

Mathematical modeling of a centrifugal pump with a spiral tap of simplified geometry with an open and closed wheel

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Annotation. Abstract: the characteristics of a pump with open and closed impellers are compared. The paper considers a model of a centrifugal pump for General industrial purposes. The simulation was carried out in the STAR-CCM + package. A graphical comparison of the influence of different pump design sizes on the parameters is made and the most suitable model is identified.

Introduction

In modern science, there are two large classes of hydraulic machines: volumetric and dynamic. Differ in the principle of action. Volumetric hydraulic machines have an overlapping flow part, that is, each volumetric machine has its own working volume. Dynamic hydraulic machines have a non-overlapping flow part. The most common are the latter. In this article we will talk about such a pump, belonging to the type of blade.

Vane pumps are the most common class of machines used in almost all sectors of the national economy. Questions related to the calculation of centrifugal pumps and their individual parts occupy a lot of space in the literature [1]–[5]. Vane pumps belonging to the class of dynamic pumps have become more widespread. The enormous pace of development of the national economy of the USSR, the creation of new industries have set specific tasks for pumping. It was necessary to perform a large amount of work on the development, research and development of serial production of an extensive range of new types and designs of pumps, the parameters of which differed significantly from the parameters previously produced [6].

Vane pumps are capable of pumping large volumes of liquid in a relatively short time. However, they do not create such high pressures as can create bulk. The efficiency of vane pumps varies in the region of 75-80%.

An important role in the operation of the pump is played by the size of the supply pipe. As practice shows, the show itself is better than a narrow inlet of the device, rather than wider. As we know from the theory of hydraulics, in narrower spaces the fluid velocity is greater and the pressure is less. This often leads to cavitation-the phenomenon of local transition of liquid into steam due to local pressure drop. Cavitation is a harmful phenomenon for pumps. Gas bubbles are surrounded by high pressure and collapse near the surface of the wheel or body, gradually destroying it. Therefore, in practice, often try to make the size of the inlet pipe large. In this article, the first part of the study is devoted to the construction of the pump performance characteristics depending on the size of the supply pipe.

In some cases, pumps with an open (or semi-open) impeller may be used. In particular, it can be waste-mass pumps (type cm), pumps for chemical reagents, pumps for pumping water containing



small impurities (1-3 mm), sludge water containing large particles (up to 20-30 mm), water containing long-fiber inclusions, petroleum products, liquids with a high content of abrasive, and so on.

Pumps with an open impeller have a lower efficiency, but are less prone to clogging, because the pumped medium has more space for passing the flow part. In this article, the second part of the study is devoted to the study of a pump with an open impeller with different gaps between the side of the blades and the body wall.

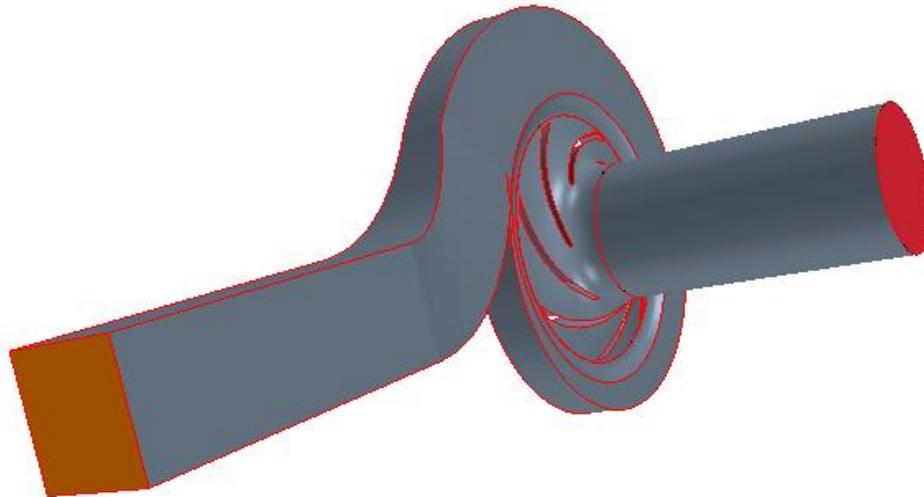


Figure 1. 3D model of the flow part of the pump with a closed impeller

In the first part of the study, a characteristic of the pump was constructed, the flow part of which was designed in accordance with the systematics [7]. Further, the diameter of the inlet pipe was changed to 1, 2 millimeters in each direction and characteristics were also built. The best one has been chosen.

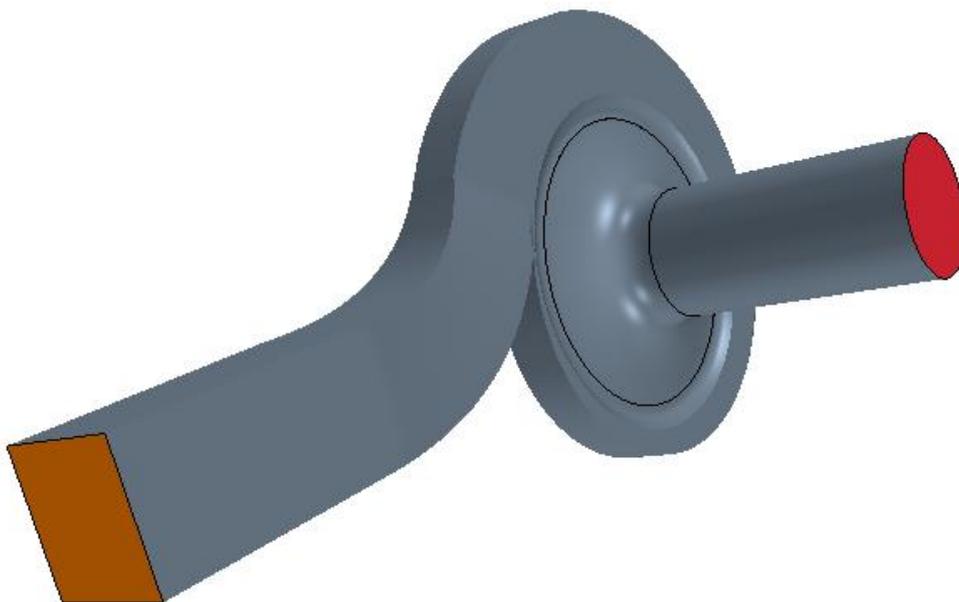


Figure 2. 3D model of the flow part of the pump with an open impeller

In the second part of the study, three characteristics were constructed corresponding to three different sizes of the gap between the side of the blades and the wall of the housing at the pump with an open impeller. The best characteristic is selected.

Methods

In General, in vector form, the Navier-Stokes equation for a liquid has the form:

$$\frac{\partial \vec{v}}{\partial t} + (\vec{v} \cdot \nabla) \vec{v} = -\frac{1}{\rho} \nabla p + \nu \Delta \vec{v} + \vec{f}$$

where t is time, ∇ is the Nabla operator, Δ is the Laplace vector operator, ρ is density, p is pressure, ν is kinematic viscosity, \vec{v} is the velocity vector field, \vec{f} is the mass force vector field.

The introduction of the Navier-Stokes equation averaged by Reynolds leaves the system of equations open, as additional unknown Reynolds stresses appear. To solve this system, we used the semi-empirical $k-\omega SST$ model of turbulence, which introduces the necessary additional equations: the equations of transfer of the kinetic energy of turbulence and the relative rate of dissipation of this energy:

$$\frac{\partial k}{\partial t} + U_j \frac{\partial k}{\partial x_j} = P_k - \beta k \omega + \frac{\partial}{\partial x_j} \left[(v + \sigma_k \nu_T) \cdot \frac{\partial k}{\partial x_j} \right]$$

$$\frac{\partial \omega}{\partial t} + U_j \frac{\partial \omega}{\partial x_j} = \alpha \cdot S^2 - \beta \cdot \omega^2 + \frac{\partial}{\partial x_j} \left[(v + \sigma_\omega \nu_T) \cdot \frac{\partial \omega}{\partial x_j} \right] + 2 \cdot (1 - F_1) \cdot \sigma_{\omega 2} \cdot \frac{1}{\omega} \cdot \frac{\partial k}{\partial x_j} \cdot \frac{\partial \omega}{\partial x_j}$$

U, P — averaged speed and pressure;

$S_{ij} = \frac{1}{2} \left[\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right]$ — instantaneous strain rate tensor;

The continuity equation also holds:

$$\frac{\partial \rho}{\partial t} + \text{div } \rho v = 0$$

The 3D models specified in the introduction were created using the CFTurbo application packages and in SolidWorks. This method is widely presented in the literature [8]–[14].

To simulate the flow, the model was loaded into the star-CCM + package. The following physical models were used: liquid, constant density, turbulent flow, $K-\omega$ turbulence model, non-stationary implicit calculation. At the entrance, the speed was set according to the required flow rate, at the exit-a pressure equal to zero.

To build the grid, we used: a surface grid generator, a polyhedral cell generator (base size 0.8 mm), a prismatic layer generator (5 layers, stretching 1.4, thickness-40% of the base size).

The calculation was carried out first in a stationary mode, in which the reference coordinate system, tied to the wheel, rotates at an angular speed equal to the frequency of rotation of the impeller. After the stationary calculation went to a constant mode (or steady-state fluctuations), the calculation continued in a non-stationary mode, in which the wheel itself rotates. By repeatedly modeling the flow, a family of points was obtained at different fluid flow rates. They formed the basis of the performance characteristics.

According to the same principle, the operating characteristics of the pump were built with other diameters of the inlet pipe. In the second part of the study, several performance characteristics were obtained in a similar way.

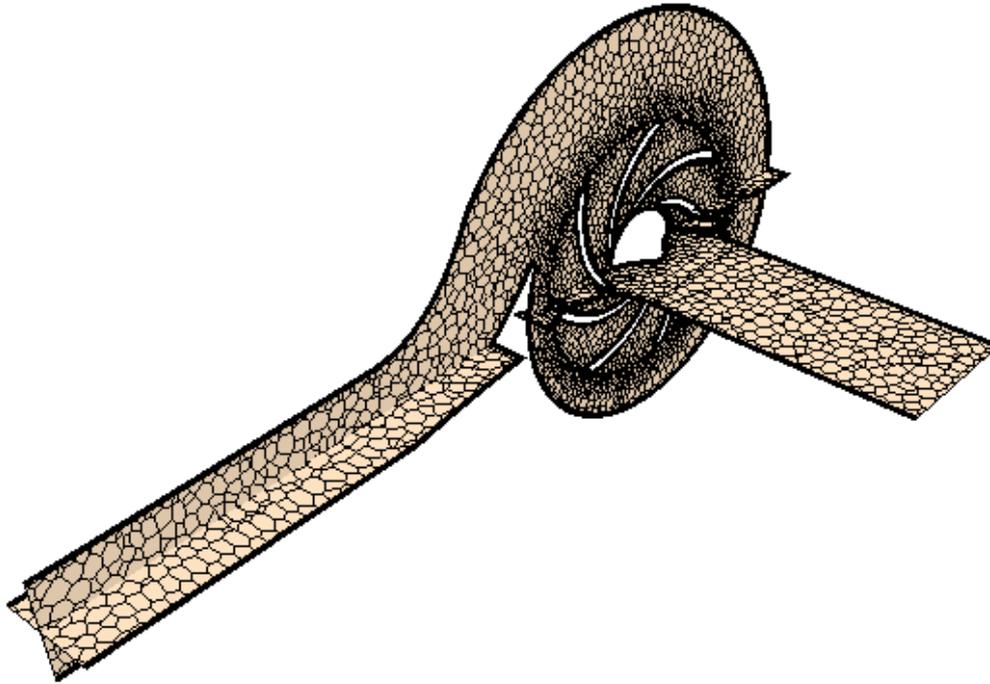


Figure 3. Calculated cross-section grid of the flow section

Results

In the process of modeling, we obtained pictures of the distribution of speeds and pressures in the flow part of the pump.

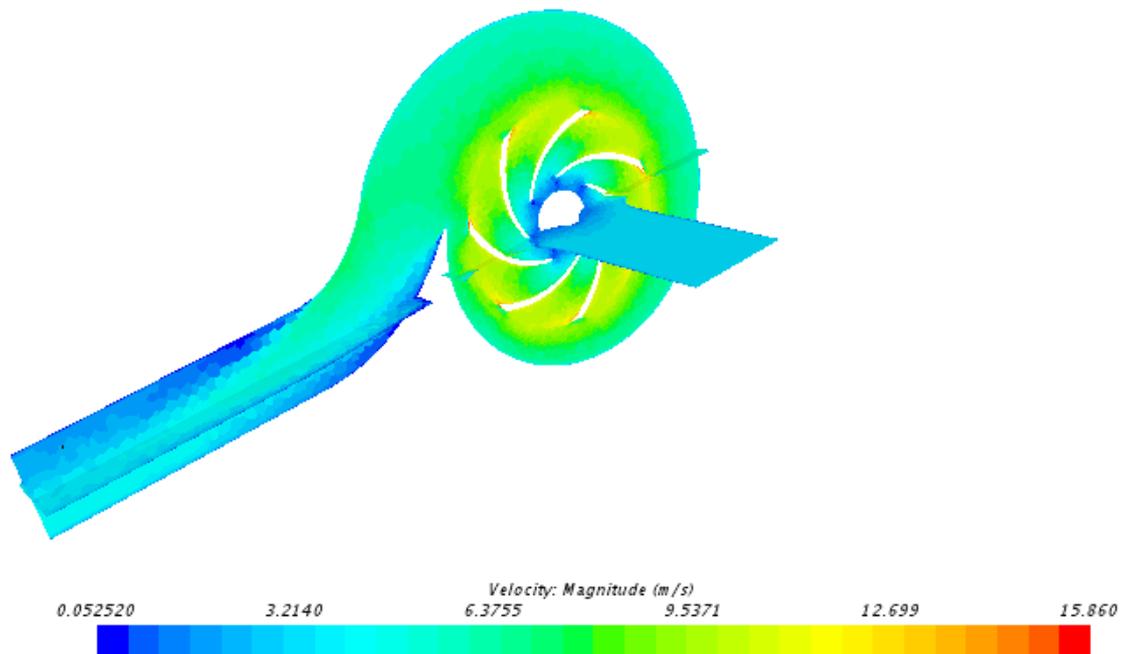


Figure 4. Speed distribution in the flow part of the pump with a closed wheel. Consumption 12 l / min

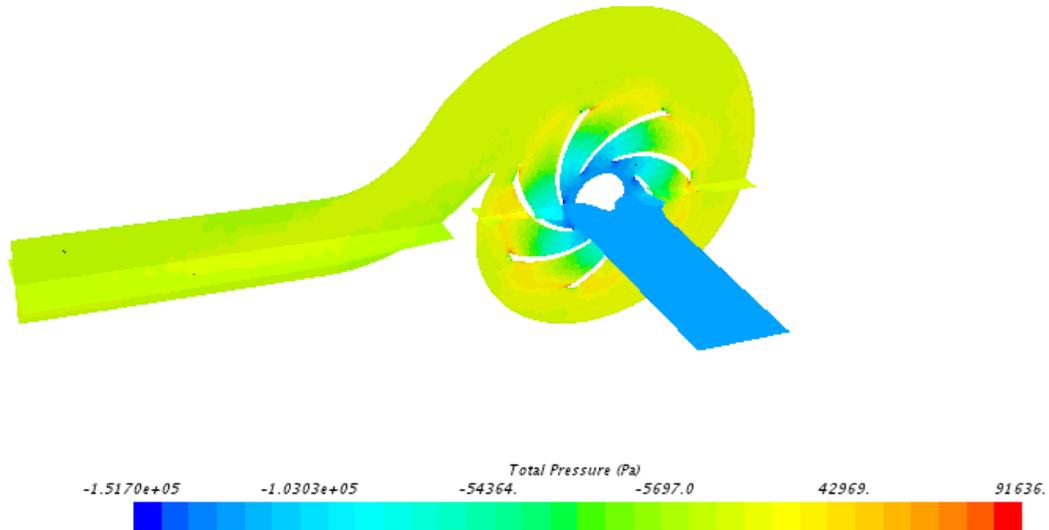


Figure 5. Pressure distribution in the flow part of the pump with the closed wheel. Consumption 12 l/min

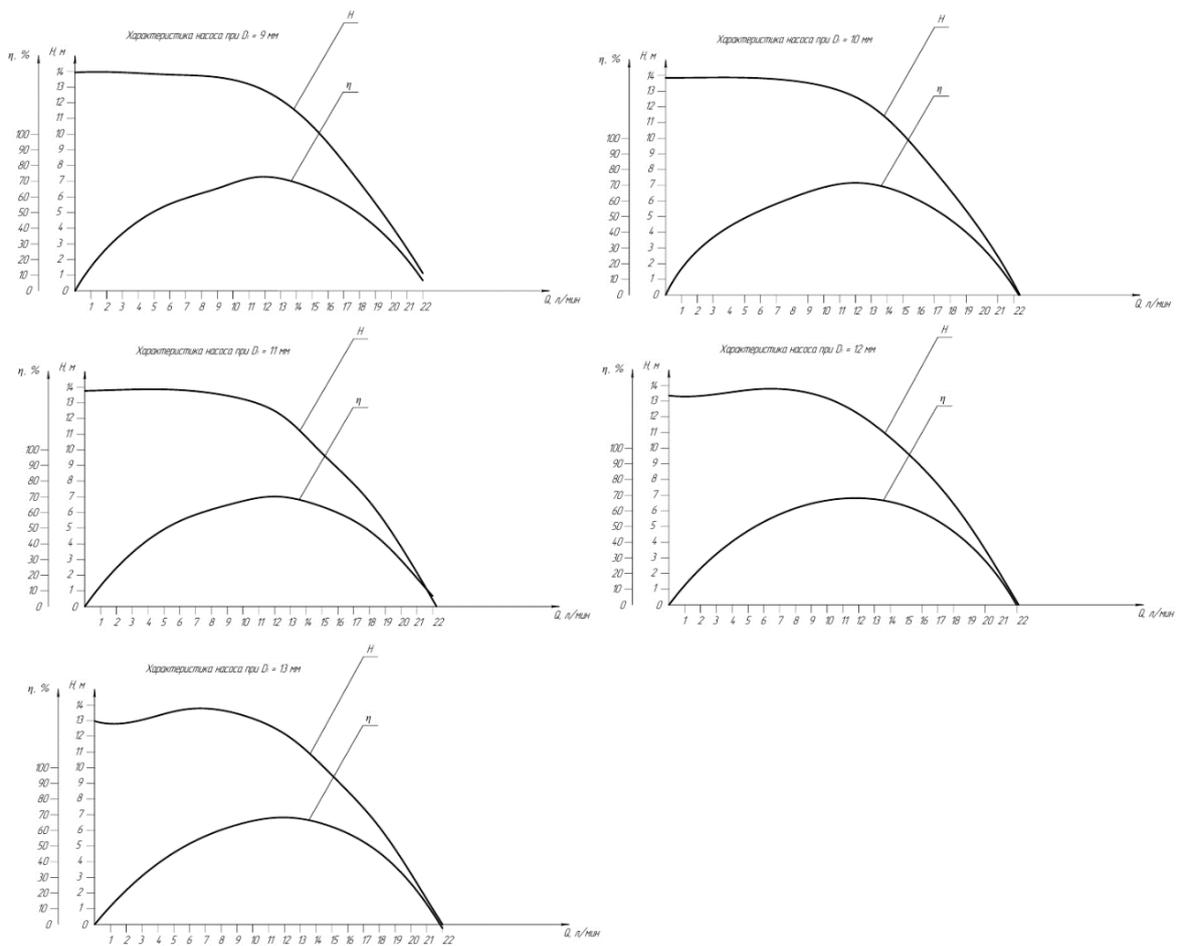


Figure 6. Pump Performance at different inlet pipe sizes

In the second part of the study, the approach was the same except that the diameter of the supply pipe did not change, but the gap between the sides of the blades and the body wall. For the calculation, a model with a diameter of the inlet pipe $D_1 = 9$ mm was taken.

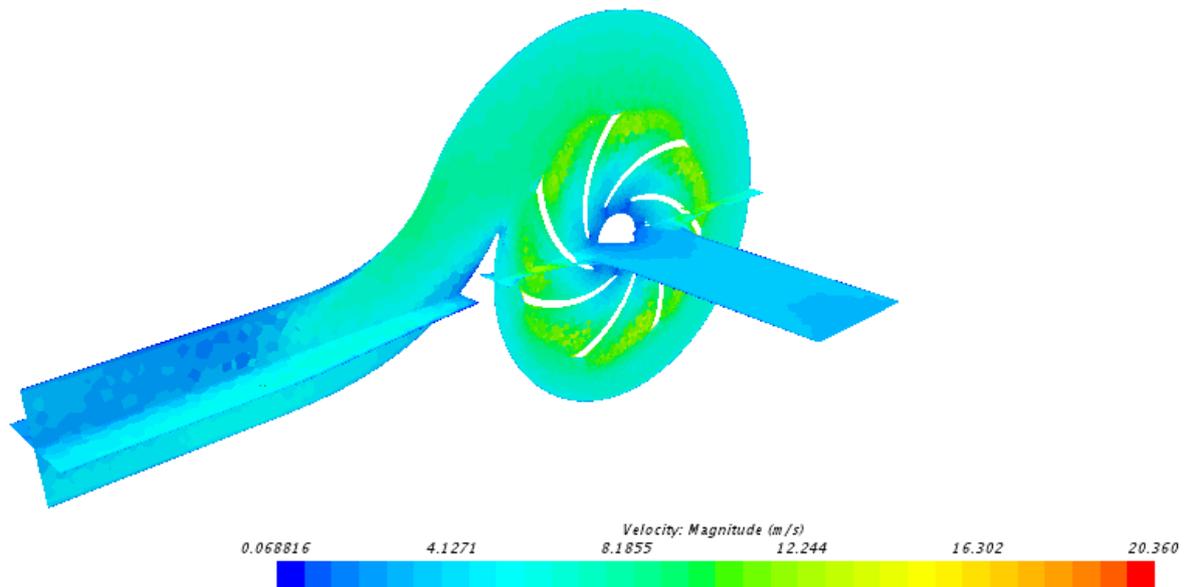


Figure 7. Speed distribution in the flow part of the pump with an open wheel. Consumption 12 l / min

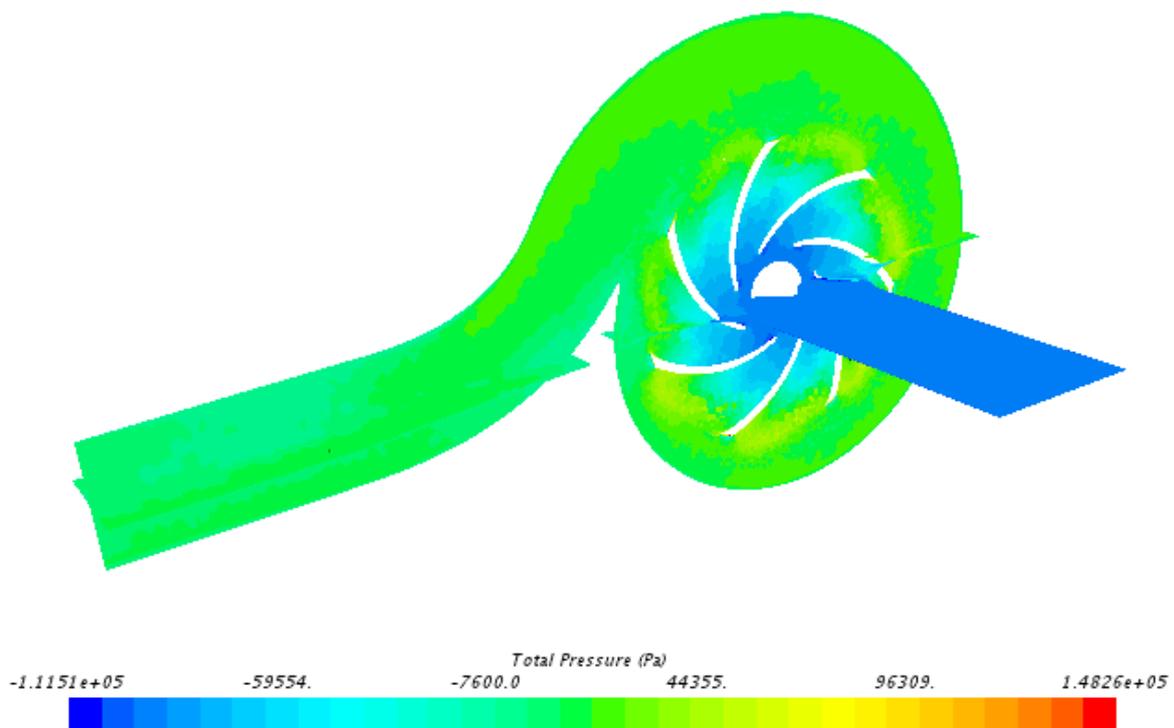


Figure 8. Pressure distribution in the flow part of the pump with an open wheel. Consumption 12 l / min

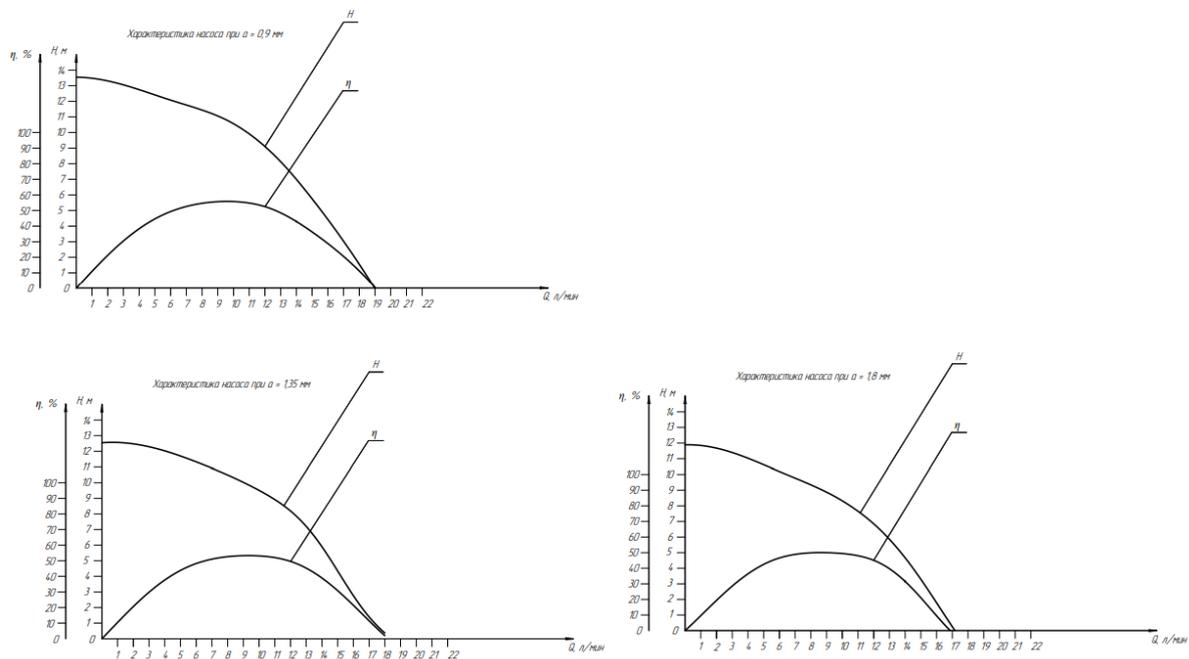


Figure 9. Operating characteristics of an open-wheel pump with different front clearance sizes

Summary

The results of the study confirmed the assumption that higher efficiencies are pumps with closed impeller, because there is no additional loss on the vortex formation because of the spread of liquid in the space between the side blades and the wall of the housing; then directed straight from the center to the periphery of the wheel well and wraps around the disks.

You can also see in the pressure distribution scenes that there is a negative pressure. This is due to the fact that the outlet pressure is assumed to be zero. This is physically incorrect, but the calculation will not change if you set any other pressure value. The pressure drop is important, not the absolute pressure value.

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