Simulation of non-stationary loads on the rotor of a pumping unit with an assembly pipeline

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Abstract. The article describes the method of calculating unsteady loads on the rotors of oil main pumps, based on the use of CFD-modeling methods. The limiting values of pressures in the pump, as well as the magnitude of the instantaneous and average values of axial and radial forces on the pump rotor depending on the intensity of transients in the pipeline system are determined. It is shown that peak loads are 20 times higher than the standard values of the forces acting on the pump rotor, which should be taken into account when designing systems and pumps for them.

1. Introduction

Modern methods of hydrodynamic modeling in centrifugal pumps, as a rule, are used to optimize the flow parts of pumps in order to increase their energy efficiency. However, one area has not been sufficiently studied by CFD modeling. These are non-stationary phenomena occurring in the pumps and the pipelines attached to them during transient processes in the hydraulic system. Especially such transients are important in systems with long pipelines, for example, in trunk pipelines [1-7].

As a result of a water hammer phenomenon occurrence with a sharp closure of valves, there appear, as the pumping experience shows, very significant axial and radial loads on the rotor of the hydraulic machine, briefly exceeding the standard by an order of magnitude or more. Such loads can lead to the destruction of the bearings and rotors of the pumps, and their magnitude should be taken into account when designing pumps and systems [8-16].

This article attempts to simulate the interaction of a pump and a system based on CFD-modeling methods for calculating a water hammer phenomenon in a pipe.

2. Methods

The STAR CCM+ software product was used in this work. It specializes in hydrodynamic modeling. In the case of calculating the flow of compressible fluid inside the pump, the following equations were applied.

The continuity equation (mass conservation equation):

\[ \frac{\partial p}{\partial t} + \frac{\partial (\rho u_j)}{\partial x_j} = 0, \quad (1) \]
where

\( u_j \) is the average value of the velocity of the fluid in the projection on the axis \( j \).

Navier—Stokes equations averaged over Reynolds (equations of conservation of momentum of motion averaged over time):

\[
\frac{\partial}{\partial t} (\rho u_j) + \frac{\partial}{\partial x_j} (\rho u_j u_j) = - \frac{\partial p}{\partial x_j} + \frac{\partial}{\partial x_i} \left[ T_{ij} - \rho (u_i u_j') \right],
\]

where

\( T_{ij} = 2 \mu s_{ij} - \frac{2}{3} \frac{\partial u_j}{\partial x_j} \) is the viscous stress tensor,

\( \mu \) is the dynamic fluid viscosity coefficient, \( \text{Pa} \cdot \text{s} \),

\( s_{ij} = \frac{1}{2} \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) \) is the strain rate tensor;

\( \rho (u_i u_j') \) are Reynolds stresses,

\( \rho \) is the averaged pressure,

\( u_j' \) is the pulsation velocity component.

To close this system of equations, the \( k-\omega \) SST (Shear Stress Transport) turbulence model was used, combining the advantages of the \( k-\varepsilon \) and \( k-\omega \) models. The model is complemented by two additional equations for the transfer of the kinetic energy of turbulence and the relative dissipation rate of this energy:

\[
\frac{\partial}{\partial t} (\rho k) + \frac{\partial}{\partial x_j} (\rho u_j k) = P_k - \rho \beta^2 \omega + \frac{\partial}{\partial x_j} \left[ (\mu + \sigma_\omega \mu) \frac{\partial \omega}{\partial x_j} \right];
\]

\[
\frac{\partial}{\partial t} (\rho \omega) + \frac{\partial}{\partial x_j} (\rho u_j \omega) = \frac{\omega}{k} P_k - \rho \beta^2 \omega^2 + \frac{\partial}{\partial x_j} \left[ (\mu + \sigma_\omega \mu) \frac{\partial \omega}{\partial x_j} \right] + 2 \cdot (1 - F_i) \sigma_{\omega^2} \cdot \frac{1}{\omega} \frac{\partial k}{\partial x_i} \frac{\partial \omega}{\partial x_i}.
\]

To account for the compressibility of a fluid, this system was supplemented with the equation of state (isothermal):

\[
p - p_0 = a^2 \cdot (p - p_0),
\]

where

\( p_0 \) is the initial pressure;

\( p_0 \) the initial density, \( a \) is the sound velocity in fluid.

As the boundary conditions when calculating the flow in the pump, the following combination was used.

1. Total inlet pressure, i.e.:

\[
p + \frac{\rho v^2}{2} = \text{const}.
\]

2. Outlet speed.

Modeling of the flow processes was carried out in a non-stationary formulation with a time step of \( 10^{-4} \) s. The criterion for the transition to the next step was the achievement of 10 iterations.

Two basic water hammer equations obtained by N.E. Zhukovsky. The expression for calculating the increase in pressure during water hammer:

\[
\Delta p = \rho \cdot a \cdot v,
\]
where
\( \rho \) is the fluid density;
\( a \) is the shock wave velocity;
\( v \) is the average velocity of the fluid in the pipeline before the water hammer occurs.

Water hammer phase is defined as

\[ \tau = \frac{2L}{a}, \quad (7) \]

where \( L \) is the pipeline length.

If the time \( t_{cls} \) of the pipeline overlap (valve closing) is less than the impact phase value \( \tau \), the water hammer will be due to the loss of all speed and the overpressure will be maximum. Such a water hammer is called a complete or direct.

Closing the valve was simulated by setting the function of the speed change from time:

\[
\begin{cases}
    v_{out}(t) = v_0, & (t < t_s) \\
    v_{out}(t) = v_0 \cdot \left(1 - \frac{t - t_s}{T_s}\right)^n, & (t_s \leq t \leq t_s + T_{cls}) \\
    v_{out}(t) = 0, & (t > t_s + T_{cls}),
\end{cases}
\]

where
\( t_s \) is the start time of valve closing;
\( T_{cls} \) is the period for which the valve should close;
\( n \) is the index number;
\( v_0 \) is the initial speed.

In this work, the linear law of variation was used: \( n = 1 \). The speed of sound was assumed to be constant and equal to 1000 m/s. The fluid is water.

The computational grid has a different topology. In the flow core, cells are polyhedral of various shapes and sizes. Cells near solid walls are multi-sided prisms, elongated in the direction perpendicular to the wall. The number of prismatic layers is assumed to be 5, with a stretch ratio of 1.3. The thickness relative to the base size of a polyhedral cell is 33%. The total number of cells was 840,000. The flow part of the pump HM 7000-210 (figure 1) was taken as the object of study.

**Figure 1.** Flow part of the centrifugal pump HM 7000-210.
To create a wave process that occurs when a water hammer, you can attach the pipelines to the discharge and suction nozzles to the flow part of the pump. At the same time, the length of the pipes does not necessarily have to be equal to the actual length, since this will significantly increase the computational grid, and, accordingly, large computational resources will be required. According to (6), the amplitude of the pressure jump upon a water hammer does not depend on the length of the pipes, but is determined by the flow rate of the medium. The length of the tube affects the period of propagation and reflection of the waves should be sufficient for the appearance of the wave process. In this case, the calculation in a non-stationary formulation must be carried out with a corresponding time step, namely, less than the magnitude of the impact phase (7).

We consider the following problem statement: one pumping unit of the type HM 7000-210 with a pipeline connected to the outlet nozzle is installed at the pump station. At the other end of the pipe a valve is installed, which regulates the mode of operation of the pump. Overpressure at the inlet is provided approximately 70 m (figure 2).

The diameter of the pipeline is 600 mm. The length was chosen to be 14 m. With this value, the size of the computational domain remains acceptable for the calculation on a laptop.

In accordance with the calculation scheme (figure 2), the total inlet pressure was assumed to be 700,000 Pa. To set the output velocity function, we define the phase of the water hammer for this system. The total length of the pipeline for the propagating wave in this problem:

\[ L = L_{\text{tube}} + L_{\text{pump}} = 14 + 5 = 19 \text{ m}, \]

where

\[ L_{\text{pump}} = 5 \text{ m} \]

is the path length for a wave inside the flow part of the pump (retraction, impeller, supply).

Then the phase of the water hammer will be:

\[ \tau = \frac{2L}{a} = \frac{2 \cdot 19}{1000} = 0.038 \text{s}. \]

Finally, we take the closing time to be

\[ T_{\text{cl}} = 0.01 \text{s}. \]

We understand that in a real situation there are no conditions for such a speed of closure, and water hammer occur in longer pipes and at longer closure times. However, in this case, the pipeline area will occupy most of the computational grid compared to the pump, which means that the computing power will be spent on modeling the processes in the pipe, and not in the pump, which is irrational, since it is the main object of study. First of all, the task of such a calculation is to understand what loads the pump unit will experience when a shock wave passes through it.
3. Results

As a result of simulating the impact of a water hammer on a centrifugal pump, the following data were obtained: pressure in the pipeline, radial force and torque on the rotor of the pump unit.

According to the obtained results, the oscillation period corresponds to the theoretical:

\[ T = \frac{4L}{a} = \frac{4 \times 19}{1000} = 0.0766 \text{s}. \]

The pressure around the valve at the moment of impact, taking into account the initial pressure in the pipe of about 2.7 MPa, should be equal to (6):

\[ p_{out} = p_{out}^0 + \Delta p = \rho \cdot a \cdot v = 2.7 + 1000 \cdot 1000 \cdot 6.88 = 9.58 \text{MPa}. \]

That is confirmed on the graph (figure 3).

![Figure 3. Pressure in the pipe with water hammer.](image)

![Figure 4. Radial force on the rotor of the pump HM 7000-210 with water hammer.](image)
Figure 5. The moment on the rotor.

Figure 4 shows a graph of the radial force acting on the pump rotor during the water hammer period. Compared with the nominal mode, the force quickly increased to 100,000 N in the first period of the impact, i.e. increased by about 20 times compared with the nominal mode. Then, according to the obtained graph, the radial force fluctuated around 20,000 N, reaching maximum values of 30,000…36,000 N.

This pump unit is equipped with sliding bearings, and such shock loads can lead to the contact of the shaft neck and the liner, which will adversely affect their service life. Also in this case, it is possible to touch the gap seals, but this is less likely, since the gap between the wheel and hub is larger than in the supports. If rolling bearings act as supports on the pump, then such loads can lead to their failure, since they are not designed to absorb shock overloads.

Figure 6. The propagation of the pressure wave during a water hammer.
In addition, the torque on the rotor (Figure 5) exceeded the nominal by 5000 N·m and then did not exceed the value in nominal mode of about 16000 N·m. But this blow may be enough for the coupling between the engine and the pump to fail, if it is not designed for such a load. In addition, it can cause loss of the synchronism of the engine of the pump unit and its emergency shutdown.

Despite the fact that the periods are small, during which the radial force and moment exceed the values corresponding to the pump zero flow, on real pipelines with a length of 100 km or more, the impact phases can reach 80…100 s, which is more than enough to break the listed components of the pump unit and oil pumping station stopping.

4. Discussion

1. The simulation shows that during rapid closure of the shut-off and control equipment in the main pumps the magnitude of forces on the rotor may exceed 20 or more times the standard values. This fact should be taken into account both in the construction of the system operation algorithms and in the pump design;

2. The proposed method for calculating such forces can be applied both for this class of hydraulic machines and for other systems with long pipelines.

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