Head pulsations in a centrifugal pump

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Abstract. This article investigated the factors, which affect to the character of the head pulsations of a centrifugal pump. We investigated the dependence of the shape and depth of these pulsations from the operation mode of the pump. Was determined, that the head pulsations at the outlet of the impeller (pulsations on the blade passing frequency) cause head pulsations at the outlet of the pump, that have the same frequency, but differ in shape and depth. These pulsations depend on the design features of the flow -through part of the pump (from the ratio of hydraulic losses on the friction and losses on the vortex formation). A feature of the researches that were conducted is also the using of not only hydraulic but also electric modeling methods. It allows determining the values of the components of hydraulic losses.

1. Introduction

The increase in the installed capacity of centrifugal pumps and the increase in the requirements for the efficiency of their operation lead to the reinforcing of the legislation of the European countries, which limits the vibroacoustic characteristics of hydraulic machines and equipment’s [1]. More and more attention is paid to researching problems, which related to improving the reliability and operation resource of power machines. In this regard, the analysis of dynamic processes and the development of ways to reduce the pump’s vibroactivity is becoming increasingly important, since this is related to the reliability and service life of electromechanical systems of water supply networks.

It is known that the vibrations of blade machines cover a wide frequency range - between 15 Hz to 20 kHz. The reason of this may be different dynamic processes: mechanical, hydrodynamic, electromagnetic and aerodynamic phenomena. Research results show that a significant part of the vibrations is due to hydrodynamic processes, in particular, head pulsations in the flowing part of the pump [2].

The main reason of such pulsations is hydrodynamic oscillations at a blade passing frequency. The pulsations on the blade passing frequency occur due as a result of the passage of the impeller blade through the cutwater, a section where the flow passes from rotational to translational motion.

The frequency and amplitude of the head pulsations are determined by the equations:

\[ f_z = \frac{zn}{60} \]  \hspace{1cm} (1)
where

- $f_z$ – pulsations on the blade passing frequency (Hz);
- $z$ – the number of blades of the impeller;
- $n$ – rotational speed (rev/min).

As you can see, the number of pulsations during the period of the main frequency (the one rotation of the pump impeller) according to the number of impeller blades. The amplitude of these pulsations depends on a number of design factors - the shape and number of impeller blades, the size, and configuration of the retraction of the radial clearance between the impeller and the cutwater, the speed of rotation of the rotor, and etc. An unfavorable relationship of these factors to each other can lead to an increase the amplitude of the head pressure in the volute of a centrifugal pump and in the adjacent discharge pipeline. Proceeding from the foregoing, the calculation of the amplitudes of head pulsations in centrifugal pumps at an early stage of design is an actual task.

Therefore, an important moment in the analysis of head pulsations in electromechanical systems with blade machines is the moment of choice the method of investigation. At the moment, the most common methods for analyzing head pulsations in blade machines is the method that applies the solutions of non-stationary equations of hydrodynamics [3] and the method for determining nonstationary pressure by integrating the Reynolds equations, while this nonstationary velocity field is determined through by method of laser anemometry [4].

A more innovative method is described in [1], which involves the using of acoustic-vortex numerical simulation of head pulsations. As the authors of the publication note, this approach makes it possible to construct an efficient and more accurate calculation method in compared with the usual methods of computational hydrodynamics.

In our article, we propose and apply a different approach that combines analysis of work processes in electromechanical systems (EMC) of water supply networks in the ANSYS CFX 14.0 program package with electrical modeling. During the research, it was established that at the stage of designing blade machines it is also necessary to take into account the possibility of imposing various kinds of pulsations and the probability of occurrence of resonant phenomena. This is can lead to an increase the amplitude of head pulsating at the pump outlet.

2. Investigation object and the geometry of the impeller

To investigate the factors that influence to the character of the head pulsation, we were selected the pump with a two-way suction and with a volute - AD2000-100-2 with the following parameters:

- flow - 2000 m$^3$/h;
- head – 100 m;
- rotational speed - 980 rev./min.

Simulations carried out concerning the pump unit with the following geometric parameters of the impeller, inlet diameter $D_0=334$ mm, outlet diameter $D_2=850$ mm, the number of blades $z=6$, the angle of the blades $\beta_2=31^\circ$ - at the outlet.

The design feature for the pump of type D is impeller of fluid double suction that compensating the axial forces, which act on the rotor. The flow of fluid that out from the impeller is directed to the pressure pipeline and passes through the volute and the diffuser of the pump. In a volute, near the cutwater, is formed unidirectional fluid flow, therefore decrease the velocity of flow and part of the kinetic energy of fluid flow is converted into pressure energy. Based on the kinematics of workflow and design features of the type D pump, flow of fluid in the outlet of the diffuser is pulsating and the number of pulsations has corresponded to the number of impeller blades.

3. Hydrodynamic modeling in ANSYS CFX

To determine the value of head and the amplitude of pulsation, we had created a model of the pump AD2000-100-2 and had conducted a numerical simulation of the flow of fluid in the flowing part of this pump. The numerical simulation had carried out for such pump operation modes: $0,6Q_{opt}$ and $Q_{opt}$. Calculation had carried the non-stationary, was set a step of time, which corresponding to the
rotation of the impeller by 5 degrees for the operation modes 0.6\(Q_{\text{opt}}\) and \(Q_{\text{opt}}\). All results, which given in this article were taken after the fifth rotation of the pump impeller.

To evaluate the change in the values of the amplitude of the head pulsations, the values of the head pulsations for various operating modes were derived. Figure 1 shows the graph of the change in head per rotation of the impeller for the \(Q_{\text{opt}}\) operation mode. Analyzing the results obtained, it should be noted that for an optimal operating mode, the minimum depth of pulsations is less than 5%.

![Figure 1](image1.png)

**Figure 1.** The graph of the head change at the outlet of the impeller and the volute of the pump for the nominal operating mode, \(Q_{\text{opt}}\).

At the same time, as the flow of fluid decreases is observed an increase in the amplitude of the head pulsations. For the 0.6\(Q_{\text{opt}}\) mode, the maximum value of the head pulsation is about 15%. Figure 2 shows the head pulsations at the outlet from the impeller and the volute of the pump for the operating mode 0.6\(Q_{\text{opt}}\).

![Figure 2](image2.png)

**Figure 2.** The graph of the head change at the outlet from the impeller and the volute of the pump for the operating mode 0.6\(Q_{\text{opt}}\).
Analyzing the results obtained, it should be noted that the value of the amplitude of head pulsations at the outlet from the volute can be greater, than the same value at the source of occurrence, the impeller of the pump. It is likely, that an increase in the amplitude of the head pulsations at the outlet from the volute of the pump is due to the appearance of resonant phenomena [5].

The nature of the change in the amplitude of the head pulsations at the outlet from the impeller and the volute, depending on the pump supply, is shown in Figure 3.

![Figure 3. Change in the amplitude of head pulsations at the outlet from the impeller and the volute, depending on pump supply.](image)

Despite the high accuracy and informativeness of the received results of the mathematical modeling of the working processes of a centrifugal pump, it has some disadvantages. They consist in the complexity of creating a computational model, in the use of many computing resources and a considerable time for obtaining results. That is why, for carrying out express researchers, we recommend the use of electrical simulation of the operating modes of the pump unit. The creation of an electric model is much simpler, and the results of modeling require significantly less time.

4. Electrical modeling
The chart of an electric model of the electromechanical system, that includes the volute of the pump and the impeller with six blades, shown in Figure 4.
Figure 4. The scheme of the electric model of the electromechanical system with pump unit, that has six blades AD 2000-100-2 (m = 6).

Consider the nature of the application of electrical methods of modeling operate modes in electromechanical systems of water supply systems by the example of the electric circuit in Figure 4 [6, 7]. We will carry out the researches the electromagnetic processes in the electric model EMS in the medium load mode, which is most common for this model, given the range of optimal flow of pump unit AD2000-100-2.

The electromotive forces $e_1 \ldots e_6$ model the physical process of energy transfer from the drive motor of the pump into the electromechanical system with simultaneous energy conversion by form and frequency. This EMF is sinusoidal, reflecting a rotating fluid flow in the centrifugal pump. The frequency of this EMF voltage as well as the speed of rotation of the pump impeller is typically different from the frequency of the power supply network. It is $\omega = \omega_m/p$. So instantaneous system sinusoidal EMF sources separate circuit branches EMS electric model (Figure 4), are as follows:

$$
\begin{align*}
    e_1 &= E_m \sin(\vartheta + \pi / 3); \\
    e_2 &= E_m \sin \vartheta; \\
    e_3 &= E_m \sin(\vartheta - \pi / 3); \\
    e_4 &= E_m \sin(\vartheta - 2\pi / 3); \\
    e_5 &= E_m \sin(\vartheta - \pi); \\
    e_6 &= E_m \sin(\vartheta - 4\pi / 3);
\end{align*}
$$

(2)

where $\vartheta = \omega t$ - variable, in which had performed all calculations.

The active hydraulic resistance $R_\text{Г}$, which is based on viscous forces between the layers of fluid and between the fluid and the walls of the channel, reflects the fact of dissipation energy as heat. The inductive hydraulic resistance of $X_\text{Г}$, which is created by forces of inertia that counteract the change of supply in a centrifugal pump in the between the area of blades channels (in the diagram Figure 4) is fed as hydraulic inductance $L_\text{Г}$.

The analysis of electromagnetic processes had conducted with using the conventional approach for devices of power electronics: the classic method of preparation and solution the system of differential equations with calculating the constants of integration. In the result had obtained:

a) system of differential equations for the circuit of the first stage of commutation (Figure 5(a)) taking into account the presence of active and reactive resistance:
e_1 - e_0 = x_\gamma \frac{di}{d\gamma} + R_T i_1 + u_{D1} - u_{D6} - R_T i_6 - x_\gamma \frac{di_6}{d\gamma}, \quad (3)
\[
e_1 = x_\gamma \frac{di_{1(2)}}{d\gamma} + R_T i_{1(2)} + u_{D1} + i_{1(2)} R_H + i_{1(2)} R_1, \quad (4)
\]

where \(x_\gamma = \omega L_\gamma\) - the inductive resistance of branches,
and the equation for the contour of the second stage of commutation (Figure 5 (b))
\[
e_1 = x_\gamma \frac{di_{1(2)}}{d\gamma} + R_T i_{1(2)} + u_{D1} + i_{1(2)} R_H + i_{1(2)} R_1. \quad (5)
\]

**Figure 5 (a).** The circuit of the first stage of commutation.

**Figure 5 (b).** The circuit of the second stage of commutation.

b) regularities of change of currents of diodes and rectified current

\[
i_{1(1)} = \frac{E_m}{2x_\gamma} (1 - \cos \gamma) \quad (6)
\]
\[
i_{1(2)} = i_{d(2)} = \frac{E_m}{Z} \sin(\gamma + \frac{\pi}{3} - \phi) + A \cos \gamma \sin \gamma \quad (7)
\]
\[
i_{1(1)} = \frac{\sqrt{3}E_m}{Z'} \sin(\gamma + \frac{\pi}{2} - \phi') + B \cos \gamma \sin \gamma \quad (8)
\]
\[
d\epsilon \quad A = \frac{E_m}{2x_\gamma} (1 - \cos \gamma) - \frac{E_m}{Z} \sin(\gamma + \frac{\pi}{3} - \phi), \quad (9)
\]
\[
B = \frac{E_m}{Z} \sin\left(\frac{2\pi}{3} - \phi\right) - \frac{\sqrt{3}E_m}{Z'} \sin\left(\frac{\pi}{2} - \phi'\right) + \left[ \frac{E_m}{2x_\gamma} (1 - \cos \gamma) - \frac{E_m}{Z} \sin(\gamma + \frac{\pi}{3} - \phi) \right] \frac{R_T}{\sqrt{R_T^2 + x_\gamma^2}} \quad (10)
\]

\(R_T = R_{T1} + R_{T2} + R_H\), \(Z = \sqrt{R_T^2 + x_\gamma^2}\) - full resistance the circuit of commutation;
\(\phi = \arctg \frac{x_\gamma}{R_T}\) - the angle phase shift on the first stage of commutation;
\(Z' = \sqrt{(2R_H + 2R_T + R_T)^2 + x_\gamma^2}\), \(\phi' = \arctg \frac{x_\gamma}{2R_H + 2R_T + R_T}\).

c) the equations to calculate the angle of commutation \(\gamma\):
\[
\cos \gamma + \frac{2(\sin(\gamma + \pi / 3 - \varphi_k) - \sin \gamma)}{\lambda} + \frac{4\cos(\gamma + \pi / 3 - \varphi_k)}{\lambda^2} = 1, \tag{11}
\]

where \( \lambda = R_x / x, \quad \varphi_k = \arctg(1 / \lambda) / \)

d) the expression of the instantaneous value of rectified voltage \([8],\) which has six identical
pulsations during the period, each of which is divided into two intervals with different regularities of changes:

\[
u_{d(1)} = \sqrt{3}E_m \sin(\theta + \pi / 2) / 2 - R_x E_m (1 - \cos \theta) / 2x; \tag{12}
\]

\[
u_{d(2)} = E_m \sin(\theta + \pi / 3 - x) E_m \cos(\theta + \pi / 3 - \varphi) / Z + R_x A e^{-R_x (\theta - \gamma)} - (R_x + R_x) E_m \sin(\theta + \pi / 3 - \varphi) / Z. \tag{13}
\]

The limited scope of the article does not allow to give the detailed description of the features
theoretical researchers that made in accordance with EMS, which contains the pump unit AD 2000-100-2 and the impeller of the pump, which has six blades.

Was conducted the simulation with using the electric model EMS of pump unit AD2000-100. Below (Figure 6) are shown the curves of the electrical model, where:

1 - the curve of rectified voltage of the electric model, which simulates the head of the impeller;
2 - the curve of rectified voltage at the output from the electric model, which simulates the head of the pump.

Figure 6. The rectified voltage of electric model of pump unit AD2000-100, operation mode 1,0 \(Q_{opt}.\)

5. Conclusions

1. For pump units with single spiral volute, the frequency of head pulsations at the outlet of the
impeller and the pump coincide with pulsations on the blade passing frequency.
2. The depth of the head pulsations at the outlet of the impeller is defined by losses on hydraulic
friction and on the formation of a vortex, which depends on the supply of pump.
3. The depth of the head pulsation of pump unit at its output depends on the losses of hydraulic
friction, on the formation of a vortex and the supply of pump unit. In the range of small feeds (to
0,7\(Q_{opt}\)) is observed the greater amplitude of head pulsations than at the outlet of the impeller.
4. According to the results of the mathematical modeling workflow of the pump unit, the electrical modeling method is acceptable for engineering calculations. For curves that show the head at the outlet of the impeller, the difference does not exceed 2%.

6. References
[1] Timushev S F and Klimenko D V 2015 Computation of pressure pulsations in the outlet devise of screw-centrifugal pump with acoustic-vortex method Vestnik SibGAU (Moscow, Moscow Aviation Institute) 16 (4) pp 907
[2] Bobkov A V 2012 Damping pressure output centrifugal pump Modern problems of science and education 5 pp 81-85
[3] Lugovaya S O and Moskalenko V V 2016 Influence of the blade output section geometry on the non-stationary interaction of the rotating impeller and the spiral volute of the centrifugal pump Pump and Equipment 4 pp 64-67
[4] Varaskin A 2003 Laser Doppler anemometry flow "gases-solid particles", problems of achievement, prospects Thermal physics of high temperatures 41 pp 746-754
[5] Chen Y N 1961 Pressure Oscillations in the Volute Casing of Storage Pumps Sulzer Technical Review pp 21-34
[6] Boiko V S, Boiko V V and Sotnyk M I 2011 Creating method of an electric model of a centrifugal pump: Patent UA № 67781, MPK G06G 7/00. Decl. 01.07.2011. Publ. 12.03.2012, Biul. No. 5
[7] Boiko V S, Sotnyk M I and Khovanskyy S A 2015 Electrical modeling of work processes in electrical systems of water supply networks Journal of Engineering Sciences 2 (2) pp 1-12
[8] Boiko V S and Sotnyk M I 2013 Adequacy of electrical modeling of work processes in a centrifugal pump Technical electrodynamics 5 pp 90-96