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Belgrade, Serbia, October 26–28, 2011

Proceedings of the 4th International Meeting on

CAVITATION AND DYNAMIC PROBLEMS IN HYDRAULIC MACHINERY AND SYSTEMS

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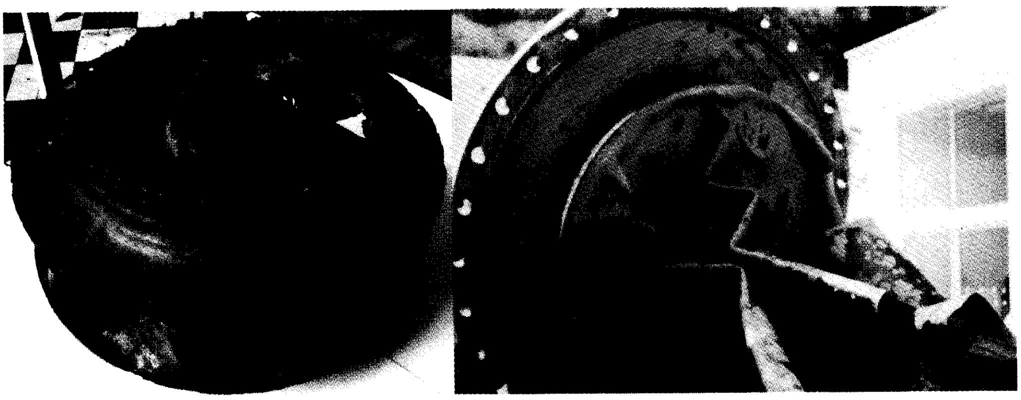

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Investigation of Small-Sized Axial-Flow Stage of a Borehole Pump for Water Supply

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Abstract

The article presents the results of investigation of small-sized axial-flow low-speed stage, used in a new design of a borehole pump for water supply. Using numerical simulation, the authors have got the velocity distributions along the radius of the flow passage of the investigated stage at near-optimum modes. Pressure distributions on the blade surfaces at various pressures at the impeller inlet of the considered stage were also obtained. The article presents the results of testing of the pump using an experimental rig, together with comparison of the experimental and simulated performance curves. Numerical research was carried out using the ANSYS CFX software. Basing on the results of the research, the conclusion concerning expediency and prospects of application of small-sized axial-flow stages in borehole pumps for water supply are made, considering their advantages in comparison with mixed-flow stages.

Keywords: borehole pumps, water supply, numerical simulation, small-sized axial-flow low-speed stage, cavitation, NPSH

1. Introduction

Every day, about a billion people experiences lack of drinkable water [1]. On one side, it is due to increase of population and very non-uniform distribution of stocks of drinkable water. On the other side, it is caused by the human activity leading to increasing pollution of surface sources. That's why, an increasing attention is paid to the water supply from artesian boreholes, due to relative purity of the water supplied, convenience of their arrangement, efficiency of operation and independence of central sources. It is evident that due to water deficit and deterioration of state of surface wells, the role of water supply from boreholes will only increase.

2. Problem statement

In general, the price for a system for water supply from boreholes consists of:

- costs for construction of a borehole, including boring. According to [2], these costs grow together with the borehole diameter, approximately in cubic dependence of it;
- expenses for operation of a pump unit, that consist of components indicated in Fig. 1 c and are determined mostly by the cost of the consumed energy [3].

Thus, it is economically expedient to reduce the borehole diameter or increase the pump capacity keeping the same borehole diameter, provided the pump efficiency is kept at a high level attained for smaller capacities (Fig. 2). Fig. 1 a and b demonstrate the examples of pump designs with high capacities and high n_s of foreign [4] and Ukrainian [5] production. They are featured, firstly, with complicated shape of impeller blades and guide vanes, and secondly, rather large axial dimensions. These factors make negative influence on the pump weight, technological quality of its design and cost of its production [4-9]. Further increase of capacity of a mixed-flow pump with restricted radial dimension inevitably leads to reduction of the pump efficiency due to too unfavorable shape of its flow passages.

As an alternative for mixed-flow stages, when increased capacity of borehole pumps within limited radial dimensions is required, it is expedient to apply a small-sized axial-flow stage developed at the department of applied fluid mechanics of Sumy State University (SSU) [10-13], which has in this case several advantages.

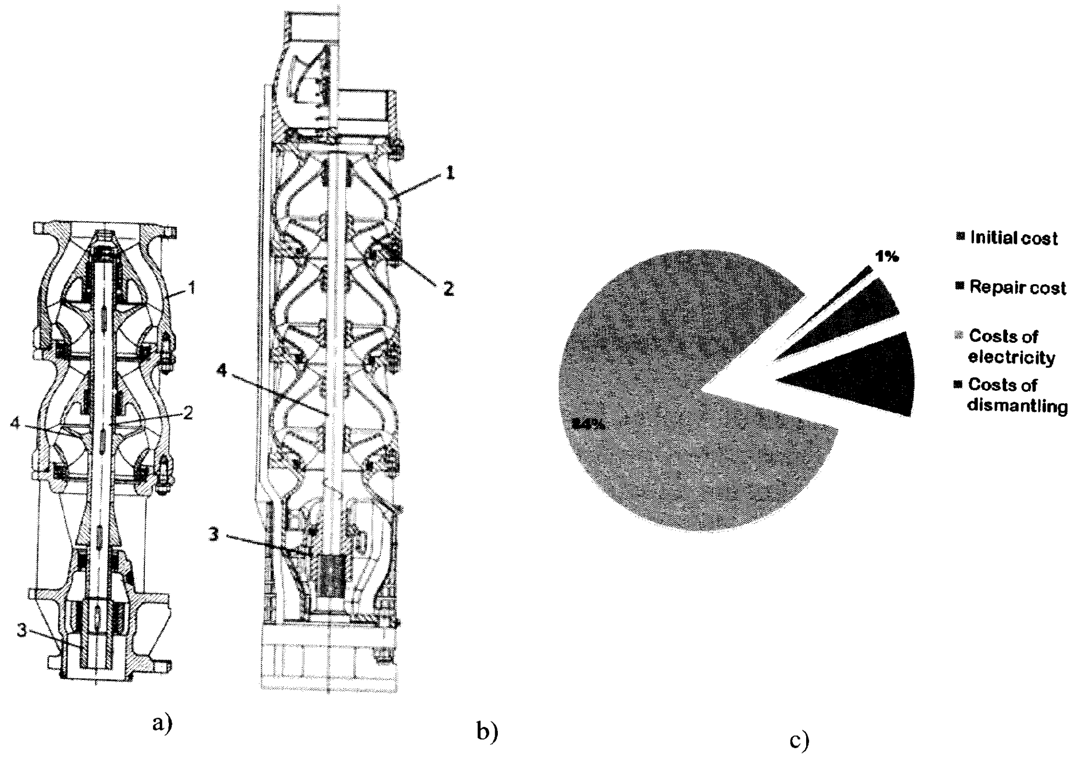


Fig. 1 Design of a borehole pump: a) Ukrainian production [5]: 1 - vaned diffuser, 2 - shaft, 3 – coupling, 4 - impeller; b) foreign production [4]: 1 – vaned diffuser, 2 - impeller, 3 – coupling, 4 – shaft; c) Life Cycle Costs of borehole pump for water [3]

It is known [14] that at large relative capacities axial-flow pumps are featured with most simple and compact design, which is especially important for the well conditions, and provide the highest hydraulic efficiency of all the types of centrifugal pumps. But the decisive advantage of the axial-flow stages in this case is a theoretical possibility to further enhance capacity (1.5 – 1.7 times) as compared to existing designs with a fixed outer diameter of the pump. For example, for a conditional 10-inch size, we have developed stage with the capacity of 480 m^3/h, n_s = 500 at 82% efficiency. As far as we know, no companies produce standard pumps in that conditional dimension with such capacity. And if in the region n_s> 400 the axial-flow stage has no competitors, at lower n_s, particularly in the area n_s = 250-350, mixed-flow and axial-flow stages can be considered competitive. Each of them has its advantages and disadvantages, and application of a certain type of stages should be based on careful analysis. This article is devoted to clarifying of the lower limit of the applicability of the axial-flow stages concerning the specific speed as opposed to mixed-flow stages.

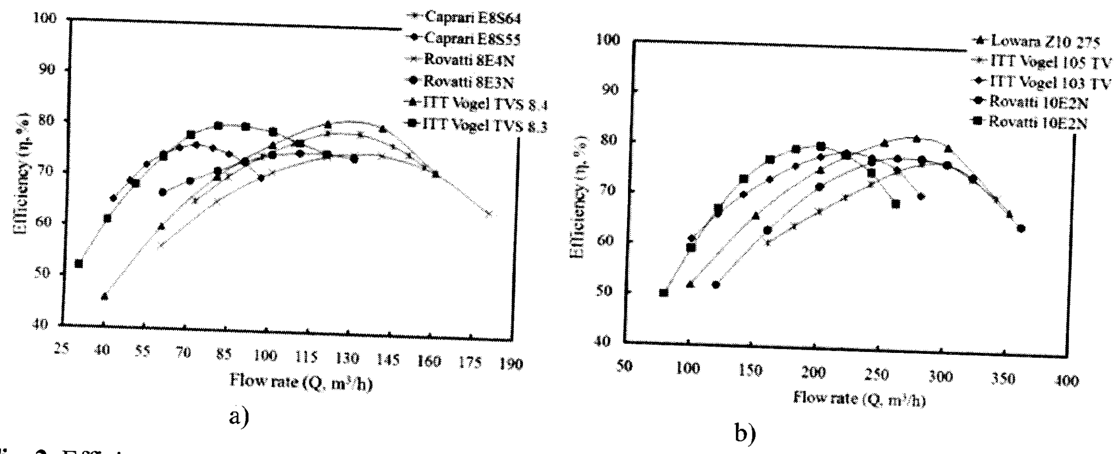


Fig. 2 Efficiency of stages of borehole pumps: a) conditional size 8 inches, b) conditional size 10 inches.

As the main restricting criteria for identifying the lower limit of n_s, below which the use of axial-flow stages is impractical, efficiency of the axial-flow stages can be adopted in this case, as compared with analog stages of mixed-flow type. Note that in general the formula for calculating n_s has three variables (rotational speed, capacity and head). Taking into account that the frequency of rotation is determined by the drive motor and has some almost constant value, and the peak value of the axial-flow head level is limited, we can assume that in this case n_s depends primarily on one variable, which is the stage capacity. Accordingly, for a variation of the specific speed of stage it is enough to change only its capacity.

3. Analysis of previous studies

Axial-flow pumps have long been known and are widely used in practice. In particular submersible axial-flow pump designs are often used for water intakes and treatment plants. They are featured with simple design and compact size, low weight, ability to transport liquids contaminated with non-abrasive impurities.

Multistage axial-flow pumps are used in missiles in the redox-oxygen units of liquid rocket engines [15]. Such pumps are characterized by large circumferential speed, high values of specific work, harsh working environments, and most importantly, increased demands on the cavitation resistance.

There are also attempts to use multi-stage axial-flow pumps in the petroleum and mining industry for pumping multiphase fluids [16-18].

As far as we know, the first attempt to create a multi-stage axial-flow pump (with cylindrical blades) for wells were performed by Papier [14, 20], but for various reasons they have remained at the level of individual prototypes.

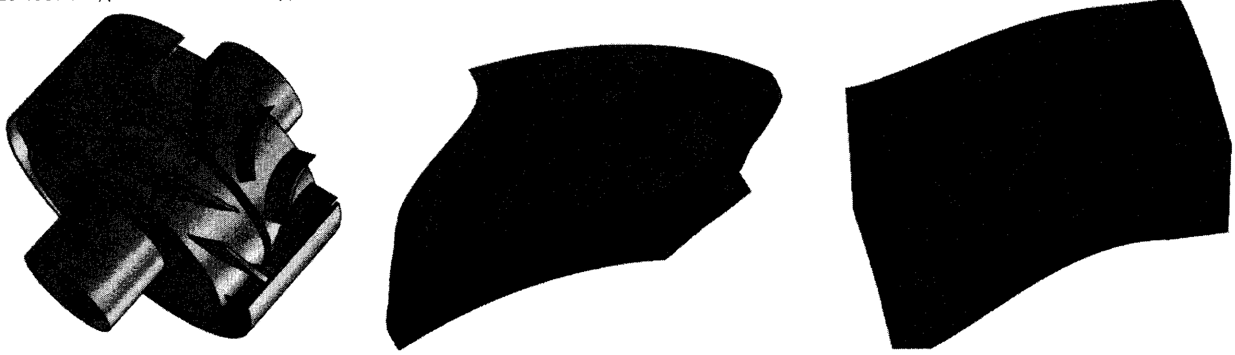
4. Subject and procedure of research

According to the results of preliminary calculations, in order to conduct experimental tests, the impeller (IMP) and stator apparatus (SA) of axial-flow stage were developed. A three-dimensional model of the stage is shown at Fig. 3 a, its main parameters are listed in Table 1. Note that the stage was designed from the requirement of ensuring the largest possible head per the stage to reduce the number of stages in the pump, and, accordingly, its size, weight and cost. As can be seen from Table 1, the investigated stage has almost the largest value of the head coefficient reachable for axial-flow stages (which according to [21] is K_H = 0,25 - 0,27) and very low flow coefficient. Using such a combination of the head and flow coefficients leads to an increase in hub ratio and blade angle of the impeller, which unfavorable affects the level of efficiency, which, as noted above, was adopted as the main limiting factor. According to preliminary calculations, this combination of parameters is close to the limit and will identify the border of applicability of the stages concerning the values of n_s.

Table 1 The main parameters of the test stage

	IMP	SA
External diameter D_2, mm	155	155
Number of blades z	8	15
Specific speed n_s	267	
Head coefficient K_H	0,249	
Flow coefficient K_q	0,185	

In a stage, in both the impeller and the stator, blade apparatus of three-dimensional blade profiles were used. Their design was carried out using both the known techniques [14] and qualifying coefficients. Photo of an impeller of the test stage is shown at Fig. 4 a.



a) b) c)

Fig. 3 a) 3D model of test stage, b) computational mesh of impeller, c) computational mesh of stator apparatus

4.1. CFD procedure

The numerical simulation in this study was performed in the software package ANSYS CFX, the university version of which is available to the SSU. To correctly simulate the conditions of the intermediate stage of the pump, a multi-connected computational domain was used, that consisted of an input element, three impellers and stator apparatus and an output element. All the results, both integral values and distributions along the radius, were obtained for the intermediate of three stages. In order to save computer resources, the geometric model used for simulation represents only a part of the total volume of liquid.

The computational mesh in the impeller consisted of about 800 thousand prismatic cells, in the stator apparatus - about 600 thousand cells. Before the numerical simulation, a verification of the mesh independence separately for each element of the computational domain was carried out. For this purpose, we constructed a mesh with different density: the first mesh - the base, the second - diluted (a cell doubled in comparison with the base mesh) and the third - thickened (cells are reduced in half compared with the base mesh). Analysis of the integral values obtained from numerical simulations for different meshes showed that the difference in the results was no more than 1%, indicating the mesh independence. Further numerical simulations were carried out for the basic computational meshes.

The value of the variable y + ranged from 10 to 100 units, which corresponds to the guidelines described in the user manual [22].

When calculating the energy and head characteristics of the stage, numerical simulation was carried out in a range of capacities from 0,8 to 1,2 Q_{opt}. To simulate the turbulence, a standard k - ε turbulence model with scalable near-wall functions has been used. The calculation was performed for both steady state (interface Stage), and transient approach (interface Transient rotor-stator). Since the difference in the integral characteristics obtained with different approaches was negligible (around 1%), and the transient calculation requires significantly more computation time, we have further used the steady state approach.

For modeling of cavitation phenomena, the model of Rayleigh Plesset was used. The calculation was performed for five capacities: 0,8 Q_{opt}, 0,9 Q_{opt}, Q_{opt}, 1,1 and 1,2 Q_{opt}, for each capacity the pressure at the inlet of the computational domain was gradually reduced in order to get values of NPSH.

4.2. Test procedure

Experimental tests of the small-sized axial-flow stage were carried out at the test rig, working on a closed circuit fluid circulation, the layout of which is shown in Fig. 4 b. All the characteristics of stage were obtained at the rotation speed of 2910 rpm. Comparatively small size and low values of power consumption of test stages do not provide enough high-quality experimental data while working with one stage. Also performances of the first and intermediate stages may differ considerably due to the influence of swirl flow at the inlet of the impeller intermediate stage [23]. Therefore, tested were the assembly units which consisted respectively of three and two (intermediate dismantled) stages.

Energy and head characteristics of the dismantled stage were obtained as the difference between the characteristics of the above described assembly units. To exclude the effect of hysteresis on the obtained characteristics of the pump stage, tests were carried out in the direction from the maximum capacity to zero, and then in the opposite direction.

Methods of testing the pump unit and the measurement error were according to [23, 24]. Dependence of the efficiency of the pump on the capacity was obtained by dividing the efficiency of the unit by the engine efficiency.

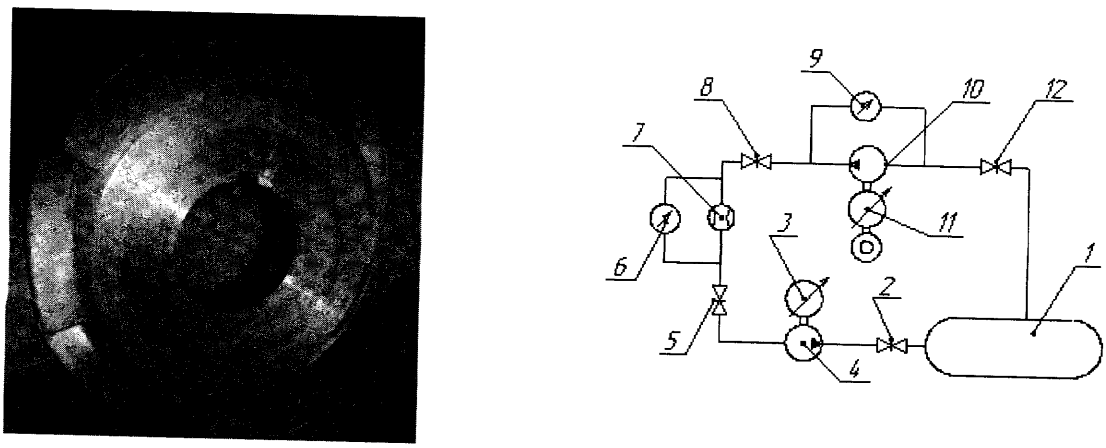


Fig. 4 a) impeller of test stage, b) layout of the test rig for small-sized axial-flow stages testing: 1 – tank, 2 – valve, 3 – motor, 4 – auxiliary pump, 5 – valve, 6 – differential manometer, 7 – orifice plate, 8 – valve, 9 – differential Manometer, 10 – experimental device, 11 – motor with torque meter, 12 - valve

5. Basic results

Integral characteristics of the stage, obtained at the test rig and as a result of numerical simulation are shown at Fig. 5 a. All the results presented below relate to the intermediate stage, if it is not otherwise indicated. Analysis of characteristics shows that the discrepancy between the results of numerical simulation and the experimental results concerning the head is about 3% with a predictable over-head at numerical simulation.

As for efficiency, it follows from the Fig. 5 a that the efficiency curve obtained in tests on the test rig provides a more narrow range of operating conditions, than that obtained by numerical simulation, while for the maximum efficiency values, acceptable discrepancy of 4% is observed. The most probable cause of this behavior of the experimental efficiency curve is the discrepancy between the geometric model of blade systems used in the calculation, and actually observed in the manufactured prototype, in particular shapes of input and output edges, the roughness of the flow passages, etc. As is evident, despite the adverse combination of the head and flow coefficients, we have obtained a sufficiently high efficiency level.

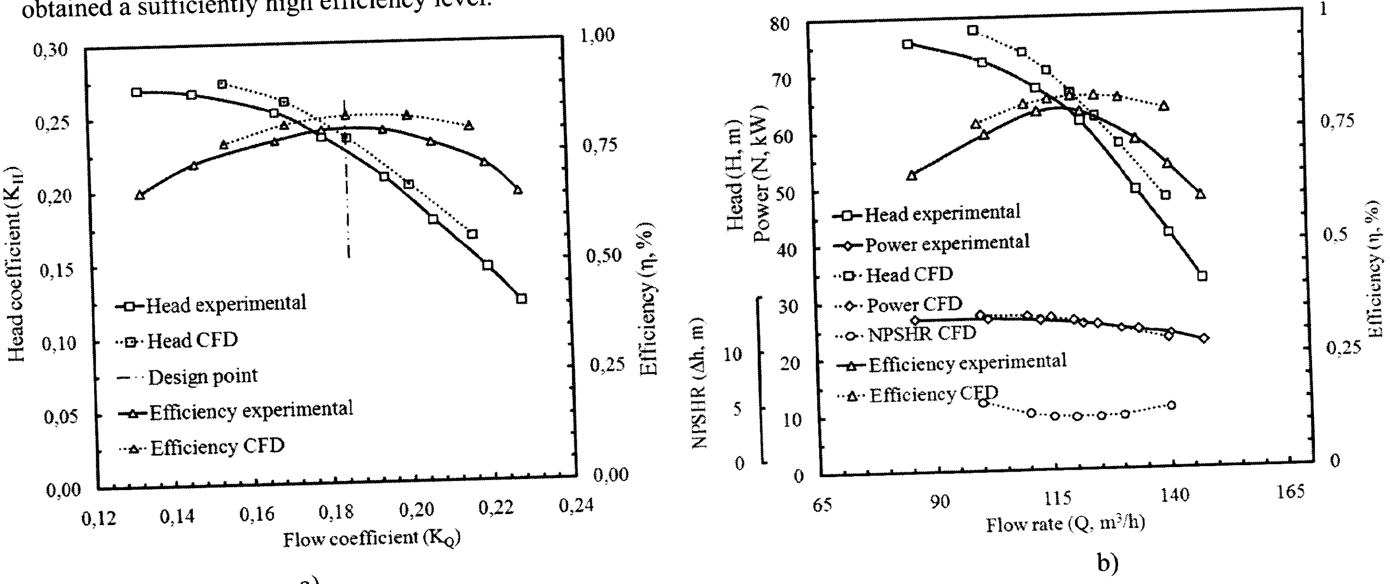


Fig. 5 a) the characteristics of stage, obtained by numerical simulation are marked by dashed lines, the characteristics of stage, obtained on an test rig - solid lines; b) the characteristics of the pump, obtained by numerical simulation are marked by dashed lines, pump performance, obtained on an test rig - solid lines;

Fig. 6 shows obtained by numerical simulation diagrams of axial and tangential component of the absolute velocity averaged over a pitch at the inlet and outlet of the intermediate IMP, related to the corresponding average velocity through the flow passage of IMP. These diagrams allow to assess qualitatively the flow pattern in the flow passages of the stage. Straight lines on the diagram show the velocity distribution assumed in the design of the impeller. Thus Fig. 6 indicates a significant deviation of real diagrams of the axial and tangential component of the absolute velocity at the inlet of the impeller from design values, due to the influence of the previous stage. As can be seen from Fig. 6, at the outlet of the impeller at the periphery of the blade there is a zone of considerable size (about 25% of the height of the blade) with a reduced axial component of absolute velocity, which appear due to blade tip effects in the interaction of the main flow with the stream flowing through the radial clearance. Simultaneously, a zone of lower radii is featured with an increase of axial component of absolute velocity, which is on the relative radius of about 0.2 reaches a maximum value.

Uneven diagrams of tangential component of the absolute velocity are even sharper, and demonstrate quite clearly the influence of restricting walls, near which the circulation increases substantially compared with the circulation in the middle cross-sections.

Thus, the flow pattern in the flow channels of the stage under consideration even under optimal conditions has a complex three-dimensional character, variation of flow parameters along the span is expressed more strongly in comparison with conventional axial flow pumps, and virtually none of the profile cross-sections does work in the design conditions, which greatly complicates the design and development of stages of this type.

Further decreasing of specific speed (which is equivalent to reducing the flow coefficient) is impractical because it leads to further growth of the blade angles of the impeller, growth of internal angles of diffuser channels and increase in hydraulic losses, which would inevitably reduce the value of stage efficiency [25, 26].

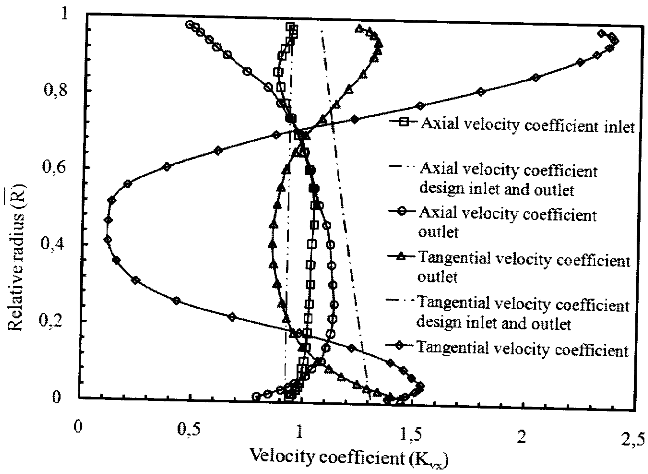


Fig. 6 Diagram of velocity coefficients

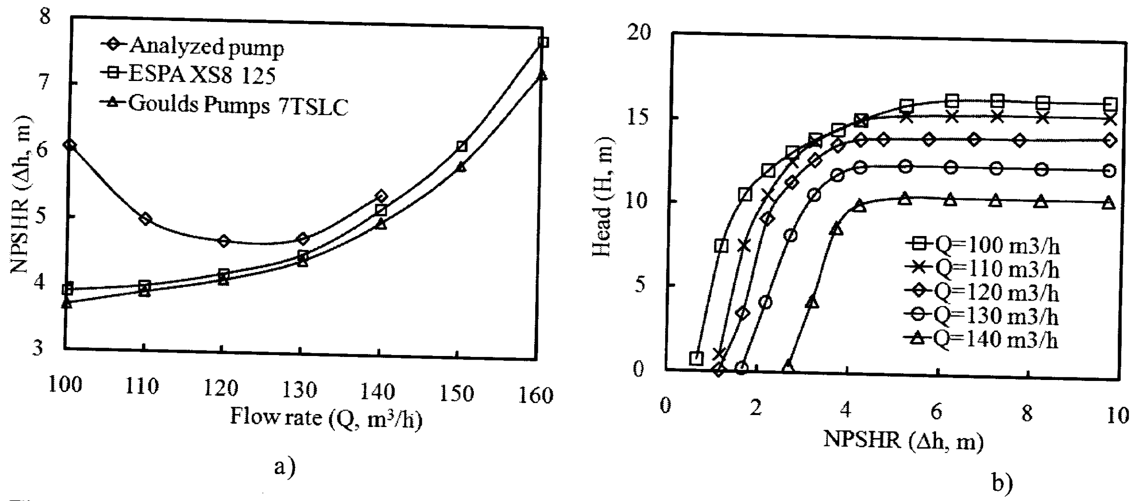


Fig. 7 a) Cavitation performance of the test pump and analogues; b) set of cavitation curves of test stage

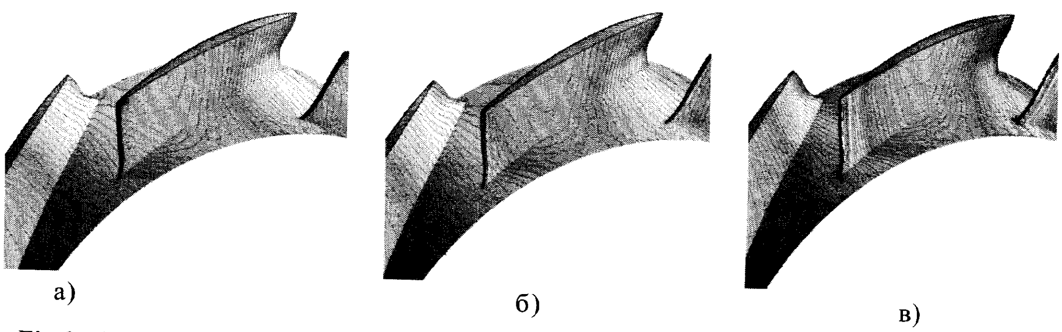


Fig. 8 Distribution of static pressure on impeller blades a) Q = 100 m^3/h, б) Q = 120 m^3/h, в) Q = 140 m^3/h

Fig. 5 b shows the experimental and numerical simulation characteristics of the pump (Q_{opt} = 120 m^3/h). As one can see, we have been obtained the value of pump efficiency, which is only slightly lower than the pump efficiency of the leading manufacturers (Fig. 2). It should be noted that the numerical simulation curves show the displacement of the maximum efficiency mode to higher flow rates compared with the design values, and with further increasing of flow rate the differences between the curve of simulation results and the experimental curves increases as well. Analyzing the head characteristics of the pump shown in Fig. 5 b it can be noted that the discrepancy between the results of numerical simulation and the experimental results at the optimum is no more than 4 %.In this study, we have carried out also the numerical simulations aimed at determining the cavitation characteristics of the stage. Fig. 7 a shows the dependence of NPSH from the flow rate, and Fig. 7 b shows a family of cavitation characteristics of the stages at different flow rates.

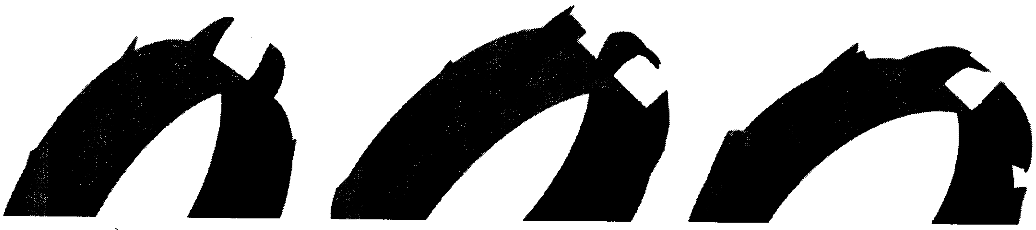


Fig. 9 Cavitation cavities on impeller blades at design flow and lowering inlet pressure: a) P_{inlet}= 0,8 atm, b) P_{inlet}= 0,5 atm, c) P_{inlet} = 0,3 atm.

Comparing the cavitation characteristics of the stage with the corresponding characteristics of similar pump stages of the leading manufacturers, it can be concluded that, despite the fact that this problem has been given low priority in the design of the stage, the obtained characteristics are acceptable.

It should be noted that for a submersible borehole pump cavitation phenomena do not have the such sharpness, which is typical for conventional pumps, since they almost always work with a significant inlet head (typically 4 - 10 m). But for some applications (horizontal installation in basins and reservoirs), the cavitation characteristics are among the determinants.

Fig. 8 shows the time averaged distributions of static pressure on the flow passage surfaces of the impeller, which illustrate the movement of low-pressure zone, representing the risk of cavitation, on the surface of the blade when changing the flow rate.

Fig. 9 illustrates the development of cavitation cavities in the flow passages of the stage for different values of NPSH at the pump inlet, obtained by numerical simulation. It can be seen that when the pressure at the impeller inlet decrease, a gap cavitation is first formed. With further reduction of inlet pressure, profile cavitation develops, covering the back of a profile in the zone of maximum speed. Note that, when the pump was tested on a test rig, it was continued to work steadily, even while reducing the inlet head to 0,5 m, and even so the noise was observed, but significant reduction in operating parameters of the pump was not observed.

6. Conclusions

As a result of numerical simulation and testing of the stage and of the pump in the test rigs, the following results were obtained:

- experimentally confirmed the possibility of using small axial-flow stages with low specific speed (up to n_s = 267) in a submersible borehole pump to obtain acceptable values of the efficiency level of stage (82%) and pump (79%);
- creation of stages with lower values of n_s decrease the flow coefficient, which will lead to further growth of the blade angles of the impeller, increasing of internal angles of diffuser flow passages and the hub ratio, which together will have, as we know, the negative impact on the efficiency level. When well-designed mixed-flow stages of high efficiency are available, it is expedient to use these stages;
- as a result of numerical simulation we have obtained acceptable cavitation characteristics comparable to the cavitation characteristics of known similar mixed-flow stages;
- the researched stage has a number of promising areas for further improvement of both the stage and the pump as a whole, which will be reported in subsequent publications.

Nomenclature

η Efficiency [%]

H Head [m]

K_H Head coefficient ($= \dfrac{H \cdot [m]}{n_{sec}^2 \cdot [rev/s] \times D_2^2 \cdot [m]}$)

K_Q Flow coefficient ($= \dfrac{Q \cdot [m^3/s]}{n_{sec} \cdot [rev/s] \times D_2^3 \cdot [m]}$)

K_{Vx} Velocity coefficient (K_{Vm} – axial, K_{Vu} – tangential)

n Speed of rotation in min [rev/min]

n_s Specific speed ($= \dfrac{3,65n \cdot [rev/\min]\sqrt{Q \cdot [m^3/s]}}{H^{0,75} \cdot [m]}$)

n_{sec} Speed of rotation in sec[rev/s]

NPSH Net positive suction head [m]

Q Flow rate ($Q_{опт}$ – optimum flow rate) [m³/s]

V_x Velocity components (V_u – tangential wise, V_m – axial wise) [m/s]

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