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CALCULATION OF THE TRANSITION PROCESSES IN THE PRESSURIZED WATER PIPES AT THE START OF THE PUMP UNIT

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ABSTRACT:

The article deals with processes of transition conduit when the centrifugal pump start determination value and hydraulic impact. Proper selection of the parameters of the transition process in the regulation of the operation of the units at the pumping station increases the reliability of the entire system, except for the pump, prevents vibrations that occur in it. Vibration damage and erosion are prevented.Using the results and graphs, it is possible to choose the right electric motor for the pump unit depending on the required pressure and water supply. Technical and economic efficiency is achieved by choosing the right engine.

The construction of the graphical characteristics of the pump, determined the magnitude of the hydraulic impact in the grapho-analytical method.

Keywords: pumping unit, centrifugal pump, valve, flap, steady motion, unsteady motion, transition process, hydraulic shock, pump characteristic, pipe characteristics, operating point, check valve.

INTRODUCTION:

By regulating the mode of operation of pumping units at pumping stations, it is possible to ensure the uninterrupted, normal operation of the system and prevent various negative consequences. Centrifugal pumps are widely used in engineering practice. The pump parameters fluctuate greatly during power outages over time, and such vibrations lead to high pressure pulsations in the flow and vibrations in the pump system. After a power outage, the transient process mainly experiences four modes, namely pump mode, brake mode, turbine mode, and escape mode[1]. Transition processes occur when the operating mode of the pump stations changes. This causes the operating parameters of the pump inside the running pump to change constantly. The interrelationships between the working parameters are determined by the corresponding equations, and different methods are used to solve these equations, depending on how accurate they are[2], [3].

FORMULATION:

When starting the pumping unit at the pumping station, the pipe is empty because the non-check valve is not installed in the pressure pipe. As soon as the pump starts, the water movement in the water supply pipe starts suddenly. The shock wave caused by the increase in pressure at the head of the water supply pipe propagates through the pipe. Therefore, the torque on the motor shaft during start-up is greater than the torque of the pumps. This difference in torque is due to the increase in the rotation frequency of the rotor of the pump unit. As the number of rotations of the impeller increases, the pressure created by the pump increases. Due to the partial opening of the post-pump valve, the water flow Q_n is transmitted through the pump. The difference between the pressure created by the pump and the pressure created in the pipe represents the loss of pressure in the valve, which is calculated as follows[4], [5]:

$$H_n - H_Q = S \cdot Q_n^2 \quad (1)$$

here H_n- the pressure created by the pump, m;
H_Q- the pressure levels generated in the pipe, m;
Q_n- water consumption by the pump, m^3/sec;
S- characteristic coefficient of the pipe, $1/m^3$.

The following formula is used depending on the characteristic coefficient of the pipe, the geometric dimensions of the pipe, the material of the pipe, the Reynolds number for fluid movement, and the local resistance coefficient:

$$S = \left(\lambda \frac{l}{d} + \sum \xi\right) \frac{8}{g\pi^2 d^4} = \frac{0.0827}{d^4}\left(\lambda \frac{l}{d} + \sum \xi\right) \quad (2)$$

here ξ- local hydraulic resistance coefficient.
λ- the coefficient of hydraulic friction resistance, which depends on the order of movement of the fluid and the unevenness of the pipe wall. The coefficient of hydraulic friction resistance is found in practice by the Alshul formula:

$$\lambda = 0.11\left(\frac{K_e}{d} + \frac{68}{Re}\right)^{0.25} \quad (2^*)$$

When pumping water with axial pumps, the outlet pipe is often not fitted with a shut-off valve, so the supply pipe is empty before the pump is started. The procedure for determining the starting parameters of the pump unit is as follows: the pump unit rotates the rotor, the torque of the motor shaft and the torque received by the pump.

When starting the centrifugal pump, the valve in the transmission line must be closed. The supply pipe must be filled with water for back-up pressure.

When water is pumped into the supply pipe, the non-return valve opens and water begins to flow through it, creating pressure in the supply pipe. As the liquid passes through the non-return valve, the water begins to shrink and the pipe expands, and as the pressure increases, the pressure wave propagates along the length of the pipe. In this case, a hydraulic shock begins to form in the pipe.

The occurrence of pressure pulsations at the pumping station leads to failure of hydromechanical devices and elements of pumping stations[6]. To prevent this, a series of steps must be taken when starting the pump unit:

- Rotation frequency synchronization;
- Increase the pressure in the pipe by synchronizing the rotation when filling the pipe;
- Start the shut-off devices after the pressure increases;
- To achieve a calculated order at the exit.

If a check valve is installed in the supply line, the pipe will be full of water when the pump is started and the valve will be closed. At t_j, when the pump is started, the water consumption is $Q_{nj} = 0$ [7].

If there is not enough water after the check valve opens, there will be unstable movement in the water supply.

The calculation of starting the pump with the gate valve open is performed in the following order: Water supply, which determines the length and number of billing sections. In this case, the length of the account section is taken as Δl. If the velocity a in the water supply does not change over the entire length, then the time to detect the pressure change is the same for all sections:

$$\Delta t = \frac{\Delta l}{a} \tag{3}$$

The water consumption per pump is determined as follows[5], [6]:

$$Q_n = v_o \cdot \omega \frac{n}{n_{pipe}} \tag{4}$$

here v_0 – the velocity of the water in the water pipe at the time of viewing, m/sec;
ω- cross-sectional area of the pipe, m^2;
n- the number of pumps operating in parallel,
n_{pipe}- the number of parallel water pipes.
The velocity of water in the unsteady motion of the water pipe can be determined from the following formula[8], [9]:

$$v = v_o + \frac{g}{a}(\varphi - \psi) \tag{5}$$

here v_o- the velocity of the water in the pipeline at the initial time ($v_o = 0$ m/sec),
g- gravitational acceleration constant, m/sec^2;
a- the velocity of a wave under pressure change, m/sec;
$\varphi - \psi$- the sum of the wavelengths of the pressure change, in the direction of water movement and in the opposite direction to water movement, respectively.
The value of the pressure height in unstable motion is determined as follows[8], [10]:

$$H = H_0 + \varphi + \psi \tag{6}$$

here H_0 – initial pressure, $H_0 = H_i$.

The suction line of the pump is short, so the hydraulic shock in the suction is often not considered. The calculation of hydraulic shock is based on the pipeline.
Zhukovsky's formula for calculating the hydraulic shock at start and stop of the pump is[11], [12]:

$$\Delta P_z = \rho v_o \frac{1}{\sqrt{\frac{\rho}{\kappa} + \frac{\rho d}{\delta E}}} \tag{7}$$

here v_o-the velocity of the water in the pipe, m/sec;
ρ-density of water, kg/m^3;
K- modulus of elasticity of water, Pa;
E- modulus of elasticity of the pipe, Pa;
δ- pipe wall thickness, m;
d- diameter of the pipe, m.

Depending on the shock pressure generated by the hydraulic shock, the reliability of the system is assessed and measures are taken to reduce the shock pressure. Hydraulic shock analysis is important in systems with centrifugal pumps. If automated centrifugal pumps are used in a small water supply system, the valve in the water supply line must be open when the pump is started. In this case, due to the small inertia in the moving fluid, the pump consumes less power, so it quickly has a sufficient rotational frequency, creating a high pressure in the pipe until the liquid moves.

In a short time, the pump starts pumping, as if the valve is closed. In this case, the liquid is compressed, the pipe being filled is filled, and the wall of the pipe expands under pressure. Under these conditions, the fluid flow during pump start-up is minimal and the generated working pressure is maximal. This can be obtained directly from the $Q - H$ characteristic of the centrifugal pump. The maximum pressure value at the start of the pump is determined from the operating point where the pump characteristics and the pipe characteristics intersect. In solving such a problem, the characteristic of a pipe in constant flow is also constructed[13], [14].

To do this, the functional graph of the pump characteristic $H = f(Q)$ and the equation of motion of the fluid in the pipe are solved together. The equation of motion of a fluid is:

$$Q = 3600 S_T \sqrt{\frac{2g}{\xi}\left(H - \frac{\beta_c c^2}{\xi g} - H_0\right) + \frac{2\beta_c c^2}{\xi^2_{exp}\left[\xi \frac{g}{\beta_c c^2}(H - H_0)\right]}} \tag{8}$$

here ξ- coefficient of hydraulic resistance,

$$\xi = \frac{2gh_y}{v_0^2} \tag{9}$$

h_y- pressure drop in steady motion, m;

$$h_y = H_i - H_0 \tag{10}$$

H_i and H_0- working and static pressure, m;
H- pressure at pump start, m;
β_c-coefficient of motion (in calculations it is possible to accept $\beta_c = 1$);

S_T- the cross-sectional area of the water supply pipe, m^2.

We explain the graph-analytical calculation procedure by the problem in the graph (Figure 1)[15]. Here, the first curve is the characteristic H = f (Q) of the pump, taken from the pump passport or the pump catalog. The second curve is a characteristic of the pipe for stable motion, which is determined by the following connection:

$$H_T = SQ^2 \tag{11}$$

here S- the characteristic coefficient of the pipe.

During normal operation of the pump, a working point A is determined where the pump characteristic and the pipe characteristic intersect. Formula (11) is used to determine the maximum pressure at the start of the pump, and based on this formula, the flow rate in the pipe is plotted. To do this, the fluid flow rate of the pump is determined by giving a series of values to H, respectively. Based on the values obtained, the fourth straight line on the graph is constructed and point C is found. Since we used the formula (11) in the construction of the graph, the loss of pressure in the water supply is found in exact values.

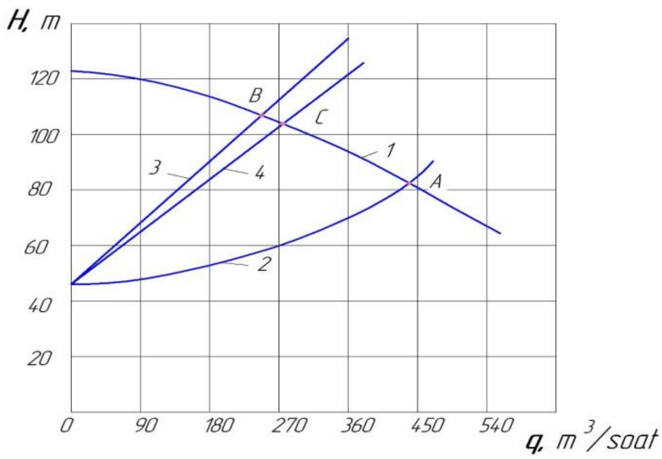

Figure 1. Combined characteristic of centrifugal pump and pipe.
1-characteristics of the pump, 2- the characteristic of a tube in constant motion of a fluid, 3,4- the characteristic of a pipe in unstable motion of a fluid.

RESULTS AND DISCUSSION:

We use Zhukovsky's formula to calculate the pressure due to the hydraulic shock generated during pump start-up:

$$H = H_0 + \frac{cv}{g} \tag{12}$$

This formula differs from formula (10) above. Since v is a variable in the formula, the third line in the figure is constructed and the working point B is found.

When the pump is stopped, if the check valve is fitted with a check valve, the hydraulic shock will not pose a significant risk to the water line. Basically, an increase in pressure occurs when the flow is disrupted. Before the calculation, it is necessary to determine whether the hydraulic shock is in a continuous flow or a continuous flow is broken.

If the continuity in the flow is not disturbed, the maximum pressure increase is determined by the following formula[4], [14]:

$$H = H_0 + \frac{cv_0}{g}\left(2e^{-\eta T} - 1\right) + \frac{h_P}{2}\left(1 + e^{-\eta T}\right) \tag{13}$$

here T-time, sec:

$$T = T_1 = \frac{2l}{c}$$

l_1- the distance from the pump station to the point of view, m;

e- the basis of the natural logarithm (e=27183);

η- the coefficient of attenuation of the wave is determined by the following formula:

$$\eta = \frac{h_P g}{2v_0 l}, \qquad sec^{-1}$$

h_P- loss of head, m;

l- the length of the pipe, m;

The condition $T > \frac{2l}{c}$ must be met to protect against hydraulic shock and to suppress the impact. As a result of the experiment, formula (13) was studied in detail. Therefore, the calculated pressure in high-power pumps differs slightly in practice, while analytical calculations for small pumps provide an accurate value[14].

If the pump is not fitted with a check valve, the calculation is complicated because if the check valve is not fitted, there is a reverse flow when the pump is stopped and the pump impeller rotates, i.e. the pump operates as a turbine.

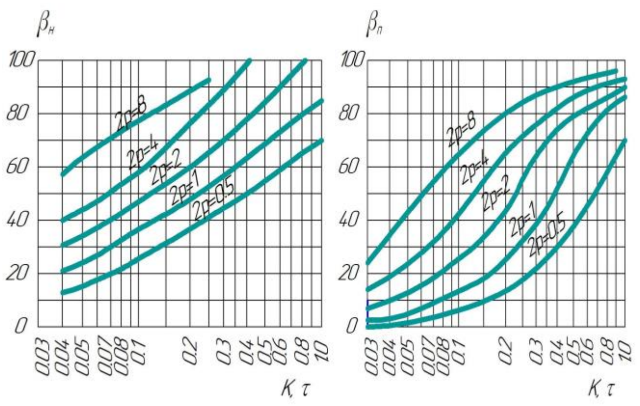

Figure 2. Graph for determining the maximum increase and decrease values of the pressure in the pumps.

If the moment of inertia in the rotor of the pump unit is too high, the non-check valve will not be installed. It is possible to determine the value of pressure increase and decrease in the pump. This is done using the approximation method for estimating hydraulic shock[16], [17]. Based on such a calculation, a graph is created(Figure 2)[15].

The ordinate of the graph is set to the coefficient of decrease β_H and the coefficient of increase β_P. These coefficients are obtained as a percentage of the normal operation of the pump.

The abscissa axis is denoted by $K_1\tau$, where:

$$K_1 = \frac{1800QH}{GD_a^2 n^2 \eta} \tag{14}$$

here Q- water consumption of the pump, $\frac{m^3}{sec}$;

H- nominal pressure of the pump, m;

n- the number of rotations of the pump impeller, $\frac{rot}{min}$;

η- the efficiency of the pump unit;

GD_a^2- the moment of inertia of the rotor of the pump unit, tm^2;

D_a- diameter of the rotor, m;

τ- hydraulic forging phase.

The value of D_a^2 is selected from the catalog of electric motors as follows[4]:

In axial pump

$$D_a^2 = (1.05 \ldots..1.07)D_g^2.$$

In low pressure centrifugal pumps $\mathrm{D_a^2} = (1.05 \ldots..1.07)\mathrm{D_g^2}$.

High pressure centrifugal pumps

$$\mathrm{D_a^2} = (1.05 \ldots..1.07)\mathrm{D_g^2}.$$

The value of the curve P in the graph is calculated as follows:

$$2\mathrm{P} = \frac{\mathrm{cv_0}}{\mathrm{gH_0}} \tag{15}$$

Based on the values of $\mathrm{K_1\tau}$ and 2P, the values β_H and β_P were determined. The decrease and increase of pressure are defined as follows:

$$\mathrm{h_{kam}} = 0.01\beta_\mathrm{H}\mathrm{H}$$

$$\mathrm{h_{osh}} = 0.01\beta_\mathrm{P}\mathrm{H}$$

Maximum pressure drop in the pump line:

$$\mathrm{H_{min}} = \mathrm{H_p} - \mathrm{h_{kam}} \tag{16}$$

Maximum pressure increase:

$$\mathrm{H_{max}} = \mathrm{H_o} + \mathrm{h_{osh}} \tag{17}$$

here $\mathrm{H_p}$- working pressure, m;

$\mathrm{H_o}$- static pressure, m.

Proper analysis of the hydraulic shock in the pumping system allows you to easily and efficiently organize the operation and increase the reliability of the system.

CONCLUSION:

Proper selection of the parameters of the transition process in the regulation of the operation of the units at the pumping station increases the reliability of the entire system, except for the pump, prevents vibrations that occur in it. Vibration damage and erosion are prevented.

Proper analysis of the hydraulic shock generated in the pumping system increases the reliability and efficiency of the system, and, accordingly, the operating parameters of the pump are selected. According to the data studied and obtained above, by reducing the hydraulic shock, it is necessary to increase the stopping time by preventing tension during starting and stopping the pump, which requires the use of additional devices.

Using the results and graphs, it is possible to choose the right electric motor for the pump unit depending on the required pressure and water supply. Technical and economic efficiency is achieved by choosing the right engine.

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