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ANALYSIS OF THE SPATIAL STRUCTURE OF THE FLOW IN HIGH PRESSURE HYDRAULIC TURBINES

Summary. The article discusses the study of the velocity field and pressure conducting on the hydraulic turbine type RO500 (Francis turbine radial-axial 500 pressure) and the basic hydrodynamic equations used in the process of mathematical modeling of turbulent flow are based on the fundamental laws of conservation: mass, momentum, energy. This paper presents the distribution of flow rates before and after the impeller of high-pressure RO500, as well as the loss of flow energy in the spiral chamber and guide vanes.

*Key words**:** HPP, hydraulic turbine, spiral chamber, impeller, Francis turbine, energy.*

Introduction

In the 1950s and 1960s of the last century, in connection with the widespread development of Siberian rivers, as well as the construction of a cascade of hydro power stations in Ukraine, Central Asia, the Caucasus and the Baltic states, the development of experimental and computational-theoretical works in the development of flowing parts and impellers received a great impetus hydraulic turbines of various speeds.

During these years, the domestic nomenclature (industry standard) was developed. The efficiency level of the developed model hydraulic turbines reached 87–89% at the optimum.

With the development of two-dimensional and three-dimensional methods for calculating blade systems in the years 70–80, it was possible to increase the efficiency to 91–92%.

But still, as in the 50–60 years, for example, when developing the impellers of the Krasnoyarsk or Sayano-Shushenskaya hydro power plants, up to 20 impellers and their modifications were tested on a model installation, the experimental refinement of the flow parts and impellers was the main and most laborious part of the job [9-11].

With the widespread introduction of mathematical modeling based on computer technologies into the process, which made it possible to increase the level of optimal efficiency to 93 ÷ 94% (with D1mod = 0.5 m), doubts arose about the importance and need for further experimental studies of the flow structure [2,7].

The paper substantiates the feasibility of experimental studies and their importance for improving the mathematical models used in computer technology.

The relevance of the study

The basic equations of hydrodynamics used in the process of mathematical modeling are based on the fundamental laws of conservation: mass, momentum, angular momentum, energy.

In the process of developing methods for mathematical modeling of turbulent flows, simplifications are introduced into the process which is under consideration.

First, diffusion and convective transport is carried out as the simultaneous transfer of mass, energy, momentum and angular momentum.

Naturally, the desire to take into account in the process of mathematical modeling this complex transfer process. However, it is known that the Navier-Stokes equation is a differential analog of the momentum conservation law, and the continuity equation is a differential analog of the mass conservation law. The law of conservation and transfer of angular momentum are not directly applied in mathematical modeling.

Secondly, a system of time-averaged Reynolds and continuity equations is used to describe the behavior of a turbulent flow.

Averaging the flow over a selected cross section or over a time interval can be carried out in various ways:

For example, when determining the average speed, you can set the task **a)** average speed should provide the same flow rate that occurs in a real stream, or **b)** same amount of movement (momentum), or **c)** same moment of momentum (angular momentum), or **d)** the same kinetic energy.

Of course, these will be different values of the averaged speeds.

Averaging the Navier-Stokes equations (equations of momenta), we obtain an adequate model of force interaction on an elementary liquid volume with a violation of the energy balance and the balance of the angular momentum as applied to this liquid volume.

An attempt to clarify the problem is to use the mathematical modeling of the equations of balance and dissipation of turbulent kinetic energy (the κ–ε model of turbulence).

As known, writing the Navier-Stokes equations in a conservative (energy) form by multiplying term by term (the tensor form of each term) is averaged over energy.

Furthermore, the equation of balance of the total kinetic energy is obtained, and the equation of balance of the averaged kinetic energy is subtracted from this equation.

The influence of the angular momentum on the process, as a rule, is not taken into account.

Subsequent, in the process of implementing the calculation on a computer, the simplifications of the mathematical model are not amenable to complete analysis from the point of view of the introduced errors and the accuracy of the results obtained.

Thirdly, the influence of the boundary conditions is so significant on the results obtained, both in terms of the accuracy of the averaged flow parameters specified in the input section and in terms of such turbulence characteristics as scale, intensity, and other parameters [1].

As a result of the above mentioned, the experimental verification of the flow parameters obtained as a result of mathematical modeling, as well as the verification of the accuracy of setting the boundary conditions, is the main criterion for the accuracy of the calculated results.

Flow structure analysis

In the case of solving the direct problem of the flow around the impeller blades, the actual flow form at the entrance to the blade system, and for some (channel) methods and at the exit from it, it is necessary to know in order to set the boundary conditions. With the known of the distribution of velocities and pressures in the characteristic sections of the turbine cavity, one can find the corresponding averaged characteristics, which is primarily necessary for determining the energy balance of the hydraulic turbine.

The analysis of the energy balance thus compiled allows us to study a number of important issues in the theory of the working process: the degree of influence of individual elements of the flow part and the establishment of the most effective ways to increase speed, energy and cavitation indicators.

The study of the velocity and pressure fields was carried out on a hydraulic turbine bench in the flow part of the radial-axial turbine model with impellers Francis turbine RO500 I-2b and RO500/683 (Russian standard РО500 I-2б и РО500/683) [3-6]. The main elements of a hydraulic turbine model are characterized by the following parameters:

1) Circular spiral camera with a coverage angle in plan φ=345° and the radius of the input section ρ=133mm ($\alpha_{\text{rated}} \frac{V}{\sqrt{H_p}} = 0.6$), right rotation calculated by law $v_u r = const$;

2) Stator with 13 profiled columns (including spiral tooth);

3) Profile of guide vanes normalized, asymmetric, positive curvature, $\frac{D_0}{D_1} = 1.2$; $\frac{b_0}{D_1} = 0.08$; $z_0 = 24$;

4) Impellers (D_1=400 mm): RO500 I-2b (Francis turbine), $z = 17$, $\frac{D_2}{D_1} = 0.686$; RO500/683, $z = 19$, $\frac{D_2}{D_1} = 0.686$;

5) Suction pipe (draft tube) - straight conical with a taper angle 5°, height $h = 3.18D_1$.

The velocity and pressure field in the turbine cavity was measured in two sections: before of the impeller (section *I-I*) (fig.1) and after it (section *II-II*).

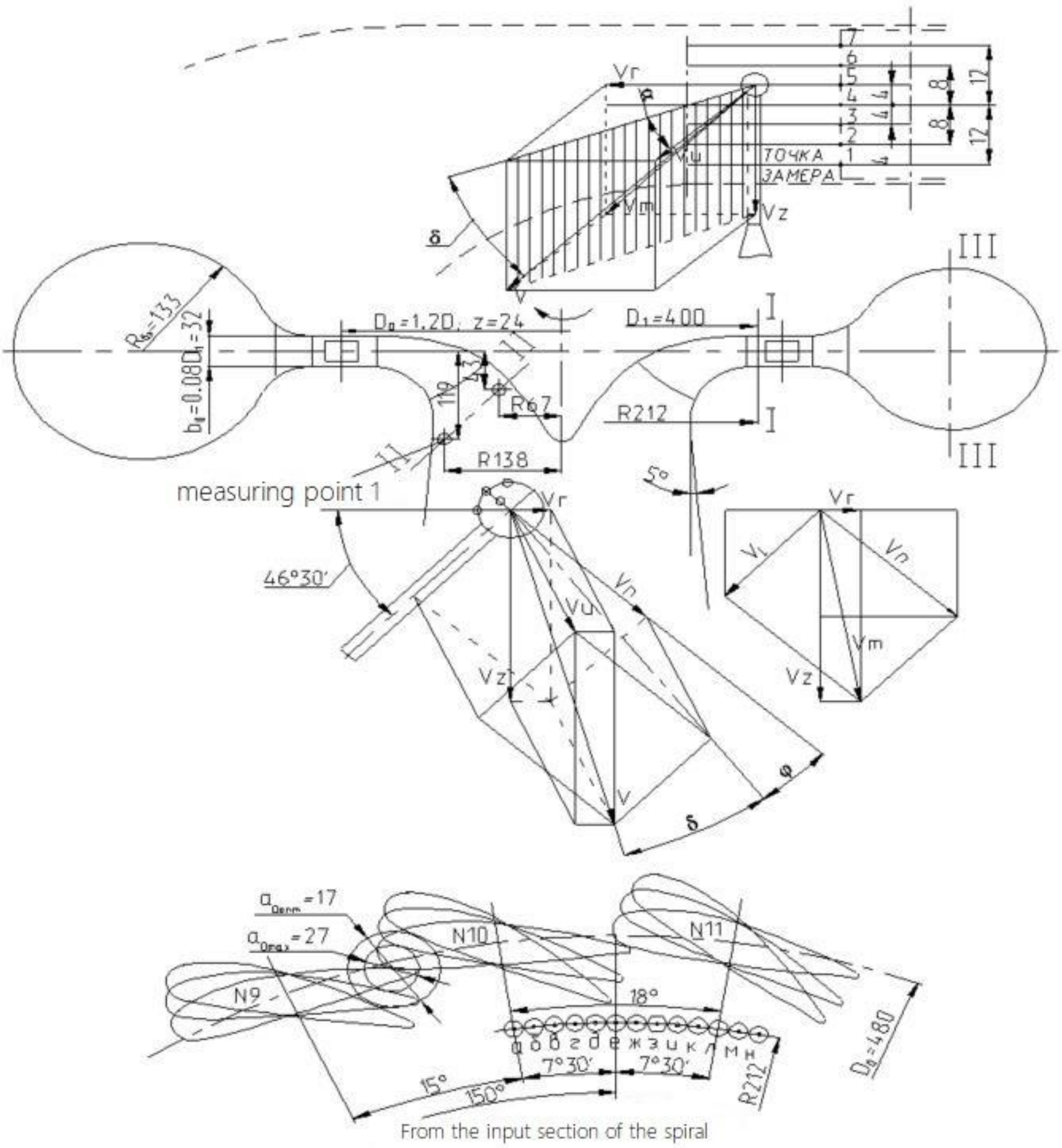


Figure 1. The location of the dimensional sections in the turbine cavity and the decomposition scheme of the absolute velocity vector

In the plan, the cylindrical section *I-I* is located to the output section of the spiral (in the direction of rotation of the impeller) at an angle φ=142°30′÷160°30′ ($\Delta\phi = 18°$) at the impeller RO500 I-2b and at an angle φ=142°30′÷162°($\Delta\phi = 19°30'$) at the impeller RO500/683. Section *II-II* in plan is at an angle φ=276° to the output section of the spiral.

When assessing the influence of the shape of the flow in a spiral on its structure in front of the impeller, it became necessary to additionally study the flow in section *III-III*, which is located at an angle φ=112°30′ to the input section of the spiral and passes through the center of the circle of the section. The flow was studied using probes in the cavity of a hydraulic turbine with an impeller PO500I-2b in 22 heads of pressure, and with the impeller RO500/683 (Francis turbine) – in six heads of pressure $H = 4.5 \div 5.5$m.

The following can be cited as the main regularities of the structure of the averaged flow (Fig. 2 - 6):

1. The energy loss of the flow in the spiral chamber and the guiding vanes for the studied zone of regimes does not significantly depend on the regime of operation of the turbine and amount to about 2 ÷ 3.5%. At the upper and lower rings of the guide vane, the total energy is 1 ÷ 2% less than in the central part.

2. The nature of the unevenness in height of the guide vane averaged in the circumferential direction of the flow velocity is determined primarily by the value of a_0 and does not depend on the number of rotations of the impeller. This is indicated by dependency matching $\frac{\bar{V}u_{I(\phi)}}{Q_I^{'}}$; $\frac{\bar{V}r_{I(\phi)}}{Q_I^{'}}$; $\frac{\bar{V}z_{I(\phi)}}{Q_I^{'}}$ (given to 1 m head and averaged in the circumferential direction) at a_0 =17 mm and differences $n_I^{'}$ ($n_I^{'} = 80$; 75; 65; 55 and 46,5 min^{-1}).

By comparing these graphs with similar ones in the case of a cavity without an impeller or with other impellers, it was established that the pattern of change in speed before of the impeller along the height of the guide vanes is specific for hydraulic turbines of this speed. With increase a_0, uneven values $\frac{\bar{V}_{u_{I(\phi)}}}{Q_I^{'}}$; $\frac{\bar{V}_{r_{I(\phi)}}}{Q_I^{'}}$; $\frac{\bar{V}_{z_{I(\phi)}}}{Q_I^{'}}$ the height of the guide vane is leveled.

3. When $n_I^{'}$ changes over a wide range (from values $n_I^{'} = 80$min^{-1} untill $n_I^{'} = 46.5$min^{-1}) the angle change graph $\bar{\alpha}_{(\phi)}$ (between the direction of absolute speed and its peripheral component) the height of the guide vane practically does not change and only at changing a_0 angle $\bar{\alpha}_{(\phi)}$ is changing, increase with increasing a_0.

If we compare the nature of the change in the angle α with the same openings a_0 for the case with an impeller of RO500 $n_I^{'} = 65$min^{-1} and without an impeller, we find that in the absence of an impeller, the angle α is less in 2°÷4°.

Thus, the influence of the lattice of the impeller blades on the flow behind the guide vanes is established. This influence consists in some equalization of the flow unevenness in the circumferential direction and in the height of the guide vanes and in an increase (by 2°÷4°) of the angle.

It was found that uneven flow patterns ($\bar{V}_{u_{I(\phi)}}/Q_I^{'}$; $\frac{\bar{V}_{r_{I(\phi)}}}{Q_I^{'}}$; $\frac{\bar{V}_{z_{I(\phi)}}}{Q_I^{'}}$) is a function of the angle $\bar{\bar{\alpha}}$ flow averaged over the entire cylindrical surface and the dependence is obtained $\bar{\bar{\alpha}}$ from opening the guide vane.

Thus, in advance, when profiling a new impeller, it is possible to take into account the uneven flow along the height of the guide vane.

4. As a result of the analysis of graphs of changes in the angle $\bar{\beta}_{(\phi)}$ flow (between relative and peripheral speed) height of the guide vane for various a_0 and $n_I^{'}$ we can conclude that when changing $n_I^{'}$ ranging from $n_I^{'} = 80$ min^{-1} until $n_I^{'} = 46.5$ min^{-1} angle β varies accordingly from the values 22° at the bottom ring of the guide vane, 20° in the central part and 27° at the upper ring when $n_I^{'} = 80$ min^{-1} until 112° at the lower ring, 133° in the central part and 122° at the upper ring when $n_I^{'} = 46.5$ min^{-1}.

Thus, at turbine operating conditions other than optimal, the flow can flow onto the impeller blade system at angles of attack varying from -50° ÷ -60° до 50° ÷ 60°. In all likelihood, this is one of the main reasons for the sharp change in efficiency when changing $n_I^{'}$ at a_0=*const*. Such unfavorable conditions of flow inflow limit the possibility of installing high-pressure hydraulic turbines at hydro power plants with significant pressure fluctuations, as well as their operation at starting (low) pressures.

At $n_I^{'} = const$ ($n_I^{'} = 65$ min^{-1}) when it changes a_0 (from a_0=9 mm until a_0=29 mm) angle β from 59° ÷ 78° until 43° ÷ 47°. Smaller values β occur at the lower ring of the guide vane, and larger values at the upper.

5. Vortex flow structure determined by $\bar{\Omega}_{r(\phi)}$, $\bar{\Omega}_{u(\phi)}$, $\bar{\Omega}_{z(\phi)}$, does not change significantly when changing the opening of the guide vane ($n_I^{'} = const$) and almost does not change at a_0=*const* and variable $n_I^{'}$.

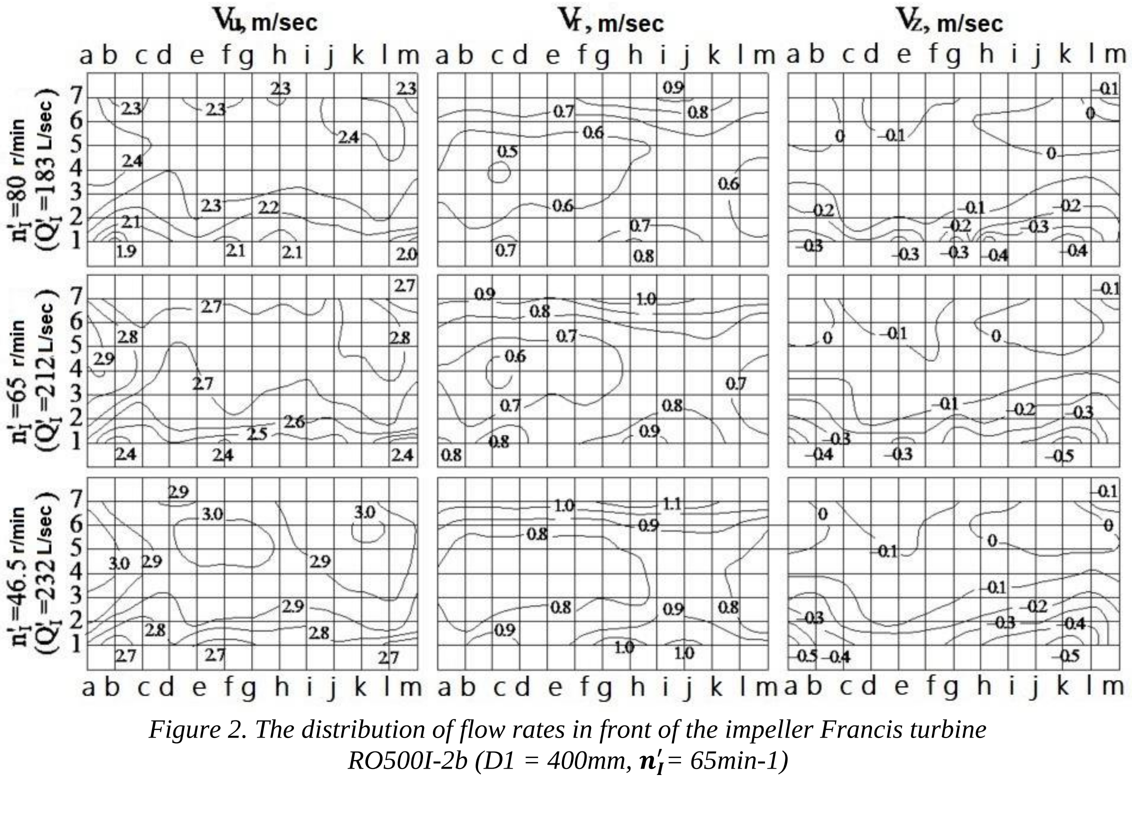


Figure 2. The distribution of flow rates in front of the impeller Francis turbine RO500I-2b (D1 = 400mm, $\boldsymbol{n_I'}$= 65min-1)

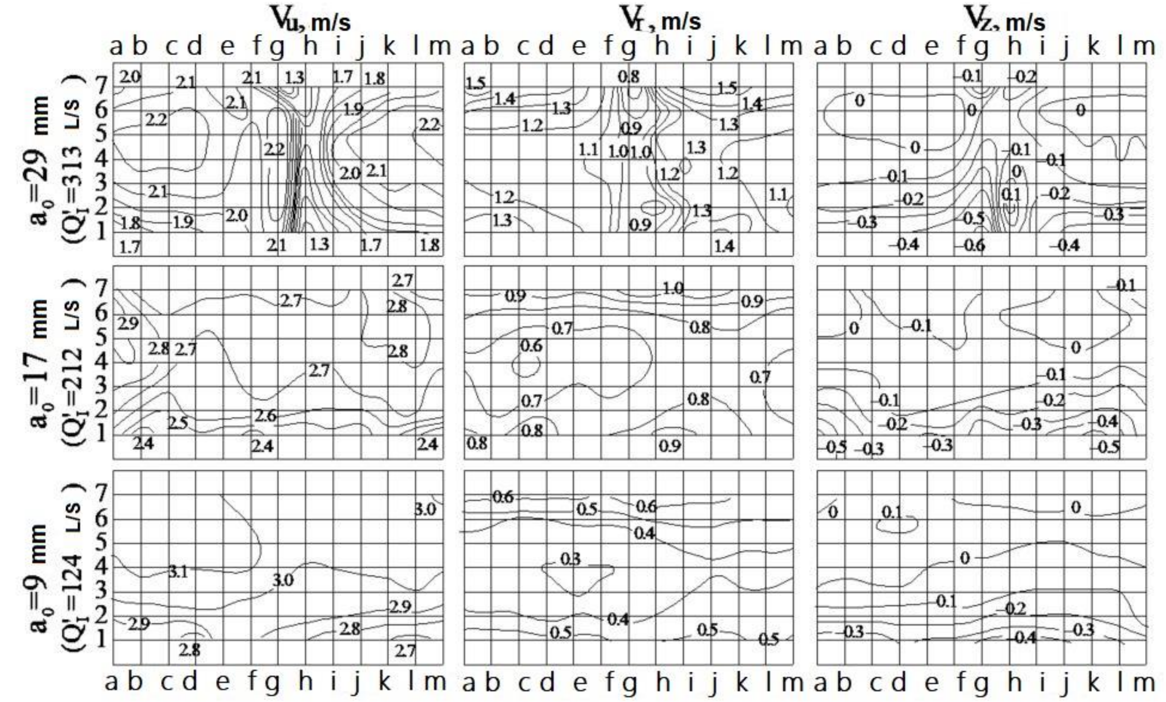


Figure 3. The distribution of flow rates in front of the impeller Francis turbine (Russian standard RO500I-2b) (D1 = 400mm, $\boldsymbol{a_0}$ = 17mm)

As a result of the analysis of the parameters of the stream averaged over the cylindrical surface in front of the impeller, the following regularities were established.

1) Lines $\bar{\bar{\alpha}} = const$ coincide with the lines $a_0=const$ and the difference between $\bar{\bar{\alpha}}$ flow and $\alpha_{н.а.}$ (angle of the midline of the profile of the blade on the output edge), almost equal to zero (0° ÷ 0.5°).

2) In the studied area of the turbine, the angle $\bar{\bar{\beta}}$ varies within 30° ÷ 120°.

3) The energy loss in the spiral chamber and guide vane varies slightly (from 2% to 3.5%) when changing the turbine operating regime from a_0=9 mm to a_0=29 mm and from $n_I' = 46.5$ min^{-1} to $n_I' = 80$min^{-1} and are less important at large values n_I'.

4) For small values n_I' ($n_I' = 46.5 \div 55$min^{-1}) $E_к \approx E_п \approx 50\%$, for large values $n_I'$and Q_I' $E_к \approx 30\%$ and $E_п \approx 70\%$ from full energy.

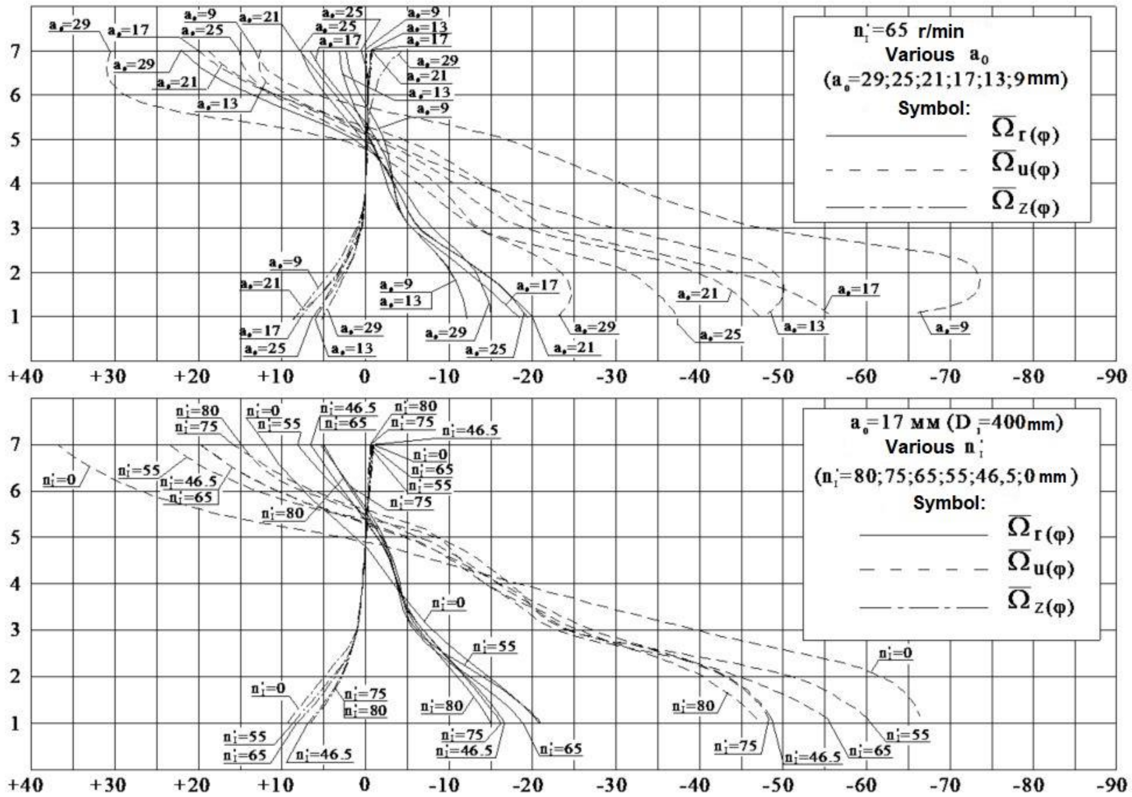


Figure 4. Flow characteristic in section I-I before the impeller Francis turbine (Russian standard RO500I-2b) (D1 = 400mm)

5) Quantities $(\overline{v_u r})_I^{'}$ decrease with increasing $n_I^{'}$and $Q_I^{'}$ from $(\overline{v_u r})_I^{'} = 1.6$ to $(\overline{v_u r})_I^{'} = 1.05$ in the study area of the turbine (9mm ≤ a_0 ≤ 29mm; 46.5 min^{-1} ≤ $n_I^{'}$ ≤ 80 min^{-1}).

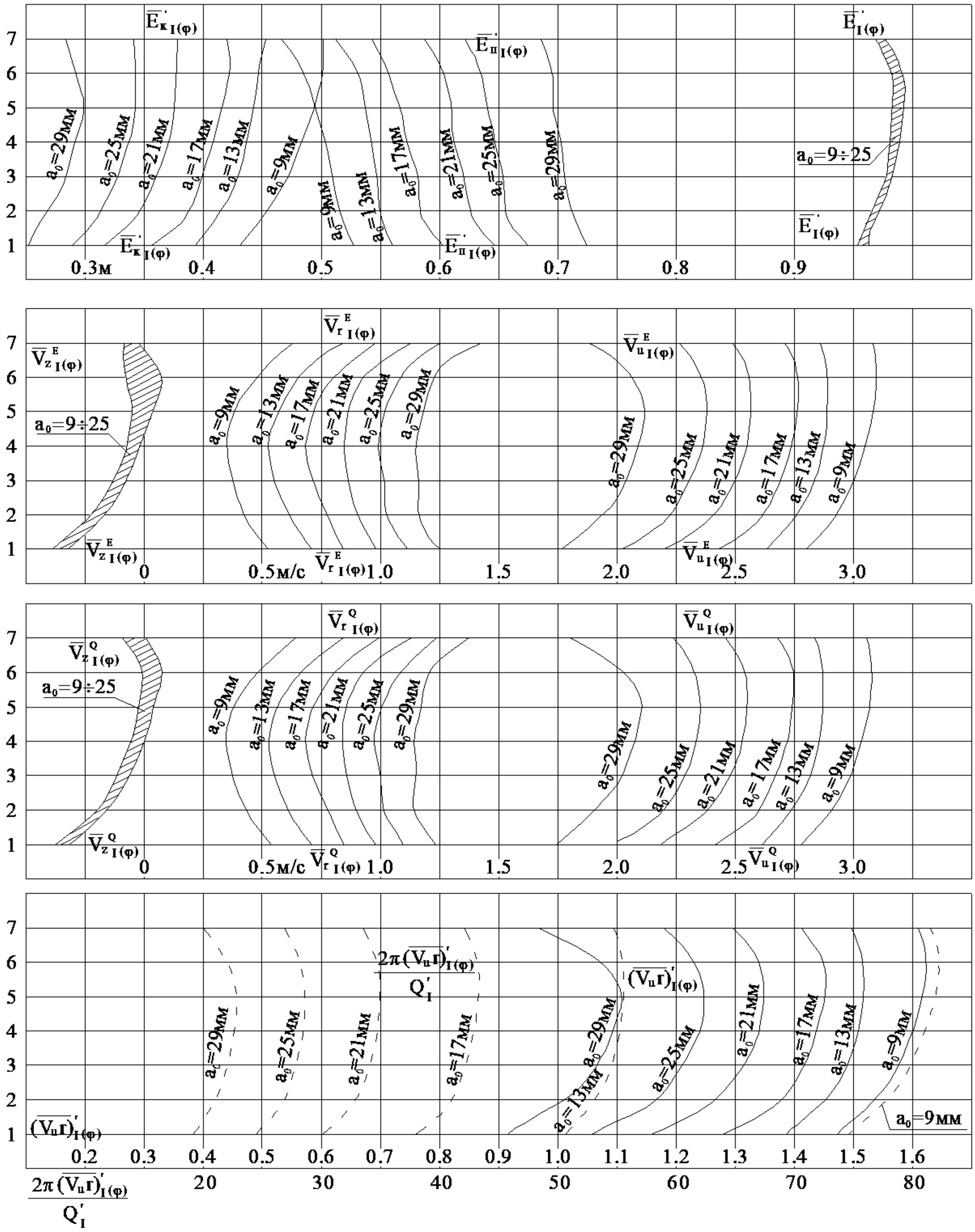


Figure 5. Average flow parameters in front of the impeller Francis turbine (Russian standard RO500I-2b)

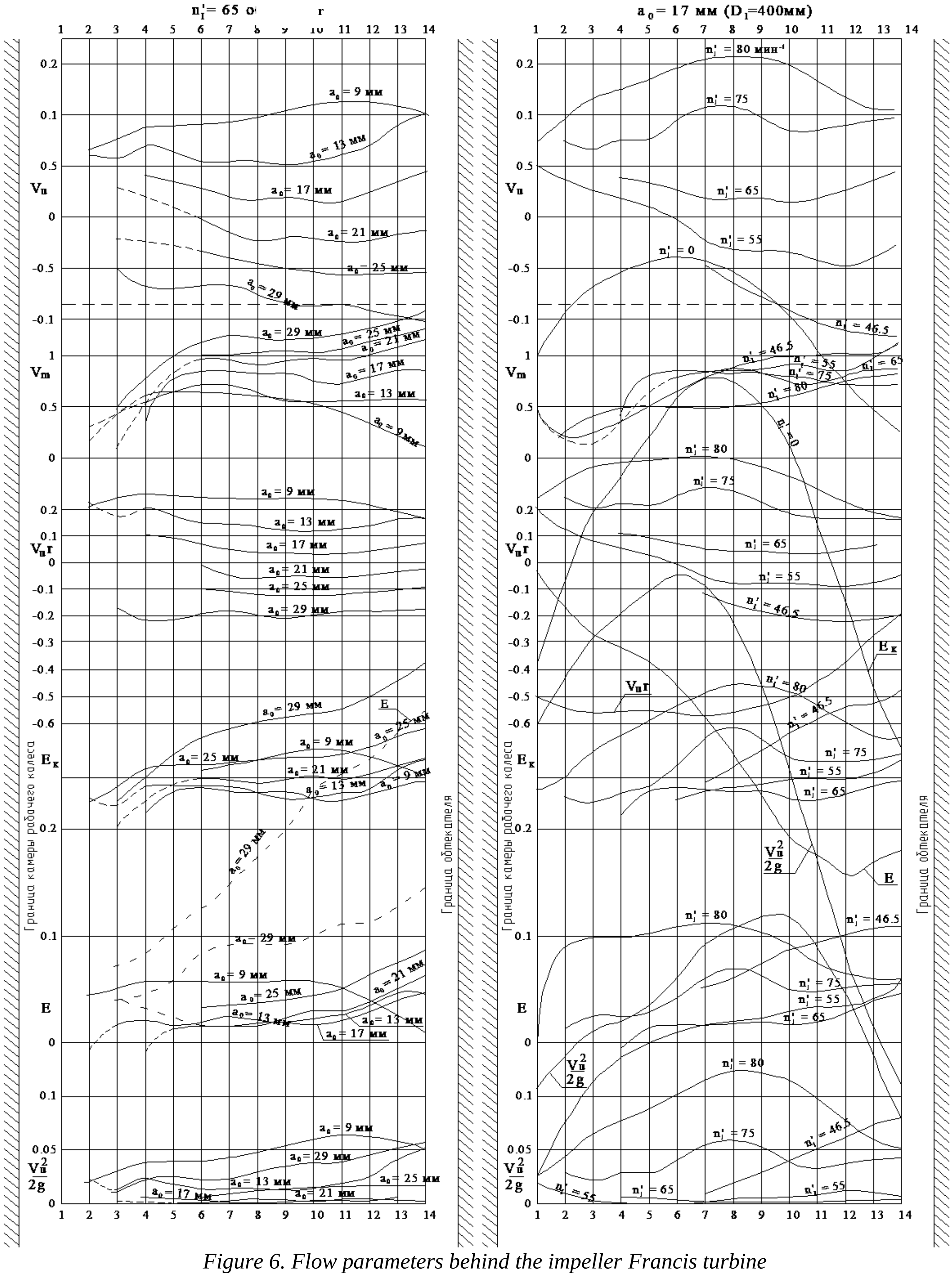


Figure 6. Flow parameters behind the impeller Francis turbine (Russian standard RO500I-2b)

In the optimal regime, there is a minimum value of the kinetic energy of the flow in the suction pipe of the order of 3-4%. At the a_0=9 mm and a_0=25 mm, $n_I^{'} = 65$ min^{-1}, $E_к \approx 7\%$, and at the a_0=29 mm, $n_I^{'} = 65$ min^{-1}, the kinetic energy of the flow reaches 10-12% of the energy of the input streamflow.

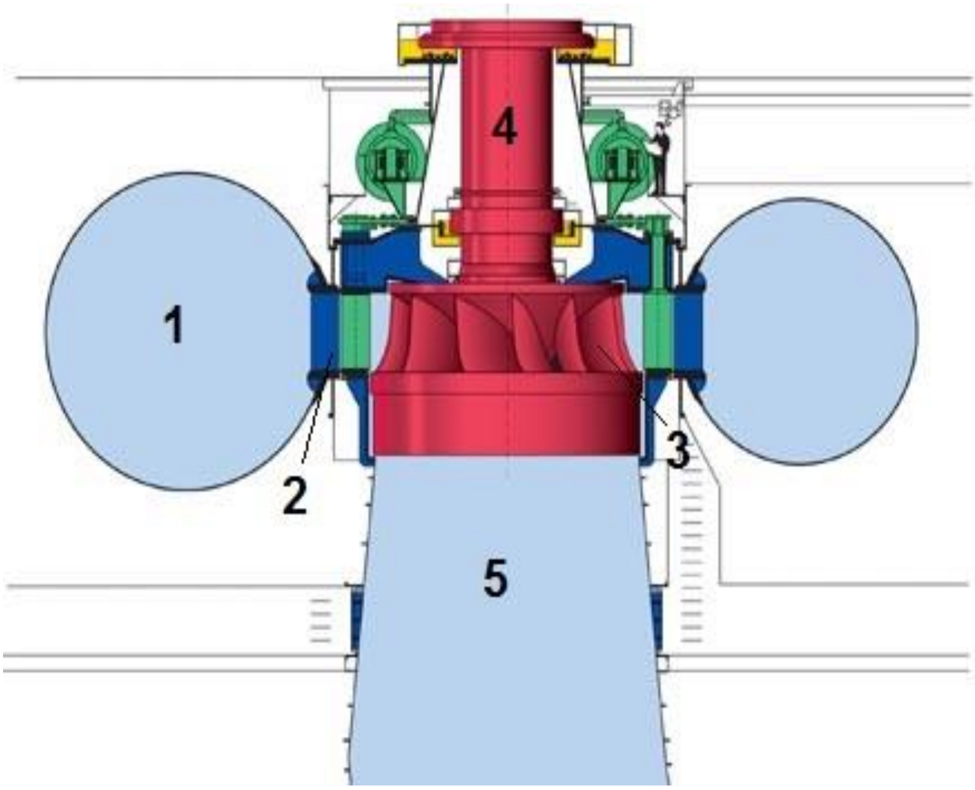
The potential energy of the flow in the suction pipe changes insignificantly when the turbine operating regime changes ($n_I^{'}$and$Q_I^{'}$), as a rule, it has a negative value (pressure is less than atmospheric) and in the optimal regime $E_{n_I^{'}}$ is about 1-2%.

Thus, the energy loss of the flow in the suction pipe at the optimum regime is equal to 1÷2% and increase to 4% at a_0=25 mm and $n_I^{'} = 65$ min^{-1}, and to $\approx$ 8÷9% at the a_0=29 mm, $n_I^{'} = 65$min^{-1}; also to $\approx$ 9÷10% at a_0=17 mm and $n_I^{'} = 80$min^{-1}. Basically, these losses are associated with the energy of the circulation component of the velocity $\frac{(v_{u_I})^2}{2g}$, which at the optimum regime is close to zero.

The optimum universal characteristics of the best impellers are achieved with a positive, close to zero, value of the averaged circulation at the exit of the impeller.

The presented on figures 2-4 graphs of the distribution of flow parameters in a cylindrical section R=212 mm (for D_1=400 mm) impeller Francis turbine RO500I-2b (modification PO400I-2b – stock number 3515) can be used in the development of new blade systems for pressure 400÷500 m, as well as for mathematical modeling of the flow as boundary conditions.

It should be noted that the use of modern three-dimensional methods of mathematical modeling of the flow, based on rather complex and rigorous turbulence models, requires the same high degree of accuracy of the task, as the averaged flow parameters, and the characteristics of the flowing turbulence as boundary conditions. in the input section.



1 – spiral chamber; 2 – guiding vanes; 3 – impeller; 4 – shaft; 5 – draft tube
Figure 7. General view of a section along a radial-axial hydraulic turbine

Conclusions

This article was concluded by the process of developing methods for mathematical modeling of turbulent flows. Firstly, diffusion and convective transport as the transfer of mass, energy, momentum and angular momentum. In the process of mathematical modeling is this complex transfer process. However, it is known that the Navier-Stokes equation is a differential analog of the momentum conservation law, and the continuity equation is a differential analog of the mass conservation law. The law of conservation and transfer of angular momentum are not directly applied in mathematical modeling. Also in the optimal regime, the minimum kinetic energy of the flow in the draft tube and the flow behind the impeller in the spiral chamber and the guide vanes changes slightly (from 2% to 3.5%) with a change in the turbine operating regime.

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SYSTEMATIC RESEARCH FOR ENHANCEMENT OF COMPETITIVENESS OF SHIPBUILDERS COLLEGE IN THE SPACE OF PROFESSIONAL PRE-HIGHER EDUCATION.

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СИСТЕМНЕ ДОСЛІДЖЕННЯ ПІДВИЩЕННЯ КОНКУРЕНТОСПРОМОЖНОСТІ КОЛЕДЖУ КОРАБЕЛІВ У ПРОСТОРІ ФАХОВОЇ ПЕРЕДВИЩОЇ ОСВІТИ

Summary. This work is dedicated to solving an important scientific and practical task of analytical forecasting of processes of professional pre-higher education development. For this matter a system-analytical research has been made. This research includes formulation of criteria for improving efficiency of innovational and research activities of Information technology department of Shipbuilders Collage of National university of shipbuilding and building cause and effect diagrams for the matter of situation forecast.

Анотація. Робота присвячена вирішенню важливої науково-прикладної задачі аналітичного прогнозування процесів розвитку фахової передвищої освіти, для чого виконано системно-аналітичне дослідження, яке включатиме у себе формулювання критеріїв підвищення ефективності інноваційної та дослідницької діяльності відділення інформаційних технологій Коледжу корабелів Національного університету кораблебудування імені адмірала Макарова (м. Миколаїв) з побудовою для прогнозу ситуації причинно-наслідкових діаграм.