Optimization of the flow part of the pump for abrasive-containing liquids by hydrodynamic modeling methods

O Martynyuk¹ and A Petrov¹,²

¹Bauman Moscow State Technical University, 5 Second Baumanskaya Street, Moscow, 105005, Russian Federation
²E-mail: alexepetrov@bmstu.ru

Abstract. In this article the severe operating conditions of pumps for abrasive media, as well as the complexity of obtaining an analytical solution to the wear problem are described. Optimization criteria for such pumps are selected. A mathematical model for calculating a ground pump is presented. The STAR CCM+ software package modeled 15 models with different geometries and identified the main geometry parameters that affect the rate of erosion. A weight function was created that characterizes the success of model optimization based on the selected criteria. According to the accepted parameters, 20 models were modeled and the best geometric dimensions of the flow part were selected. The results are analyzed and conclusions are drawn.

Introduction

Pumps for abrasive media have a very wide range of applications. They are used for pumping waste and polluted water, in quarries, mines, mining and processing plants, and dredgers. The pumped liquid contains a lot of solid inclusions, which leads to erosion of the pumps. The service life of pumps is estimated in hundreds of hours and rarely exceeds 4000–5000 hours, so they try to achieve primarily a reduction in abrasive erosion, but also to obtain the maximum possible efficiency in their design. The length of the life cycle in such conditions is mainly affected by the rate of erosion [1]. Therefore, it was decided to optimize the pump for abrasive-containing hydraulic mixtures for these two parameters using hydrodynamic modeling [2]. The main difficulty of this method is that the existing wear models are purely empirical and require verification. Although a large amount of information has been collected on the operation of pumps for abrasive-containing liquids, the problem of flow wear and simulation of two- and three-phase flows has not yet been solved analytically and numerically [3].

Methods

Hydrodynamic modeling of the existing model ground centrifugal pump was performed in the STAR CCM+ software package [4,5]. This research method allows solving various problems of continuum mechanics [6–10].

Mathematical model:

- Continuity equation (mass conservation equation):
  \[
  \frac{\partial \rho}{\partial t} + \text{div}(\rho \cdot \vec{u}) = 0,
  \]

where \(\rho\) — density;
\( t \) — time;
\( \vec{u} \) — velocity vector

- Reynolds equations (equations of conservation of the amount of motion averaged over time):

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

where \( T_{ij} = 2\mu s_{ij} - \frac{2}{3} \frac{\partial u_i}{\partial x_j} \) — the tensor of viscous stresses;

\( \mu \) — dynamic coefficient of liquid viscosity, Pa\( \cdot \)s;

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

\( \rho u'_i u'_j \) — Reynolds stresses;

\( u' \) — pulsation component of the speed;

\( p \) — the averaged value of the pressure.

- The transport equation of turbulent kinetic energy:

\[
\frac{\partial}{\partial t}(\rho k) + \frac{\partial}{\partial x_j}(\rho u_j k) = P_k - \rho \beta' k \omega + \frac{\partial}{\partial x_j}\left[(\mu + \sigma_k \mu_t) \frac{\partial k}{\partial x_j}\right],
\]

where \( k \) — kinetic energy of turbulence;

\( P_k \) — energy generation of turbulence by the velocity shear;

\( \sigma_k, \beta' \) — the coefficients of the circuit;

\( \mu_t \) — coefficient of turbulent viscosity.

- Transfer equation for the relative rate of dissipation of this energy:

\[
\frac{\partial}{\partial t}(\rho \omega) + \frac{\partial}{\partial x_j}(\rho u_j \omega) = a_k^\rho p_k - \rho \beta' \omega^2 + \frac{\partial}{\partial x_j}\left[(\mu + \sigma_\omega \mu_t) \frac{\partial \omega}{\partial x_j}\right] + 2 \cdot (1 - F_i) \sigma_\omega \cdot \frac{1}{\omega} \frac{\partial k}{\partial x_i} \frac{\partial \omega}{\partial x_i}.
\]

Where:

\( \omega \) — velocity of kinetic energy dissipation of turbulence;

\( a, \beta, \sigma_\omega \) — the coefficients of the circuit;

\( F_i \) — weighting function.

The constructed grid of 362,330 cells is shown in figure 1.

**Fig. 1. Source pump grid**

Further, a simulation of the movement of the second phase of the flow (solid) was added to the calculation model, for which the trajectories of the particles were determined based on the Lagrange method.
The selected particle material is sand with a density of 2650 kg / m³. Particles with a diameter of 2.75 \times 10^{-4} \text{ m}.

The model of the particle drag force is determined by the drag coefficient, which is the Schiller-Nauman correlation.

Restitution coefficients (tangential and normal recovery coefficients) are used to predict the angle at which a particle bounces off a solid wall. For steel, they are defined using FORDER’s formulas:

\[
e_r = 1 - 0.78 \alpha_p + 0.84 \alpha_p^2 - 0.21 \alpha_p^3 + 0.028 \alpha_p^4 - 0.022 \alpha_p^5
\]

\[
e_N = 0.988 - 0.78 \alpha_p + 0.19 \alpha_p^2 - 0.24 \alpha_p^3 + 0.027 \alpha_p^4
\]

The disadvantage of this calculation is that the equations for these coefficients are derived only for carbon steel (based on experimental dependencies).

A method of modeling erosion of an Eye.

An injector generating particles with a resolution of 50x50 is installed at the entrance to the supply.

To assess the quality of the simulated flow parts of the pump, a weight function was created that characterizes how low the erosion rate is and how high the efficiency is in the designed flow part [11, 12].

\[
f = 1 - 0.7 \cdot \nu + 0.3 \cdot \eta,
\]

where \(f\) — weighting function;

\[
\nu = \frac{\nu_m}{\nu_i}
\]

\(\nu\) — dimensionless coefficient;

\(\nu_m\) — the rate of erosion of the model in question;

\(\nu_i\) — rate of erosion of the original model;

\(\eta\) — efficiency of the model under consideration.

The coefficients of the target function in this case are taken for reasons of prevailing low abrasive erosion over the requirement to obtain maximum efficiency, and can be changed to solve other similar problems.

Results

To determine the parameters that most affect the rate of erosion, models with variable geometry were previously modeled before optimization, and table 1 was constructed based on the simulation results [13].

| №   | Type of change                  | Erosion rate, g / hour | The efficiency | Force, H | The pressure of the fluid, m | Weighting function |
|-----|---------------------------------|------------------------|----------------|----------|-------------------------------|-------------------|
| 1   | Source model                    | 0.813                  | 0.77           | 3900     | 68                            | 0.531             |
| 2   | Increased inlet to the pump     | 0.816                  | 0.74           | 3880     | 65                            | 0.519             |
| 3   | Ring outlet                     | 0.767                  | 0.74           | 13125    | 67                            | 0.562             |
| 4   | Round section of the tap        | 0.939                  | 0.78           | 4380     | 68                            | 0.426             |

Окончание таблицы
It turned out, the erosion rates are mostly affected by the use of ring devices instead of helically-ring, the shape of the blade, the angle $\beta_1$, the login form on the blade, the fillet radius and the width of removal and angle of the blades. Graphs of the dependence of the efficiency and rate of erosion on some parameters are shown in figures 2–5.

| №   | Type of change                                      | Erosion rate, g / hour | The efficiency | Force, H | The pressure of the fluid, m | Weighting function |
|-----|-----------------------------------------------------|------------------------|----------------|----------|-------------------------------|-------------------|
| 5   | Increased tongue                                    | 0,897                  | 0,75           | 3000     | 68                            | 0,453             |
| 6   | The increased width of the outlet at the same bandwidth | 0,901                  | 0,77           | 2300     | 68                            | 0,455             |
| 7   | Increased bandwidth                                 | 0,8976                 | 0,74           | 2600     | 67                            | 0,449             |
| 8   | The increased angle of the blade                    | 0,903                  | 0,71           | 3080     | 61                            | 0,436             |
| 9   | The entrance to the blade is sharper                | 0,759                  | 0,73           | 4000     | 65                            | 0,565             |
| 10  | Larger angle $\beta_1$                              | 0,734                  | 0,73           | 3750     | 65                            | 0,587             |
| 11  | Larger angle $\beta_2$                              | 0,794                  | 0,53           | 2300     | 63                            | 0,475             |
| 12  | The thickness of the blades at the outlet           | 0,803                  | 0,72           | 3300     | 64                            | 0,525             |
| 13  | The thickness of the blades at the inlet            | 0,892                  | 0,73           | 3700     | 65                            | 0,451             |
| 14  | Another form of the blade                            | 0,697                  | 0,74           | 4200     | 65                            | 0,622             |
| 15  | The combined outlet                                  | 0,913                  | 0,73           | 8582     | 67                            | 0,433             |

**Fig.2** Dependence of the erosion rate on the radius of rounding of the branch section
Fig. 3 Dependence of efficiency on radius

Fig. 4 Dependence of the erosion rate

Fig. 5 Dependence of efficiency from the blade girth
However, as an object of optimization, the annular tap is not suitable because of the large radial force on the wheel rotor, which in this case acted as a restriction for the pump under study. Therefore, the wheel with the best shape of the blade and the shape of the blade at the inlet is used as a model for optimization [14]. In further calculations, the following parameters were used as variable parameters: the blade coverage, the angle $\beta_1$ at the inlet, the width of the branch and the radius of rounding of its section. The blade coverage varied from 88 to 94°. When changing it, it was necessary to take into account the value of the minimum conditional passage, which is 80 mm. The angle $\beta_1$ at the inlet varied from 36° to 40°; the width of the branch at the same throughput - from 342 to 390, the radius of the rounding section of the branch-from 75 mm to 20 mm [15]. The results are shown in table 2.

| № of model | Erosion rate, g / hour | The efficiency | Force, H | The pressure of the fluid, m | Weighting function |
|------------|------------------------|----------------|----------|-----------------------------|-------------------|
| 1          | 0.647                  | 0.75           | 3982     | 67.5                        | 0.668             |
| 2          | 0.72                   | 0.72           | 4843     | 63                          | 0.596             |
| 3          | 0.627                  | 0.70           | 2880     | 64                          | 0.670             |
| 4          | 0.577                  | 0.75           | 3971     | 68                          | 0.728             |
| 5          | 0.576                  | 0.72           | 2904     | 64                          | 0.720             |
| 6          | 0.697                  | 0.71           | 3916     | 63                          | 0.613             |
| 7          | 0.656                  | 0.75           | 4840     | 69                          | 0.660             |
| 8          | 0.631                  | 0.77           | 3804     | 68                          | 0.688             |
| 9          | 0.561                  | 0.72           | 3900     | 64                          | 0.733             |
| 10         | 0.578                  | 0.70           | 3788     | 62                          | 0.712             |
| 11         | 0.715                  | 0.72           | 1241     | 64                          | 0.600             |
| 12         | 0.555                  | 0.75           | 1700     | 65                          | 0.747             |
| 13         | 0.58                   | 0.74           | 3874     | 67                          | 0.723             |
| 14         | 0.605                  | 0.74           | 4750     | 65                          | 0.701             |
| 15         | 0.573                  | 0.73           | 4391     | 65                          | 0.726             |
| 16         | 0.636                  | 0.74           | 3726     | 66                          | 0.674             |
| 17         | 0.647                  | 0.77           | 4164     | 68                          | 0.674             |
| 18         | 0.572                  | 0.73           | 3604     | 65                          | 0.727             |

The highest weight coefficient was found in the model number 12. It is shown in figure 6, and the results in 7.8 and 9.

Its parameters are:
Radius of rounding of the branch section 20 mm;
Tap width 380 mm;
Blade coverage angle 92.5;
The angle $\beta_1$ of 38°.
Model characteristics:
Efficiency 75%;
Erosion rate 0.555 g / hour;
Head 65 m;
Radial force 1700 H.
Discussions
Preliminary results of the study showed that for the lowest rate of erosion, it is necessary to design wider blades in the meridional section. Reduce the surface areas perpendicular to the flow, namely: use a slightly elongated inlet to the blade and a square section of the outlet with the minimum allowable radii of rounding the corners. In this case, it was possible to achieve a significant reduction in the calculated erosion rate at an acceptable efficiency value. However, the results obtained require experimental verification.

References
[1] Vane pumps for abrasive hydraulic mixtures Zhivotovsky L. S. Smolovskaya L. A. m: Mashinostroenie, 1978. — 223 p., II.
[2] Lomakin V. O. Petrov A. I. Verification of calculation results in the package of hydrodynamic modeling of the ster CCM + flow part of the centrifugal pump Ah 50-32-200 / /
Izvestiyahigher educational institutions. Mechanical engineering 2012. — Spets. the release of the Spec. issue" Works of students and young scientists of Bauman Moscow state technical University". — P. 6–9

[3] Lomakin V. O. Petrov A. I. Kuleshova N. S. Investigation of two-phase flow in an axecenter wheel by hydrodynamic modeling methods // Science and education: scientific publication of Bauman Moscow state technical University 2014. — № 9

[4] Lomakin V. O. Chaburko P. S. Numerical simulation of liquid flow in a jet pump // mechanical engineering: network electronic scientific journal 2014. — № 3. — P. 55–58

[5] n. Isaev et al. 2019 VGD Conf. Serial.: Mother. Sci. Eng. 589 012009

[6] eMorozova et al. 2019 IOP Conf. Serial.: Mother. Sci. Eng. 589 012008

[7] A Protopopov et al 2019 IOP Conf. Serial.: Mother. Sci. Eng. 589 012003

[8] K Abramov 2019 IOP Conf. Serial.: Mother. Sci. Eng. 589 01201

[9] Petrov A. I. Fraz A. S. Gavryushina O. S. Hydrodynamic modeling of flow in the flow part of a centrifugal pump running on a high-viscosity liquid // Hydraulics 2018.- #5

[10] Lomakin V. O. Petrov A. I. Stepanyuk A. I. Optimization of geometric parameters of the oil main pump discharge of the NM type // Science and education: scientific publication of Bauman Moscow state technical University 2012-no. 3

[11] Lomakin V. O. Artemov A.V. Petrov A. I. Determining the influence of the main geometric parameters of the pump discharge NM 10000-210 on its characteristics // Science and education: scientific publication of Bauman Moscow state technical University 2012. — № 8

[12] m. Saprykina and V. Lomakin 2019 IOP Conf. Serial.: Mother. Sci. Eng. 589 012017

[13] Lomakin V. O. Cheremushkin V. A. Influence of the shape of the impeller blades on the pressure of the centrifugal pump // Engineering Bulletin 2016. - no. 1

[14] inCheremushkin and Polyakov 2019 VGD Conf. Serial.: Mother. Sci. Eng. 589 012001

[15] withBoyarshinov et al. 2019 VGD Conf. Serial.: Mother. Sci. Eng. 589 012014