Optimization of the output device of a disk pump for high viscous fluid

V Cheremushkin1,2 and A Polyakov1
1Bauman Moscow State Technical University, 5 Second Baumanskaya Street, Moscow, 105005, Russian Federation
2Email: vcheremushkin@bmstu.ru

Abstract
In this paper the problem of designing an output device for dynamic pumps operating at a high fluid viscosity on the example of a disk pump was considered. The main objective of the study was to increase the pump efficiency by optimizing the geometry of the wet part of its volute. The dependences of the efficiency on the geometrical parameters of the output device estimated section were obtained. The results of the study are represented in a form of graphs and scalar field distributions.

Introduction
A disk pump is a friction pump which moving elements make a fluid move via viscosity forces. Thus, the mechanical energy of the fluid increases due to the viscous interaction[1].

A disk pump (Fig. 1) consists of an impeller 1 located in housing with a discharge device 3. The impeller 1 consists of a set of flat disks, which are connected to each other along the periphery.

Figure1. Disk pump construction.
Disc pumps are capable to pump highly viscous liquids, liquids containing abrasive inclusions, and also may provide gentle pumping, i.e. not damage the particles within the pumped medium. These qualities give advantages to disc pumps in comparison with more common centrifugal pumps in some industries [1-3].

The methods of calculation available at current stage of technical development are not quite comprehensive. In addition, experimental data on operation of disk pumps at high viscosity are few and do not provide a complete description and explanation of the transfer process itself [4,5]. This problem leads to difficulties in the designing process and characteristics prediction of disk pumps[6].

In this work, the question of designing a volute output for a disk pump operating at a high viscosity (3000 cSt) is considered. The existing methods of diverting devices designing for centrifugal pumps are focused on the turbulent flow of a fluid, where thickness of a boundary layer is small compared to the channel width and the friction force is not significant[7,8].

In the case of a highly viscous fluid, this condition is not satisfied, and a disk pump with such a discharge device has unsatisfactory efficiency. A possible solution to the problem is to increase the output devise flow capacity.

Mathematical model and methods
The task of designing the diverting device is solved by means of CFD (computational fluid dynamics). The implementation of the numerical simulation method is based on solving the basic hydrodynamic equations discrete analogs [9-11]. In case of an incompressible fluid model, when the density is constant, these are:

Continuity equation

\[ \frac{\partial \bar{u}_j}{\partial x_j} = 0, \]

where \( \bar{u}_j \) - j axis projection of an averaged value of the fluid velocity (\( j=1,2,3 \));

The momentum conservation equation (Reynolds averaged):

\[ \rho \left[ \frac{\partial U_i}{\partial t} + U_j \frac{\partial U_i}{\partial x_j} \right] = -\frac{\partial P}{\partial x_i} + \frac{\partial}{\partial x_i} \left[ \bar{T}^{(v)}_{ij} - \rho u_i u_j \right]; \]

where \( U, P \) - averaged velocity and pressure;

\( \bar{T}^{(v)}_{ij} = 2\mu\bar{s}_{ij} \) - viscous stress tensor for incompressible fluid;

\( \bar{s}_{ij} = \frac{1}{2} \left[ \frac{\partial \bar{u}_i}{\partial x_j} + \frac{\partial \bar{u}_j}{\partial x_i} \right] \) - instant strain rate tensor;

\( \rho u_i u_j \) - Reynolds stresses.

The introduction of the Reynolds averaged Navier-Stokes equation makes the equations system not closed, as additional unknowns, Reynolds stresses, appear.

To solve this system of equation, the k-\( \omega \) SST turbulence model was used. It introduces the necessary additional equations: the transport equations for the turbulent kinetic energy and the relative dissipation rate of this energy:

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

\[ \frac{\partial \bar{\omega}}{\partial t} + U_j \frac{\partial \bar{\omega}}{\partial x_j} = \alpha \cdot S^2 - \beta \cdot \omega^2 + \frac{\partial}{\partial x_j} \left[ (v + \sigma_\omega \omega T) \frac{\partial \bar{\omega}}{\partial x_j} \right] + 2 \cdot (1 - F_l) \cdot \sigma_{\omega 2} \cdot \frac{1}{\omega} \cdot \frac{\partial k}{\partial x_l} \cdot \frac{\partial \omega}{\partial x_i} \]
Despite the fact that the expected type of fluid motion in a disk pump with specified parameters is laminar, the k-ω SST model can be used. This is because of the fact that the generation of turbulent kinetic energy is directly related to flow nature, the magnitude of the velocity and its derivatives. That is, in case the flow regime is essentially laminar, the generation of turbulent kinetic energy and the value of the turbulent viscosity are automatically reduced to very small values. The paper further confirms the possibility of using the k-ω SST turbulence model in the calculation.

When selecting the optimal geometric parameters of the diverting device, a combination of global (stochastic) and local (directional) search was used [12]. The range of parameters change is given in Table 1.

Table 1. Geometric parameters change ranges

| Parameter | Minimum value, м | Maximum value, м |
|-----------|------------------|------------------|
| Width W   | 0,0165           | 0,0760           |
| Height B  | 0,0370           | 0,0754           |

To achieve maximum efficiency of the output, losses in the diffuser of the output are necessary to be minimal. That is achieved at a diffuser angle of 10°. Thus, the output diameter of the pump is defined as dependent on the parameters of the design section of volut.

\[
D_{out} = \sqrt{\frac{4 \cdot W \cdot B}{\pi}} + 2 \cdot L_d \cdot \tan(5°),
\]

where \( D_{out} \) — pump outlet diameter;

\( L_d \) — diffuser length.

Figure 2 shows the computational grid, which has polyhedral cells in the flow core, and prismatic cells near the solid walls [13].
Discussion and results
Research of the pump characteristics was carried out with the following parameters:
\[ Q = 20 \frac{m^3}{hr} \] – pump volumetric flow,
\[ n = 2900 \text{ rpm} \] – rotation speed,
\[ \nu = 3000 \text{ cSt} \] – fluid viscosity.
Table 2 shows the values of the geometric parameters of the models, as well as the characteristics obtained from the results of numerical simulation.

Table 2. Parameters and characteristics of the calculated models.

| №  | W, m  | B, m  | D_out, m | Efficiency | Head, m | Area, m² | Volumetric capacity, m³ |
|----|-------|-------|----------|------------|---------|----------|------------------------|
| 1  | 0.0190| 0.0600| 0.1112   | 0.2725     | 26.8702 | 0.0011   | 0.0094                 |
| 2  | 0.0378| 0.0600| 0.0892   | 0.2719     | 26.7614 | 0.0023   | 0.0188                 |
| 3  | 0.0378| 0.0600| 0.0850   | 0.2717     | 26.5076 | 0.0023   | 0.0188                 |
| 4  | 0.0384| 0.0600| 0.0888   | 0.2713     | 26.7810 | 0.0023   | 0.0190                 |
| 5  | 0.0618| 0.0600| 0.0970   | 0.2713     | 26.8395 | 0.0037   | 0.0306                 |
| 6  | 0.0384| 0.0639| 0.0900   | 0.2699     | 26.7666 | 0.0025   | 0.0199                 |
| 7  | 0.0578| 0.0600| 0.0879   | 0.2699     | 26.7367 | 0.0035   | 0.0287                 |
| 8  | 0.0652| 0.0639| 0.1136   | 0.2689     | 26.7594 | 0.0042   | 0.0339                 |
| 9  | 0.0207| 0.0600| 0.1014   | 0.2687     | 26.6817 | 0.0012   | 0.0103                 |
| 10 | 0.0749| 0.0600| 0.1106   | 0.2678     | 26.7769 | 0.0045   | 0.0371                 |
| 11 | 0.0652| 0.0524| 0.0852   | 0.2677     | 26.5665 | 0.0034   | 0.0291                 |
| 12 | 0.0378| 0.0754| 0.1141   | 0.2673     | 27.0433 | 0.0029   | 0.0222                 |
| 13 | 0.0372| 0.0524| 0.0848   | 0.2671     | 26.5835 | 0.0019   | 0.0166                 |
| 14 | 0.0327| 0.0408| 0.0826   | 0.2640     | 26.5906 | 0.0013   | 0.0120                 |
| 15 | 0.0760| 0.0370| 0.0772   | 0.2629     | 26.1659 | 0.0028   | 0.0257                 |
| 16 | 0.0435| 0.0370| 0.0775   | 0.2617     | 26.0790 | 0.0016   | 0.0147                 |
| 17 | 0.0367| 0.0370| 0.0889   | 0.2603     | 26.4486 | 0.0014   | 0.0124                 |
| 18 | 0.0372| 0.0408| 0.0932   | 0.2601     | 26.5151 | 0.0015   | 0.0136                 |
| 19 | 0.0760| 0.0408| 0.0678   | 0.2367     | 24.1646 | 0.0031   | 0.0279                 |
| 20 | 0.0503| 0.0370| 0.0649   | 0.2219     | 23.1522 | 0.0019   | 0.0170                 |

For better results clarity Figure 3 shows the efficiency dependence on the geometric parameters.
In order to visualize the flow inside the wet part some scalar field functions distributions are shown on fig. 4 and 5 for two volute variations: initial one designed for water by an existing method and the one with enlarged estimated section.

**Figure 3.** Efficiency dependences on the geometric parameters.
Figure 4. Scalar fields for the pump with initial volut.
Figure 5. Scalar fields for the pump with extended volute.
Table 3 shows a comparison of the characteristics of the pump with the initial output device and with the extended one. In addition, to confirm the possibility of using the k-ω SST turbulence model for laminar flows, the data obtained via the calculation of the optimized wet part in the laminar mode are presented.

| №  | Model name       | Mode          | Efficiency, % | Head, m | Radial force, N |
|----|------------------|---------------|---------------|---------|-----------------|
| 1  | Initial model    | Turbulent, k-ω SST | 14            | 15,67   | 293,52          |
| 2  | Optimized model  | Turbulent, k-ω SST | 27,26         | 26,87   | 78,36           |
| 3  | Optimized model  | Laminar       | 27,4          | 26,89   | 77,95           |

Figure 6 shows the distribution of scalar values for an extended diverter, calculated in laminar mode.

Figure 6. Scalar fields distribution for laminar mode.

For a better understanding of the results of the work, characteristics of the pump with initial output device and optimized volute are shown on Figure 7. Red dots mark the efficiency dependence on the pumps flow and the blue ones - the head dependence.
Conclusions
During the work it was shown that the volute designing according to classical methods (mainly for water) leads to a non-optimal design when dealing with high viscosities. This is apparently connected with the fact of speed momentum conservation hypothesis violation. The flow mode inside the wet part of the pump at a viscosity of 3000 cSt is laminar, as it was shown in the work, which causes difference of the velocity profile from that adopted in classical methods of volute designing.

The output device was optimized via two geometrical parameters variation - the width and height of the estimated section with the maximum efficiency criterion. According to the results of the calculations, the head was increased by 71.5 % and the efficiency - by 94.7 %. According to the
presented graphs an obvious efficiency dependence on the height of the estimated section can be observed. The dependences on the width and estimated section volumetric capacity are not so evident, possibly due to the limitation of the varied geometrical parameters.

In the future, it is planned to expand the range of numerical research by adding the geometrical dimensions of the impeller too their optimization parameters.

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