Investigation of hermetic low-speed pumps performance when operating on highly viscous liquids

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Annotation

During the operation of hermetic low-speed vane pumps in a wide range of viscosities of the working fluid, their characteristics change significantly. Experimental dependencies of the effect of viscosity on these characteristics for hermetic vane pumps do not exist. To determine these changes, 3D-models of the flow parts for various pumps are designed. By means of hydrodynamic modeling characteristics for each model are calculated when operating on highly viscous liquids. The results obtained are summarized in the form of relative head ratio and relative efficiency, which determine the performance characteristics of a hermetic pump for various operating modes. A comparison of the results with existing ones is carried.

Introduction

To date, there is no single method for designing centrifugal pumps operating in a wide range of the working fluid viscosities. First, the fundamental works of D.Ya. Sukhanov. [1] should be noted. In his work Sukhanov D.Ya. conducted a series of experiments on typical models of centrifugal pumps operating on viscous fluids. The results were presented in the form of dimensionless coefficients of performance recalculation, allowing based on the existing characteristics of the pump running on water to obtain a new characteristic when working on another viscosity of the working fluid. Also, Sukhanov D.Ya. formulated recommendations for the design of pumps operating in a wide range of viscosities in order to improve its energy efficiency. High efficiency is one of the key requirements imposed on manufactured machines [2]-[4]. Since Sukhanov studied the performance of typical centrifugal pumps these methods are not applicable for designing low-speed pumps. In such pumps hydraulic friction in relatively narrow channels of the stators significantly affects the performance of the device.

Recently, technical requirements to modern mobile installations have increasingly stiffened and can only be satisfied with introduction of up-to-date simulation methods [5]-[8]. Lately the development of computer technology in the field of CFD modeling has made it possible to accurately predict hydrodynamic processes occurring during the operation of centrifugal pumps [9]-[15]. With the help of hydrodynamic modeling, it became possible to design pumps without conducting multiple full-scale tests.

Method
In this work the main task was to study changes in the characteristics of low-speed hermetic pumps when operating on highly viscous liquids by means of hydrodynamic modeling with the aim of reducing the number of full-scale tests [16]. In order to fulfill it, a two-stage pump design with shrouded impellers and a discharge device in a form of a three-channel guide vane was developed. A 3D - model of the pump flow part is shown in Fig. 1.

![3D model of the pump flow part](image)

**Fig. 1.** 3D – model of the pump flow part.

Nominal specifications of the pump are the following:

\[ H = 40 \text{ m}; \quad Q = 10 \frac{1}{\text{min}} = 1.67 \cdot 10^{-4} \text{ m}^3 / \text{s}, \quad n = 2900 \text{ rpm} = 48.3 \text{ rps} \]

Working fluid: ethylene glycol antifreeze.

The required range of the kinematic viscosity of the working fluid:

\[ \nu = [4;800] \cdot 10^{-6} \text{ m}^2 / \text{s} \]

As the object of the study 4 pump models with extended impellers and guide vanes were additionally designed [17], [18]. Parameters of the configurations are presented in Table 1 and Table 2.

**Table 1.** Impeller dimensions: D2 – diameter of the impeller at the outlet; b2 – width of the meridional section channel on the D2 diameter; D1 – diameter of the impeller at the inlet; db – diameter of the barrel; z – number of impeller blades.

| Model # | D2, mm | b2, mm | D1, mm | db, mm | z  |
|---------|--------|--------|--------|--------|----|
| 1       | 122    | 2      | 25     | 12     | 12 |
| 2       | 122    | 3      | 30     | 12     | 12 |
| 3       | 122    | 4      | 35     | 12     | 12 |
| 4       | 122    | 5      | 40     | 12     | 12 |
| 5       | 122    | 6      | 45     | 12     | 12 |

**Table 2.** Guide vane dimensions: b3 – width of the spiral channel; h1 – height of the spiral channel; h2 – height of the diffuser; h3 – height of the shifting channel; h4 – height of the return channel.

| Model # | b3, mm | h1, mm | h2, mm | h3, mm | h4, mm |
|---------|--------|--------|--------|--------|--------|
| 1       | 4      | 4,2    | 8,4    | 3      | 15     |
| 2       | 5      | 4,2    | 8,4    | 4,6    | 16,3   |
| 3       | 6      | 4,2    | 8,4    | 5,7    | 18,3   |
Numerical modeling is based on the equation of continuity, the Navier-Stokes Reynolds averaged equations and two additional differential equations that are responsible for modeling turbulence structures. In this case k-ω SST turbulence model was used [19]. A structured prismatic computational grid with 5 layers near solid walls and an unstructured multi-faceted mesh in the flow core were used to solve the equations. The number of mesh cells for each model was more than 450 000, Fig.2.

Fig. 2. Calculation grid for the pump under consideration.

Results
For all models normal characteristics were calculated. Obtained data was represented as the following dependencies \( h(q) \) and \( \eta(q) \), where \( h = H / (nD)^2 \) - dimensionless head, \( q = Q / nD^3 \) - dimensionless flow rate. These dependences are presented in Fig. 3 and Fig. 4.
Fig. 3. Dependence of the dimensionless head on the dimensionless flow rate for the models under consideration with kinematic viscosity $6 \cdot 10^{-6} \, m^2 \, / \, s$

Fig. 4. Dependence of the efficiency on the dimensionless flow rate for the models under consideration with kinematic viscosity $6 \cdot 10^{-6} \, m^2 \, / \, s$
Fig. 5. Dependence of the dimensionless head and efficiency on the dimensionless flow rate for the models under consideration with kinematic viscosity \( \nu = 400 \cdot 10^{-6} \text{ m}^2 / \text{s} \)

By viscosity of the working fluid increasing, pump characteristics begin to drop significantly, Fig. 5 and Fig. 6.

H'(Re) characteristic (Fig. 7) represents the dependence of the relative head ratio on the blade Reynolds number: \( h' = h_v / h_\omega \), where \( h_v \) — head calculated when the pump is operating on a viscous fluid, \( h_\omega \) - head calculated when the pump is running on water.

\( \eta'(Re) \) characteristic (Fig. 8) represents the dependence of the relative efficiency on the blade Reynolds number: \( \eta' = \eta_v / \eta_\omega \), where \( \eta_v \) — efficiency calculated when the pump is operating on a viscous fluid, \( \eta_\omega \) - efficiency calculated when the pump is operating on water.
It can be noted that the characteristic of Sukhanov D.Ya. differs from the results of the simulations, Fig. 7 and Fig. 8. Since the width of the impeller and the guide vane flow part of low-speed pump are relatively small, hydraulic losses are more significant there [20] - [22].

![Graph](image)

**Fig. 7.** The dependence of the relative head ratio on the blade Reynolds number.
Fig. 8. The dependence of the relative efficiency on the blade Reynolds number.

Discussion
Simulation has shown that with an increase in the viscosity of the working fluid, the pressure characteristic of an ultra-low-speed vane pump drops significantly faster than that of pumps with greater specific speed. At the same time, the pump efficiency in the design mode decreased from 55 % to 21 %. Increasing the guide vanes channel width managed to extend the pump operation range by 116 % with the viscosity of the working fluid up to $800 \times 10^{-6} \text{m}^2 / \text{s}$.

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