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Research paper

Interaction of the Mortar Flow with of a Mortar Pump's Valve Ball

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Abstract

The article deals with the process of interaction of the valve ball of a mortar pump with the mortar flow. The schemes of the working chamber with attached forces to its elements are drawn up. The hydrodynamic pressure is determined in the analytical form and as the results of the experiment. The structural parameters of the working chamber and the rheological parameters of the mortar are analyzed, which have the greatest influence on the magnitude of the hydrodynamic force.

Keywords: *hydrodynamic pressure, mortar, mortar pump, rheology, working chamber.*

1. Introduction

Mortar pumps of volumetric activity are the most widely used for pumping mortars through the pipeline. The principle of bulk pumps work is the periodic extrusion into the discharge pipe of the volume of liquid from the closed pump cavity [1]. The volume pumps are divided into the piston [2, 3] and rotary ones according to the type of the working body and to the movement they execute. In a piston pump, the fluid pumping occurs under the action of the piston, plunger and membrane [4] during the change in the volume of the working chamber, which is limited to valve or ball valve type. When the volume of the working chamber increases, the pressure in it decreases below the atmospheric level, and the liquid fills the working chamber through the opened suction valve. The flow energy is used to overcome the inertial and hydraulic resistance in the suction system. In the reverse movement, the piston creates excessive pressure [5] in the chamber, under the action of which the suction valve closes and the fluid is squeezed through the discharge valve into the pipeline. The lowering of the valves begins at the moment when the action of the flow from the valve seat bore to the ball decreases as a result of decreasing the velocity to the level of the own weight of this ball. The final closing of the valve occurs under the action of the reverse mortar flow. In this case, part of the useful volume of the pumped medium is lost in the form of leakage through the cluster gap [6]. Forced control of the valve operation is not widely used, because it leads to the complication of the design of the mortar pump.
In the development of new samples of mortar pumps, an actual and necessary analysis of methods for quantitative evaluation of the properties of mortars [7, 8, 9, 10, 11] is needed in order to reveal the mechanisms of manifestation of these properties when pumping using the mortar pump [12]. To create the optimal valve design in terms of minimum resistance and shortening the time of closing, it is necessary to evaluate the interaction of the valve closing element - the ball with the flow of the pumped medium [13, 14, 15]. Solving this problem using only purely analytical methods of hydraulics and mechanics of continuous medium is complicated by the action of many factors that influence the process of hydrodynamic interaction [16, 17] in a closed round valve space of the the mortar pump's working chamber. Consideration of the main factors such as rheological properties [18, 19, 20, 21, 22] and flow velocities, geometric parameters of the valve unit, is possible by developing analytical dependencies in a generalized form with empirical coefficients, which are determined experimentally.

2. Main body

The result of the interaction of the valve ball with the mortar is the force of the hydro-dynamical pressure on the ball from the side of the mortar flow. The origination of this force is due to several factors:
1. The presence of a cumulative mortar stream from the hole in the valve seat bore.
2. The action of normal and tangential stresses that arise in the boundary layer on the surface of the ball and are caused by the availability in the pumped medium of structured viscosity.
For the force of hydro-dynamic pressure F_{hd} we will write

$$F_{hd} = F_s + F_f \, , \tag{1}$$

where F_s – force of the ram pressure from the cumulative mortar stream from the valve seat bore;
F_f – the force of lateral friction with the flow of balls by a structured fluid in the limited space.
The forces F_s of frontal pressure on the ball from the side of the cumulative stream and lateral friction F_f when the ball is being flowed with the structured liquid in the limited space are of the

same type as when putting the ball in the fixed mortar, as mentioned above. Therefore, the interaction force of the valve ball with the mortar can be represented as the result of the elementary forces of pressure and friction on the ball surface.
These forces are proportional to the velocity of the flow, that is flowing around the ball. Taking into account the structural features of the valve unit – the presence of a seat and the limitation of the vent chamber, the force of pressure F_s on the ball from the side of the cumulative stream, is advisable to be considered proportional to the average velocity of the flow, which is going from the valve seat bore, and the force of the friction resistance F_f on the ball surface – to the average flow velocity in the maximum mid-section of the ball. The static component of the ball and mortar interaction is considered to be independent from the flow pattern. Then, on the basis of the above mentioned assumptions, the hydro-dynamical pressure dependence on the design parameters of the valve unit and the flow characteristics will be the following:

$$F_{hd} = C_p \cdot \pi \cdot \eta \cdot r_{v} \cdot \upsilon_s + C_\tau \cdot \pi \cdot \eta \cdot r_{v} \cdot \upsilon_{mid} + \pi^2 \cdot r_v^2 \cdot \tau_0 , \quad (2)$$

where r_v – the valve ball radius, m;
υ_s – the average velocity of the flow which is coming out from the valve seat bore, m/s;
υ_{mid} – the average flow velocity in the maximum mid-section, m/s;
η – the coefficient of the structured mortar viscosity, Pa·s;
τ_0 – the limit stress of the mortar displacement, Pa;
C_p, C_τ - coefficients of the ram pressure and friction resistance.
It should be noted that during continuous flowing around the balls in the limited volume, the introduced coefficients will be equal to: $C_p = 2$, $C_\tau = 4$. As the distance between the ball and the seat changes during the operation of the valve, the effect on the head pressure from the cumulative stream also changes. The closer the ball is to the seat and the smaller area of the valve slit is, the bigger is the pressure manifestation on the side of the stream. When the ball is estranged from the seat, the gap increases, and the effect of the cumulative stream pressure decreases. The nature of the change in the cumulative stream pressure on the ball is determined by the coefficient C_p and depends on the lift height of the ball, its diameter and the diameter of the valve seat bore. The parameter C_τ of the expression (2) characterizes the effect of the limitation of the flow, which washes the ball surface and is determined by the structural conditions of the valve chamber. The determination of the coefficients C_p and C_τ is possible on the basis of experimental study of the interaction process of a ball with the mortar in the confined space by means of physical modeling. During the valve operation there is redistribution of flow rates in the characteristic sections of the valve unit. This process is described by the Westfal's law, which determines that when a ball moves in the direction of flow, the flow-rate of mortar through the valve seat bore is higher than the flow-rate of mortar through the valve slit. The mathematical form of this law's record has such a form:

$$Q_{se} = Q_s \pm Q_v , \quad (3)$$

where Q_{se} – the flow-rate of mortar through the valve seat bore, m^3/s;
Q_s – the flow-rate of the mortar through the valve slit, m^3/s;
Q_v – the flow-rate of the mortar occupied by the under-valve space, m^3/s.
The "+" sign is used in the directional motion of the ball and the stream, and "-" when the ball is lowered against the flow direction.

Since, according to the flow continuity law, the flow rate in the maximum mid-section Q_{mid} is equal to the flow-rate of the mortar through the slit, then on the basis of expression (3) we will write

$$Q_{se} = Q_{mid} \pm Q_v . \quad (4)$$

The flow rates contained in (3), (4) are determined by the formulas:

$$Q_{se} = \upsilon_{se} \cdot S_{se}; \; Q_s = \upsilon_s \cdot S_s; \; Q_{mid} = \upsilon_{mid} \cdot S_{mid} , \quad (5)$$

where S_{se}, S_s, S_{mid} – the areas of the sections of the valve seat bore, the slit and the maximum mid-section, respectively, m^2;
υ_{se}, υ_s, υ_{mid} – average flow velocities in the valve seat bore, the slit and the maximum mid-section, respectively, m/s.
The flow-rate, which occupies the under-valve space at the lifting of the ball, can be approximated by the product

$$Q_v = \upsilon_v \cdot S_{se} , \quad (6)$$

where υ_v – the velocity of the valve ball, m/s.
Since the flow-rate through the valve seat bore is equal to the flow rate in the working chamber Q_p, then

$$Q_{se} = Q_p = \upsilon_p \cdot S_p , \quad (7)$$

where υ_p – the speed of the plunger, m/s;
S_p – the area of the bottom part of the plunger, m^2.
When the valve is opened, the ball moves in the flow direction, the relative velocity of the flow of the ball decreases by its velocity value, that is, the expression (2) will look like that:

$$F_{hd} = C_\mathrm{p} \cdot \pi \cdot \eta \cdot r_v \cdot (\upsilon_{se} - \upsilon_v) + C_\tau \cdot \pi \cdot \eta \cdot r_v \cdot (\upsilon_{mid} - \upsilon_v) + \pi^2 \cdot r_v^2 \cdot \tau_0 .$$

Taking into account that $\upsilon_{se} = \upsilon_p \cdot \frac{S_p}{S_{se}}$, and $\upsilon_{mid} = \frac{\upsilon_p \cdot S_p - \upsilon_v \cdot S_{se}}{S_{mid}}$ (see (4) – (7)), we will write down the following:

$$F_{hd} = C_\mathrm{p} \cdot \pi \cdot \eta \cdot r_v \cdot \left(\upsilon_p \cdot \frac{S_p}{S_{se}} - \upsilon_v \right) + C_\tau \cdot \pi \cdot \eta \cdot r_v \times \times \left(\frac{\upsilon_p \cdot S_p - \upsilon_v \cdot S_{se}}{S_{mid}} - \upsilon_v \right) + \pi^2 \cdot r_v^2 \cdot \tau_0 . \quad (8)$$

Having denoted the current height of lifting the ball over the seat with coordinate x, we will write the expression (8) in such a form:

$$F_{hd} = C_\mathrm{p} \cdot \pi \cdot \eta \cdot r_v \cdot \left(\upsilon_p \cdot \frac{S_p}{S_{se}} - \dot{x} \right) + C_\tau \cdot \pi \cdot \eta \cdot r_v \times \times \left(\frac{\upsilon_p \cdot S_p - \dot{x} \cdot S_{se}}{S_{mid}} - \dot{x} \right) + \pi^2 \cdot r_v^2 \cdot \tau_0 , \quad (9)$$

where $\dot{x}$ – the first derivative of the position of the ball x above the seat, which determines its velocity, ie υ_v.
Let's consider the work of the free-acting valve of the mortar pump under the action of external forces (Fig. 1-4) applied to the ball.
Valve operation is carried out in the vicinity of the "dead" points of the plunger: at the top point - the closure of the suction valve and the opening of the forcing one; in the bottom point - the closing of the forcing valve and opening of the suction one.

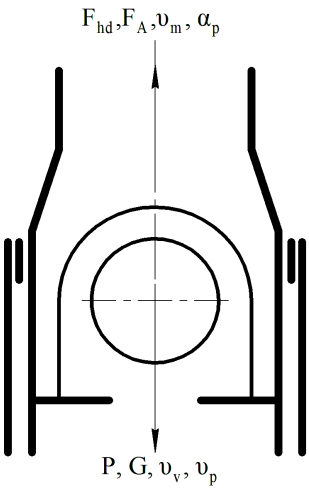

Fig. 1. The calculation scheme of the forcing valve's work: the beginning of the ball lowering at the moment of the plunger approaching to the bottom of the "dead" point

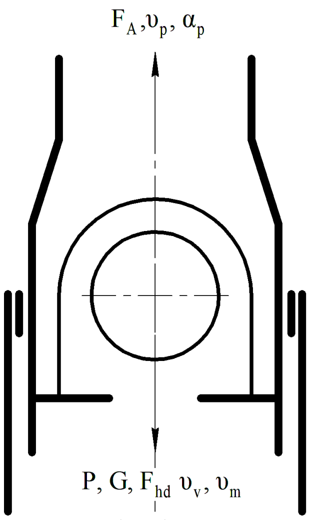

Fig. 2. The calculation scheme of the forcing valve's work: lowering when the plunger is removed from the lower "dead" point

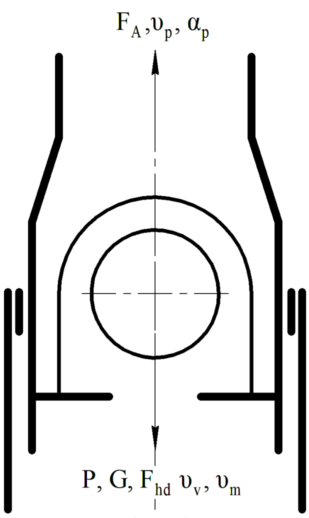

Fig. 3. The calculation scheme of the forcing valve's work: the start of lifting after the plunger has passed the upper "dead" point

Of particular interest is the forcing valve's operation, because when it closes, there are flow-rates at the nominal pressure of the feed, that is, the mortar, which is pumped out of the working chamber into the compensation chamber with costs of the nominal power of the pump drive. In addition, this valve unit moves together with the plunger, which additionaly influences its work.

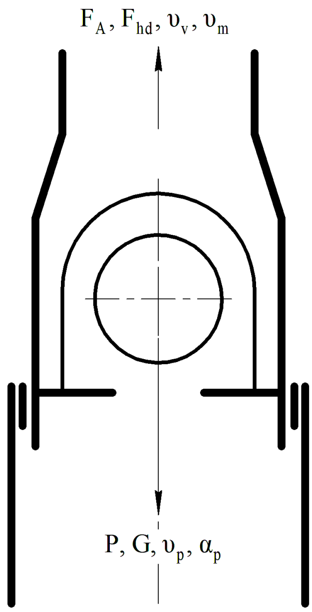

Fig. 4. Operational scheme of the forcing valve: lifting at the distance from the upper "dead" point

Therefore, it is relevant to analyze the movement of the ball of the mortar pump's forcing valve in the flow of the mortar mixture on the basis of determining the forces influencing the ball in the working process.

During the movement of the working body from the upper "dead" point, the ball is pressed against the limiting bracket (see Fig. 1) and moves along with the plunger and with the seat down with the speed υ_p . When a plunger is braked near the bottom of the "dead" point, the following situation occurs:

– firstly, the velocity of the flow and the hydrodynamic force are reduced with it;

– secondly, with a sudden stop of the plunger's motion, the removing of the ball from the bracket and its movement towards the seat due to its own inertia is possible.

In this case, relatively to the seat, the acceleration of the ball will be numerically equal to the acceleration of the plunger seat's braking α_p . That is, the effect of "sling" takes place – during the sudden and rapid plunger braking, the ball tends to break away from the bracket and to continue moving towards the seat under the inertial force effect

$$P = m_v \cdot \alpha_p , \qquad (10)$$

where m_v – the mass of the valve ball, kg;

α_p – the plunger acceleration, m/s^2.

After the plunger has passed the lower "dead" point (see Fig. 2), the seat begins moving towards the ball with the acceleration α_p .

Therefore, considering the ball motion relatively to the seat, it is reasonable to say that the ball moves towards the seat with additional acceleration α_p and the corresponding exciting force $P = m_v \cdot \alpha_p$.

The occurance of the inertial force P is stopped by the hydrodynamic force F_{hd} (Fig. 1, 3), caused by the difference in the hydrodynamic pressure in the cross flow before the ball is flown over and after it. While the valve is closed, the ball is located on the seat (Fig. 3), and the pressure drop is significant. In this case, the occurance of the force P is almost not noticeable.

A similar occurance of the inertial force is also possible when the valve is opened (see Fig. 3, 4). In this case, the ball tries to break away from the seat. During significant accelerations, the inertial

force of a steel ball with the diameter of 50 mm can have a strong effect on the functional operation of the valve.
In addition, the weight force G and the buoyancy force F_A are acting on the valve ball. Based on the joint analysis of the proposed calculation schemes, it can be assumed that the ball motion will take place under the influence of the resulting force, equal to the geometric amount of forces applied to the ball. Since the motion of the ball occurs along the action line of these forces, it can be written in the projection on the vertical axis

$$m_v \cdot \ddot{x} = F_{hd} + m - G + F_A . \tag{11}$$

The direction of the force P is determined by the opposite direction of the acceleration vector of the plunger α_p, and the direction F_{hd} by the vector of the flow velocity of the mortar υ_m in the space around the valve. In turn, the flow velocity υ_m is opposite to the plunger speed direction υ_p.
The developed dependences (8) and (9) indicate that the force F_{hd} is directly proportional to the flow velocity of the mortar, which is determined by the plunger velocity $\vec{\upsilon}_p$, and depends on the position of the ball, the size dimensions of the valve unit and the characteristics of the mortar. That is, in general, it can be represented as the functionality:

$$F_{hd} = f\left(xv, \dot{x}_v, \upsilon_p, \eta, \tau_0, r_v, r_{se}, D_{va}\right), \tag{12}$$

where x_v – the coordinate, which determines the instantaneous position of the ball over the seat;
$\dot{x}_v$ – instantaneous velocity of the ball (υ_v) in the position x_v;
υ_p – plunger speed;
η, τ_0 – rheological characteristics of the mortar, depending on its mobility;
r_v, r_{se} – the radii of the ball and the seat bore;
D_{va} - the diameter of space around the valve of the plunger.
Taking into account the dependences (10), (12), the expression (11) can be written as the differential equation

$$\begin{aligned} m_v \cdot \ddot{x}_v = F_{hd}\left(x_v, \dot{x}_v, \upsilon_p, \eta, \tau_0, r_v, r_{se}, D_{va}\right) + \\ + m_v \cdot \alpha_p - m_v \cdot g + \frac{4}{3} \cdot \pi \cdot r_v^3 \cdot g \cdot \rho \end{aligned} . \tag{13}$$

When considering the opening of the valve $(m_v \cdot \ddot{x}_v > 0)$, the initial condition is $x_v(t_0) = 0$ that corresponds to the lower position of the ball. When closing the valve $(m_v \cdot \ddot{x}_v < 0)$, the initial condition is $x(t_0) = h$ (the upper position of the ball).
The solution of this differential equation makes it possible to obtain the law of the valve ball motion as the following function

$$x_v(t) = f\left(m_v, r_v, r_{se}, D_{va}, \upsilon_p(t), \alpha_p(t), \eta, \tau_0, \rho, t\right), \tag{14}$$

which establishes the dependence between the nature of the ball motion with time t and the structural parameters of the valve $(m_v, r_v, r_{se}, D_{va})$, the plunger motion law $(\upsilon_p(t), \alpha_p(t))$ and the characteristics of the pumped mortar (η, τ_0, ρ). With this dependence, it is possible to determine the time of closing and opening of the valve. The solution of equation (13), taking into account (9), is possible only with the use of numerical methods for solving differential equations on the computer and requires preliminary experimental refinement of the parameters of the interaction of the ball with the mortar.
It is reasonable to apply this approach while analyzing the motion of the suction valve ball. It should be noted that the motion of the valve ball affects not only the level of the backflow, but also determines the resistance to the mortar flow, the level of dilution in the working chamber and the degree of its filling in the suction process.
Since during the work of the suction valve, its seat remains fixed, and the process of hydraulic interaction does not differ from the previous case, the balance of forces acting on the ball, has the following form:

$$m_v \cdot \ddot{x} = F_{hd} - G + F_A , \tag{15}$$

i.e. the inertial component m of the plunger motion is not taken into account.
Therefore, the ball motion is described by the following differential equation

$$\begin{aligned} m_v \cdot \ddot{x}_v = F_{hd}\left(x_v, \dot{x}_v, \upsilon_p, \eta, \tau_0, r_v, r_{se}, D_{va}\right) - \\ - m_v \cdot g + \frac{4}{3} \cdot \pi \cdot r_x^3 \cdot g \cdot \rho \end{aligned} , \tag{16}$$

the solution of which is done by the described above method.
Establishing the ball's motion law will allow to determine the time of its motion from height h to the seat and to estimate the amount of backflow ΔV_v when closing the valve.

$$\Delta V_v = F_p \cdot S_p(\varphi_v), \tag{17}$$

where $F_p = \frac{\pi}{2} \cdot D^2$ – the area of the bottom of the plunger, m^2;
$S_p(\varphi_v)$ – the plunger motion during the time of closing the valve, which is characterized by the plunger motion law, m;
φ_v – the angle of shaft rotation of the cam during the time t of closing the valve.
Dependence (17) is explained by the fact that during the valve closing the plunger moves to a certain distance $S_p(\varphi_v)$ and thus changes the volume of the working pump chamber. Since the simultaneous opening of both valves is impossible, the mortar through the valve, that is currently closed, flows in the form of the backflow and compensates the change in the working chamber volume. Hence, it is reasonable to assume that the effect of the operating body of the pump movement has the double character, that is, it determines not only the time of closing the valve, but also the size of the plunger flow during this time, which directly characterizes the volume of mortar losses. That is, the choice of the working body motion law does not need to be optimized according to the time of the valve closing and the motion intensity in this period.
During the interaction of the valve ball with the flow of the pumped environment there is the hydrodynamic force, which is the result of the projection of several components:
– the force of the ram pressure of the cumulative mortar stream, flowing from the opening of the valve seat bore $F_{\mathrm{p}} = C_p \cdot \pi \cdot \eta \cdot r_v \cdot \upsilon_{se}$;
– the force of viscous friction when the sidewall of the ball is flown round in the confined space $F_\tau = C_\tau \cdot \pi \cdot \eta \cdot r_v \cdot \upsilon_{mid}$;
– the force of the adjacent shifting motion $F_{\tau_0} = \pi^2 \cdot r_v^2 \cdot \tau_0$, which is the result of the manifestation of the structural properties of the environment.
In general, the interacting force of the ball with the flow is represented by the function (2).

$$F_{hd} = F_p + F_\tau + F_{\tau_0} = C_p \cdot \pi \cdot \eta \cdot r_v \cdot \upsilon_{se} + \\ + C_\tau \cdot \pi \cdot \eta \cdot r_v \cdot \upsilon_{mid} + \pi^2 \cdot r_v^2 \cdot \tau_0 .$$

Taking into account the buoyancy force $F_A = \frac{4}{3}\pi \cdot r_v^3 \cdot g \cdot \rho$, which also occurs during the interaction of the ball and the mortar, the result of the joint action will be the following:

$$F_{\Sigma hd} = \left[C_p \cdot \pi \cdot \eta \cdot r_v \cdot \upsilon_{se}\right] + \left[C_\tau \cdot \pi \cdot \eta \cdot r_{v} \cdot \upsilon_{mid}\right] + \\ + \left[\pi^2 \cdot r_v^2 \cdot \tau_0\right] + \left[\frac{4}{3} \cdot \pi \cdot r_v^3 \cdot g \cdot \rho\right] , \qquad (18)$$

where υ_{se} – the average flow velocity, which flows from the valve seat bore, m/s;
υ_{mid} – the average flow velocity in the mid-section of the ball, m/s,
$\upsilon_{se} = \frac{Q}{S_{se}} = \frac{4 \cdot Q}{\pi \cdot d_{se}^2}$, $\upsilon_{mid} = \frac{Q}{S_{mid}} = \frac{4 \cdot Q}{\pi \cdot \left(D^2 - d_v^2\right)}$.
The components F_{τ_0} and F_A do not depend on the flow velocity, but are determined by the mortar consistency and by the radius of the ball. The forces F_p and F_τ depend both on the consistency and on the flow velocity. Since the flow velocity changes according to the working body motion law, it is of particular interest to establish the functional dependencies of these force factors on the valve unit working process parameters. One of the ways to solve this problem is to determine the coefficients C_p and C_τ. To determine these parameters, it is necessary to estimate the magnitude of the interaction force of the ball with the mortar flow at fixed design parameters of the valve, flow velocity and flow consistency.
The study of the hydrodynamic interaction of the valve ball with the flow of the pumped mortar requires experimentation with the possibility of fixation the pressure force at different positions of the ball over the seat, different ball sizes, seat bore, flow velocity, its mobility, and so on.
In fig. 5, 6 the results of search experiments for mortar solutions of different mobility in the range of 8 - 10 cm are given with the same actual mortar flow, as well as with various values of the actual mortar flow with the mobility of 10 cm. The analysis of the results of these experiments allows for the following conclusions:
– the process of hydrodynamic interaction is reversible - when the external load is reduced, the ball takes its previous position;
– reducing the mobility of the mortar with the same other conditions (actual flow rate of the mortar, size dementions of the valve) leads to the growth of hydrodynamic force, which is associated with an increase in the structural viscosity of the flow;

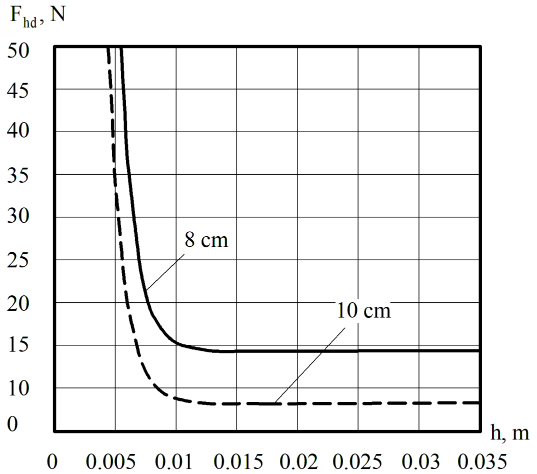

Fig. 5. Dependence of the force of the hydrodynamic flow pressure on a ball on the lifting height of the ball over the seat for mortars of various mobility at constant feed

– when increasing the actual mortar flow rate through the valve, the force acting on the ball increases almost directly proportionally to the mortar flow;
– as the distance of the ball from the seat decreases, the value of the hydrodynamic force increases, which is caused by the increase of the impact on the ball from the cumulative stream from the valve seat bore;

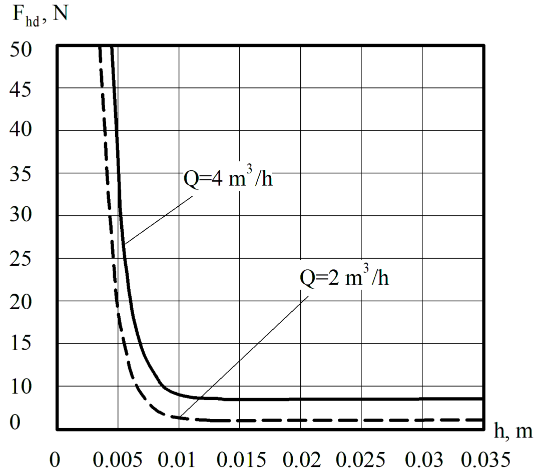

Fig. 6. Dependence of the force of the hydrodynamic flow pressure on the ball on the lifting height of the ball over the seat for the different value of the mortar flow with the mobility of 10 cm

– as the ball is pulled away for a certain distance, the further change in the height of the ball lift over the unuit does not lead to noticeable changes in the hydrodynamic force. That is, the interaction of the ball with the flow is determined by the effort that occurs when the sidewall of the ball is overfown, and the stream pressure force from the hole of the valve seat no longer affects the ball.
Consequently, the process of interaction of the ball with the mortar flow has distinct ranges of dominant influence of the components of the hydrodynamic force, depending on the height of the location of the ball over the seat h:
– viscous friction force with the flow of the lateral surface of the ball at $h > 20$ mm;
– the ram pressure force from the cumulative stream at $h < 10$ mm;
– within the limits $10 < h < 20$ force influence on a ball is caused by both components.

3. Conclusions

Therefore, the proposed approach to the operation of the ball valve of the differential solenoid pump analysis based on the considered dynamical model makes it possible to establish the ball motion law while operating the valve, taking into account the structural parameters of the valve unit, the motion law of the working element and the characteristics of the pumped environment, and theoretically define the effect of the actuator's motion law on the efficiency of the pressure and suction valves, taking into account the mortar loss at their closing.
The conducted search experiments indicate that the greatest influence on the hydrodynamic force value is caused by the distance between the ball and the seat.
In addition, according to the results of studies on the influence of design parameters of ball valves on the efficiency of mortar pumps, it is worth noting that the nature of the interaction is largely influenced by the consistency of the pumped mortar and the flow rate, as well as the geometric parameters of the ball d_v, the seat bore d_{se} and the space around the valve chamber D.

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