h^2}-\frac{b\rho a^3\ddot{h}}{12h}-\frac{k_m b a^3\dot{h}}{12h^3}-mg-k_{mp}\dot{h}-c(h+h_0)$$

or

$$(m+\frac{b\rho a^3}{12h})\ddot{h}+(\frac{k_m b a^3}{12h^3}+k_{mp})\dot{h}-\frac{2b\rho a^3}{12h^2}\dot{h}^2+c(h+h_0)=-mg$$

Where h_0— initial spring preload;

c— spring stiffness;

$k_{\text{тр}}$— coefficient of viscous friction when the locking element moves along the guide.

In this equation, it is worth paying attention that an additional term is added to the mass of the valve, which expresses the reduced mass of the liquid squeezed out of the closable gap and increases as the valve lowers.

The initial data (parameters and initial conditions) adopted during the simulation correspond to the parameters of the experimental setup:

m=0,142 кг;	a=0,006 м;
ρ=1000 кг/м^3;	b=0,157 м;
$k_м$=12;	$k_{тр}$=13,1 Н·с/м;
c=1226 Н/м;	h_0= 0;
h(0)=0.01 м;	$\dot{h}$(0)=0 м/с.

At the initial moment, the valve is raised above a flat surface to a height h (0) and accelerates downward under the action of a spring and gravity. As it approaches the supporting plane, it is slowed by the liquid layer (water) between the flat girdle at its lower end and the supporting flat surface.

Two cases were simulated. In the first case, both viscous friction forces and inertia forces occurring in the liquid layer displaced from the gap are taken into account. In the second case, only viscous friction forces are taken into account. In both cases, the maximum valve speed is approximately the same and is vmax = 0.57 m / s. During braking, the maximum calculated acceleration in the first case is 668 m / s^2, and in the second –3560 m / s^2. Accordingly, the maximum pressure (in the middle of the girdle) is in the first case 0.15 MPa, and in the second –0.93 MPa. The corresponding graphs are shown in Fig. 2 and 3.

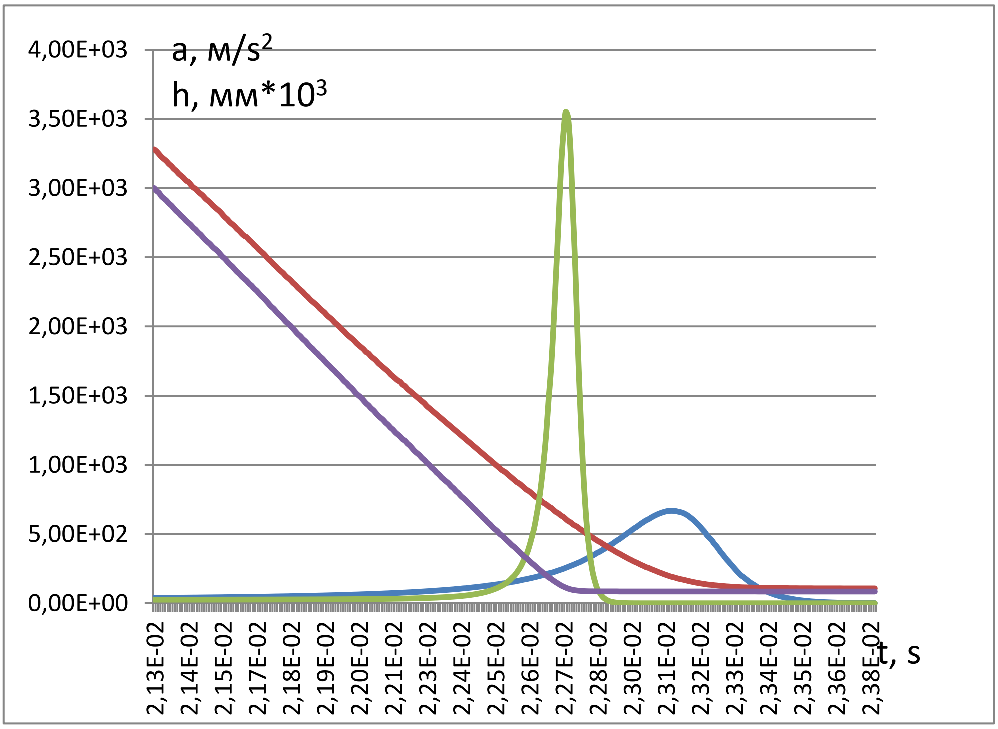


Fig. 2. The calculated dependences of the position (mm * 10^3) and valve acceleration (m / s^2) versus time (s) taking into account the inertia forces in the liquid (corresponding blue and red graphs) and without taking into account the inertia forces (green and violet graphs)

To conduct a physical experiment, the setup shown in Fig. 4 is used. It consists of a steel base with a flat upper surface, a steel guide cylinder with a cylindrical inner surface and a spring-loaded aluminum cup with a flat girdle on the lower surface, acting as a valve. The cup has the ability to freely move inside the guide cylinder but with a small gap. During the experiment, the entire structure is placed in a container of water. Grooves and holes are made in the cup, the guide cylinder and the base, which allow the water displaced from the closable gap to flow freely. To measure the cup acceleration at the fitting time, a shock accelerometer *PCB 350 C 04* is fixed in it.

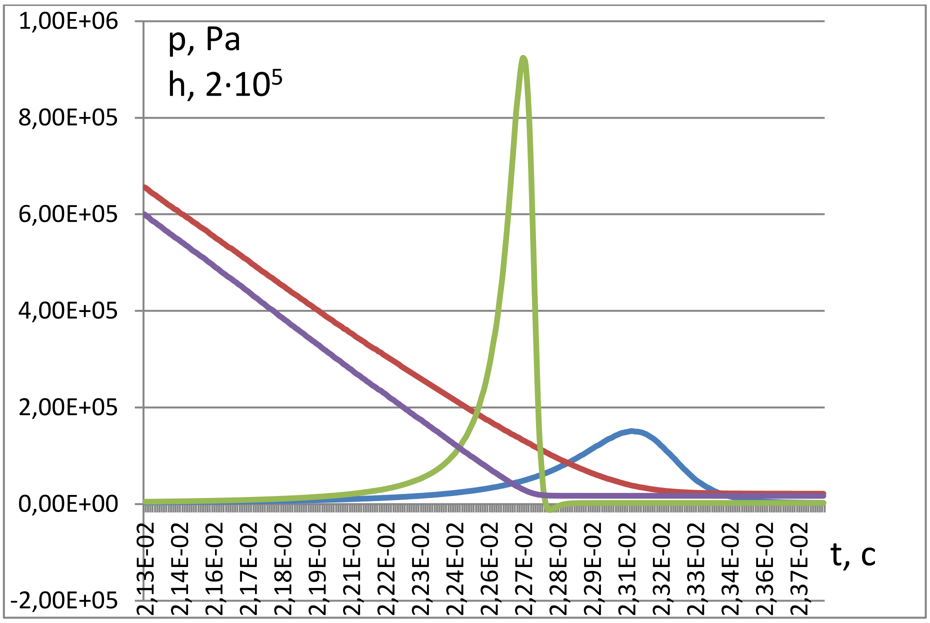
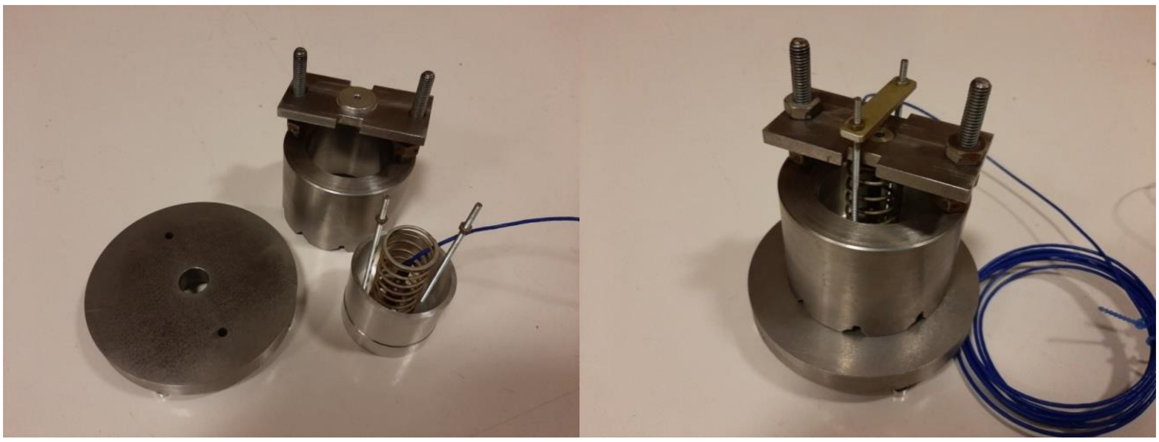


Fig. 3. The calculated dependences of the position (mm * $2 \cdot 10^5$) and the maximum (in the middle of the girdle) pressure of the displaced liquid (Pa) versus time (s) taking into account the inertial forces in the liquid (corresponding blue and red graphs) and without taking into account the inertia forces (green and purple graphics)

Setupparameter:	
– Cup weight	137 g;
– Spring weight	14 g;
– Spring stiffness	1230 N/м;
– Girdle outer diameter	56 мм;
– Girdle inner diameter	44 мм.
Parameters of shock accelerometer:	
– Sensitivity	0,1мV/(м/s²) (1,0 мV/*g*);
– Maximum impact acceleration	±49000 м/s² (±5000 *g*);
– Frequency range	(±1dB) from 0.4 to 10000 Hz
–Mass	5,4 g.



а) б)

Fig. 4. The experimental setup a) disassembled, b) assembled

The electric pulse generated by the shock accelerometer when the cup is located on the base in the presence of water in the gap and recorded by the oscilloscope is shown in Fig. 5.

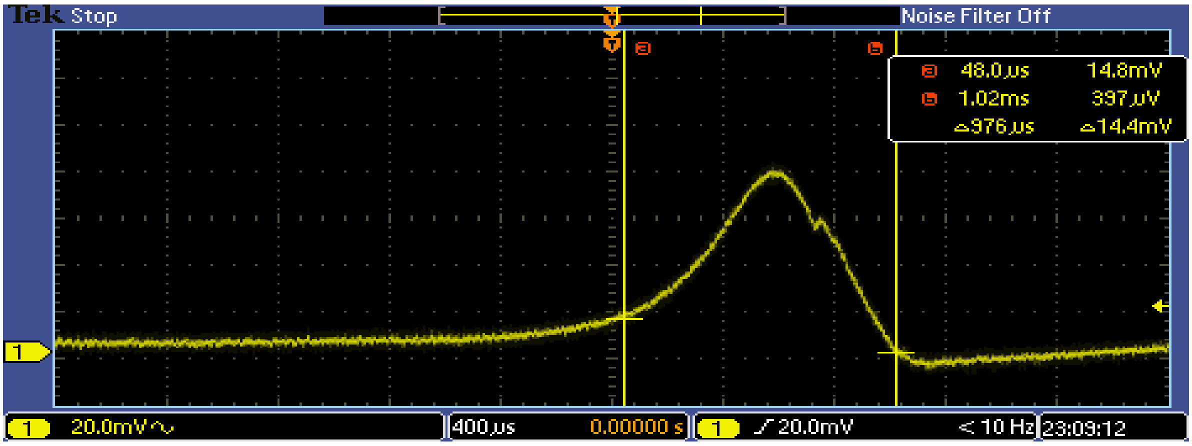


Fig. 5. Recording on an oscilloscope an acceleration pulse recorded by a shock oscilloscope during a physical experiment. Experiment graph of acceleration during the valve fitting layout taken in the experiment

Generalized data obtained as a result of processing data from computational and physical experiments:

For the physical experiment:

Maximum acceleration Impulse duration.

760м/s^2;**0,85**мs

For the computational experiment:

Maximum acceleration, duration, pressure:

– Option 1 **668**м/s^2; (-12%) **0,8**мs (-6,3%) 0,15 MPa;

– Option 2 3560 м/s^2; 0,93MPa.

Maximum speed: v_{max}=0,57м/c (approximately the same in both versions)

The data presented show a good approximation between theory and experiment, which allows us to use the above mathematical model of the valve fitting process in other studies. Figure 6 shows the calculated graph of the dependence of the largest (peak) pressure in the closing gap depending on the width of the girdle (other parameters remain the same). Reducing the width leads to a sharp, not proportional increase in peak pressure. So with a decrease in the width of the girdle to 0.5 mm, the calculated peak pressure reaches 370 MPa. Obviously, in this case, to increase the accuracy of the calculation, it is necessary to use a more complete mathematical model that takes into account the compressibility of the liquid. However, the goal of obtaining the above model was just to obtain a rational combination of valve assembly parameters at which pressures do not reach large values, and the working surfaces of the valve and saddle are durable.

In the same way, one can obtain the dependence of the peak pressure in the gap on the viscosity of the liquid (Fig. 7). The graph shows that this dependence is manifested much weaker.

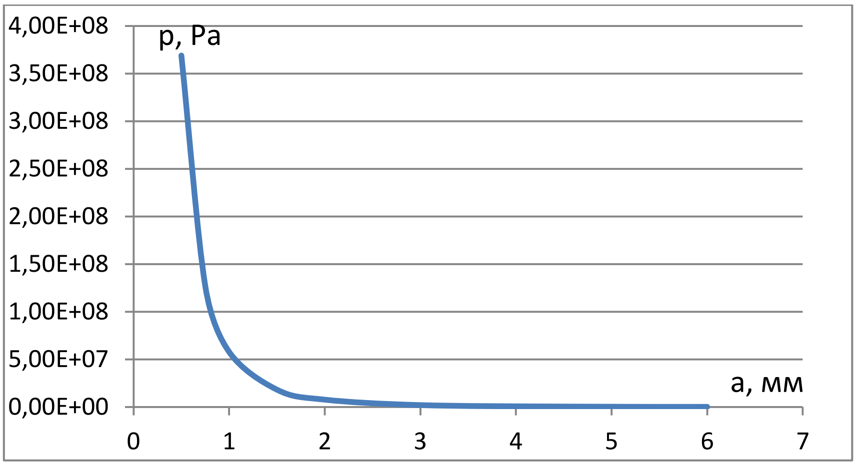


Fig. 6. Dependence of the largest (peak) pressure in the closing gap on the width of the girdle (other valve parameters are the same)

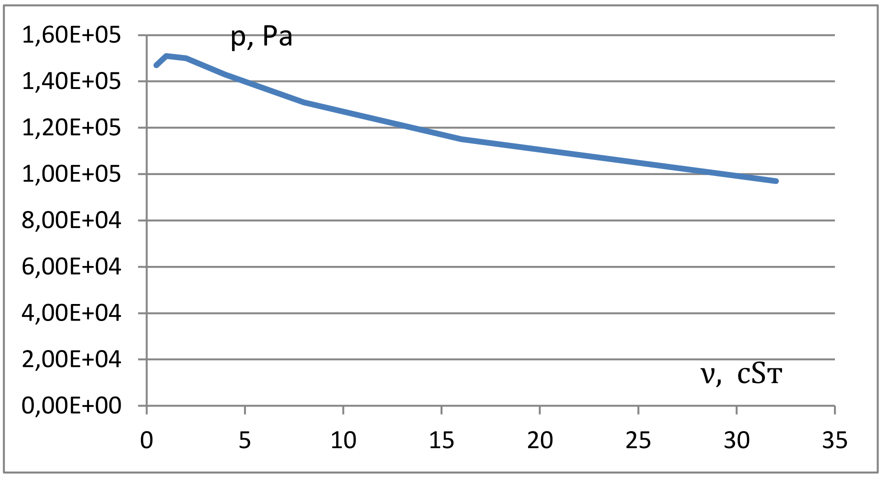


Fig. 7. The dependence of the liquid viscosity peak pressure in the gap

Conclusion

The results show that the inertia forces arising in a thin layer of fluid displaced from the closable gap between the closing valve and the saddle can be not only significant, but decisive in the process of valve seating on the seat, which requires consideration when designing volumetric pump valves.

References

[1] Semenov, S., Kulakov, D. Mathematical modeling of the mechanisms of volumetric hydraulic machines (2019) IOP Conference Series: Materials Science and Engineering, 492 (1), № 012042
[2] Rekach, F.V., Shambina, S.L., Sinichenko, E.K., Belousov, Y.V.Influence of the valve-pull-down equipment on the area of high pressure peaks in long pipelines(2019) International Journal of Recent Technology and Engineering, 8 (2 Special Issue 11), pp.
[3] Gouskov, A., Lomakin, V., Banin, E., Kuleshova, M., Chaburko, P.Investigation of the influence of centrifugal pump wet part geometry on hemolysis index(2019) IOP Conference Series: Materials Science and Engineering, 492 (1), № 012013, .
[4] Demikhov, K.E., Ochkov, A.A., Gordeeva, U.S.Experimental study of a vacuum fluid transportation system(2018) IOP Conference Series: Materials Science and Engineering, 387 (1), № 012015, .

[5] Naumov, V., Pogulyaev, Y., Baytimerov, R., Chizhov, D.New Diesel Engine Fuel Supply System Able to Control Pressure of Pre- and Post-Main Injections(2015) SAE Technical Papers, 2015-September, .

[6] Bastet.M., Rudnevs.S., Nekrasov.B. IDR. Hydraulics, hydraulic machines and hydraulic drives. Textbook for engineering universities-2nd ed., pererab. - M.: mechanical engineering, 1982 — - 423 p.