Simulation modelling of the pumping station with a high-speed motor

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Abstract. Using a quadripole network, a model has been built for a metalworking-machine pump unit consisting of a triple plunger pump. It is shown that the use of a high-speed electric motor may lead to the appearance of resonance with a pump-unit manifold, and as a result, the equipment for pulsations control fails to operate effectively. To solve the problem it is necessary either to update the pump manifold in order to change its oscillation eigenfrequency or to use a different motor speed.

1. Introduction
Hydraulic equipment of metalworking machines influence the accuracy, roughness and other parameters of fabricated products [1]. In Udmurtia, when a new product was introduced into the fabricating process at one of the plants, we were forced to replace the hydraulic equipment for the used metalworking machine.

A new pump unit was a high-speed motor activating a triple plunger pump. In the conventional approach, the processes taking place in the pump are considered quasi-stationary, and, therefore, hydraulic equipment is designed as if each plunger operates independently and the entire process is presented as a simple operation. In this connection, when the number of rotations increases causing the increase in the frequency of pulsations, the pulsation amplitude should decrease and, ideally, the value of the pressure at the pump outlet should be constant. However, the measurements have shown an unexpected result: the pressure oscillogram is presented in Fig.1. The outlet pressure does not equalize, on the contrary, its pulsation amplitude increases, and the existing pump-pulsation dampers do not operate effectively. To diagnose the current situation it has been offered to use mathematical simulation.

2. Mathematical model of pump unit
The structure of the mathematical model of the pump unit is displayed in Fig.2.

An electric motor sets in motion a crank mechanism connected to the pump plunger. The cycle of the second plunger has the shift by 120° relative to the first plunger and by 240° relative to the third one. Taking into consideration the constant angular speed of the motor ω, we can determine the rotational angle φ of the motor rotor as follows:
\[ \varphi = \omega t. \]

**Figure 1.** The oscillogram of the pressure variation at the pump outlet.

![Oscillogram of pressure](image)

\[ p(t) \]

\[ p(t) \]

**Figure 2.** The structure of the model of the pump unit: 1, 2, 3 are the pump cylinders, 4 is a pump manifold, \( v \) is the plunger speed, \( p \) is the pressure, \( G \) is the flowrate

If \( r \) is the crank radius (the maximal plunger stroke is \( 2r \)) and \( \ell \) is the length of the connecting rod, the coordinate of the plunger end \( x \) can be determined using the following dependence [2, p. 16]:

\[ x = r \left( 1 - \cos \varphi + \frac{r}{4\ell} \left( 1 - \cos 2\varphi \right) \right) \]

On differentiating the above expression, we can obtain the plunger velocity:

\[ v = r \omega \left( \sin \varphi + \frac{r}{4\ell} \sin 2\varphi \right) \]

Since the plunger is a moving wall intended for decreasing volume, at each moment of time it expels a volume equal to \( F x \), where \( F \) is the area of the plunger. This volume of a liquid can be presented as a certain fictitious flowrate:

\[ G_0 = \rho F v, \]

where \( \rho \) is the density of oil in the hydraulic unit.

Thus, it is possible to formulate equations of quadripoles for four elements of the pump unit shown in Fig.2. For each of the elements (cylinder 1 is used as an example) we have:

\[ \frac{dG}{dt} = \frac{p_1 - RG^2 - p_2}{L} \]

where \( R \) and \( L \) are the active and inductive resistance of the element.
\[
\frac{dp_1}{dt} = \frac{G_0 - G_1}{C}
\]
where \(C\) is the capacitive resistance of the element.

For closing the system of the equations, the rest of the pressures are determined according to Bernoulli law; for example, for \(p_2\):

\[
p_2 = p_7 + \frac{G_1^2}{2 \rho f^2}
\]
where \(f\) is the flow section with consideration of the coefficient of the clear opening of the valve at the outlet of the first cylinder. With regard for the design features of the valve seat, \(f\) is also the function of the pressure difference \(p_2 - p_7\).

The total flowrate at the manifold inlet is determined as follows:

\[
G = G_1 + G_2 + G_3.
\]

The determination of the inductive and capacitive resistance is performed using known formulae generalized, for example, in [3].

3. Results of simulation
The inductive and capacitive resistance turns out to be connected with the electric motor speed and determines the variation of the pressure and flowrate at the outlet of the pump unit. As a result, at a certain combination of them one can see the pattern presented in Fig.3.

Since the manifold has the oscillation eigenfrequency depending on \(C_4\) and \(L_4\), the flowrates at the outlet of the cylinders are not equal in the semi-cycle of their operation. Thus, due to the increased pressure, in the beginning the closing of the hydraulic valve takes place and the outlet flowrate is minimal. But inasmuch as there is a minimal flowrate at the manifold inlet, the pressure at the inlet decreases and at the operation of the second cylinder the closing of the valve is insignificant and the flowrate grows. As a result, the oscillation eigenfrequency of the manifold is imposed on the oscillations caused by the electric motor speed and the flowrate at the manifold inlet is such as shown in Fig.4. Correspondingly, instead of partially smoothing the inlet pulsations, the manifold increases some of them due to the presence of the oscillation eigenfrequency (Fig.5).

4. Conclusion
The results of the simulation show that the devices for smoothing the pulsations at the pump unit outlet do not operate effectively due to the presence of the oscillation eigenfrequency of the manifold. In this connection, it should be admitted that the existing designs of pump units cannot be used for the entire range of the electric motor speeds. Since the change of the design of manifolds is, as a rule, a difficult task, we think that it would be simpler to use a motor with a different speed for the installed pump unit, which resonance phenomena would not be so tangible.

Figure 3. The flowrates in the cylinders \(G\) (kg/s) depending on the time of operation.
Figure 4. The flowrate $G$ (kg/s) at the manifold inlet.

Figure 5. The flowrate $G_4$ (kg/s) at the manifold outlet.

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