Analysis and improvement of calculation procedure of high-speed centrifugal pumps

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Abstract
The model of flow around a flowing part of a high-speed centrifugal pump with a semi-open impeller is presented. The calculated ratios for design of low-flow pumps are obtained and confirmed experimentally.

1. Introduction
The development of aerospace technology, expansion of the range of problems solved with the help of power plants of aircraft put forward for researchers the problem of increase of efficiency of supply pumping systems with low consumption, their design excellence and forecasting energy parameters with a given level of stability. The solution to this problem contributes to the quality of design, speeding up processing and commissioning of modern power plants of aircraft [1].

High-speed centrifugal pumps (HSCP) at an angular speed of the rotor up to 10 000 rad / s is widely used as a part of a turbopump assembly of liquid rocket thrusters and power plants of aircrafts, which is caused by a wide range of changes in their regime parameters. For example, when the angular velocity is from 3000 to 10000 rad / s, the value \( \frac{\dot{V}}{\omega} \) reaches at least \( 10^{-7} \) m³ subject to \( \text{Re}_\omega > 10^7 \). Reduction in the supply \( \dot{V} \) in such pumps, along with an increase in the angular velocity of the rotor, usually results in a decrease \( \frac{\dot{V}}{\omega} \) less than the value \( \left( \frac{\dot{V}}{\omega} \right)_{\text{max}} = 1 \times 10^{-6} \) m³, which is the maximum permissible for centrifugal pumps with closed impeller [1]. Therefore, HSCP with an open and semi-open impeller are widely used.

2. Visualization of the flow pattern in an impeller
For different ratios of width and height of the channel \( b/h \), it has been presented visualization pictures of flow over a rectangular cavity [2]. By decreasing the width of cavity under primary vortex, the secondary one starts to grow. When the ratio of width to height is tend to zero, an infinite sequence of vortices is formed, each vortex is weaker than its predecessor. In order to clarify the main features of hydrodynamics in semi-open channels of impellers, series of experiments are carried out, including experiments on flow visualization in such impellers.

Tests over a wide range of regime pump parameters (\( \dot{V} = 0...1.5\dot{V}_{\text{nom}} \)) and change of the axial clearance from 0.5 to 3 mm revealed that for different radii the constancy of the ratio of the radial component of velocity \( \nu_r \) to the circumferential velocity is hold: \( \frac{\nu_r}{U} = \varphi_r \). This
gives grounds to express the speed of radial flow in an axial gap from the periphery of the impeller to the circumferential direction with respect to the impeller in the form

$$\omega_R = U/R = UK\phi_\lambda,$$

where $K_R$ is the experimental constant, $K_R = 0.45$.

Consequently, as at the entrance of the impeller with semi-open blades and on its radius, not all of the flow channels of the impeller is filled with the stream moving from the entrance of the impeller to the exit. Partially its flowing part is filled by backflow. Swirling in the direction of rotation of the impeller, reverse currents flow into the adjacent channel to a smaller radius and are addicted to the back of the flow (figure 1). This creates a vortex zone in which the liquid is not involved in the supply flow through the pump. With increasing supply through the pump with $\omega = const$ parameters of the vortex zone are reduced, resulting a decrease in the flow of the radial twist.

![Figure 1](image)

**Figure 1.** Scheme of the turbulent flow of the stream in the channel under the flow around the blade

Based on the studies on stream visualization [3] and measuring the hydrodynamic parameters of flow in the gap between the rotating impeller with open ends of the blades and smooth hull, a model of the jet-vortex flow around channels of an impeller can be represented (figure 2). The liquid flow in the inter-blade channel of the impeller is exposed to direct force influence of the blades. The liquid in the axial gap $a_1$ is twisted due to friction forces and slips relative to ends of the blades of the impeller. Thus, on the same radius the fluid particles
in the channel and the axial gap \( a_i \) move with different circumferential speed, which leads to their relative movement in the radial and axial directions.

**Figure 2.** Scheme of interaction of the incoming stream of axial clearance and the stream in the pump impeller channels:

1 – the zone of recirculating flow in the impeller channel;
2 – the zone of incoming flow of axial clearance;
3 – the borders of the mixing zone.

In the circumferential direction of fluid flow in the gap \( a_i \) relative to the blades causes the formation circulation zone in channels of the impeller as a result of the stream flows around cavities. It is known that the fluid flow in the channel behind a poorly streamlined body (in our case these are the blades of HSCP) having separated flows, which are characterized by the formation of reverse currents and vortices. This process is largely determined by the ratio of blade width \( b \) to the distance between them \( d = t - \delta \), the boundary layer thickness on the wall in front of the channel and a relative height of the channel. The interaction between the jet and the fluid in the channels of the impeller leads to the appearance of the circulation flow in the channel behind the blade. In the cross section under \( y = 0 \) the stream due to its compression in the axial gap \( a_i \) is uniform, and as a result of automodelity of turbulent jets [4], the parameters of the circulation zone do not depend on the number \( Re \). Then for the mixing zone 2 the value of relative velocity of the flow is expressed as:
3. The boundaries for the different zones of flow

Using the basic regularities for free flat jets [4], based on the known experiments on strain of any irregular profile to a jet one at a very small distance from the blade edge, we obtain bounds for the different flow zones under the flow in an interblade channel of finite length.

For symmetric about the axis X boundary of the mixing zone 2 dividing the flow in the channel from the stream in the gap $a_i$, its position can be determined by the expression [4]

$$y = \pm 0.08829x.$$ (3)

Comparing the obtained pressure distributions based on visualization of flows in the fixed cavity and in rotary channels of the semi-open impeller gives grounds to assume that in the channels there is an intense vortex flow with the axis of vortex directed along the radius of the impeller and with displacement of the vortex to the pressure side of the blade. The circulation area in the channel of the semi-open impeller (figure 3) let us roughly divide into two areas: the area of intensive rotational motion along a circle with radius $r_v$ and the area of circulation along the trajectories of the elliptical nature. The size of vortex zone radius on base of equation (3) equal to

$$r_v = 0.519(b - 0.08829d).$$ (4)

One of the characteristics of the vortex zone is the ratio of the circumferential velocity and the velocity of the incident flow $W_u$ at the external boundary of the vortex zone

$$\varpi_v = \frac{W_v}{W_u} = f\left(\frac{b}{a}\right).$$

Analysis of special experiments [5, 6, 7] on determining the estimated value $\varpi_v$ gives the basis to take it for our investigations equal to 0.3. The same value $\varpi_v$ is shown in a number of works for the fixed channel. Then the expression for the angular speed of rotation of the vortex in channels of the impeller on the radius $R$, with taking into account the value $\varpi_v = 0.3$ and the dependence (2), can be written as

$$\omega_v = \frac{\varpi_v(1 - \varpi_i)\omega R}{r_v} = \frac{\varpi_v(1 - \varpi_i)\omega R}{0.519(b - 0.08829d)}.$$ (5)

Experimental data generally confirm the adopted model for calculating channels of the semi-open impeller in the form of three-zone jet-vortex flow, which allows to calculate its geometric parameters.

The parameters of the mixing and vortex flow zone have decisive influence on the sizes of the impeller channel. On the basis of the experiments of works [2, 3, 4], it should be noted that for flow around the square channel $b/d = 1$ there is one stable vortex rotating almost like a solid body. With further increase in the channel depth to $b/d = 2$, there are two vortex cells observed that arranged one above the other and having opposite directions of rotation. In
order to minimize hydraulic losses in the impeller channels, a multivortex flow is unacceptable both in depth and in the width of the channel. Proceeding from this condition and assuming that the average velocity at the end of the section \( d_1 \) for the forward and reverse flows of circulating zone are equal (figure 2), according to the decision G.N. Abramovich [4], for the area in which energy of the reverse single-vortex flow reaches a maximum, we get

\[
d_1 = (4.75 \pm 5.2)b. \tag{6}
\]

Similarly, we can get the size of the second section with length \( d_2 \) from the terms of the approximate equality of the averaged values of the energy in the forward and reverse flows in cross section \( x = d_1 \):

\[
d_2 = (0.980 \pm 0.852)b. \tag{7}
\]

Thus, the total length of the deep channel for single-vortex flow is

\[
d = d_1 + d_2 = (6.18 \div 6.05)b. \tag{8}
\]

The flow in the shallow channel is characterized by an elongated boundary of the mixing zone and its accession to the surface of the channel at the point \( x = d_1 \). Taking into account the relation (3) we have: \( d_1 = 11.36b \). Then, taking into account that the vortex flow on the pressure side of the blade is formed by the jet overflowing to backside, we find the greatest value of the channel values:

\[
d_{\text{max}} = (11.36 + 1.14)b = 12.5b. \tag{9}
\]

From (4) the minimum possible value of the channel is

\[
d_{\text{min}} = 1.1423b. \tag{10}
\]

The minimum width blade at the outlet of the impeller can be finding from the condition of conservation of the stream core in the inter-blade channel that will at the same time the condition of applicability the theory to the calculation of impeller channels [4].

The performed experimental research, analysis and visualization of streamlines data obtained fully confirmed the calculated ratio of the sizes of interblade impeller channels that are expressed in the form of dependencies for the entry parameters on the blade with thickness \( \delta_1 \) and the channel angle \( \beta_1 \):

\[
\left( \frac{\pi D_1}{z} - \frac{\delta_1}{\sin \beta_1} \right) \geq K_1 \delta_1, \tag{11}
\]

where \( K_1 \) is the coefficient of impeller channel width at the entrance, \( K_1 = 1.14 \).

For the output from the impeller we similarly write
\[
\left( \frac{\pi D_2}{z} - \frac{\delta_{2\Lambda}}{\sin \beta_{2\Lambda}} \right) \leq K_2 b_2,
\]

where \( K_2 \) is the coefficient of impeller channel width at the output, \( K_2 = 6.18...12.5 \).

Value for impeller channel width at the output taking into account density \( \tau_2 \) of the profiles lattice is

\[
b_2 = \frac{(1-D_1) D_2}{K_2 \tau_2 \sin \left( \frac{\beta_{1\Lambda} + \beta_{2\Lambda}}{2} \right)} - \frac{\delta_{2\Lambda}}{K_2 \sin \beta_{2\Lambda}}
\]

Similarly, with regard to (12) for the impeller parameters at the density lattice \( \tau_1 \) we obtain:

\[
b_1 = \frac{(1-D_1) D_1}{K_1 \tau_1 \sin \left( \frac{\beta_{1\Lambda} + \beta_{2\Lambda}}{2} \right)} - \frac{\delta_{1\Lambda}}{K_1 \sin \beta_{1\Lambda}}
\]

Dependencies for two values of channel width coefficient \( K_2 =12.5 \) and 6.18 calculated by the expression (13), in which limits all the experimental points of HSCP with the half-open impeller lie, are shown in figure 3. It should be noted that in the finishing process the number of HSCP to achieve acceptable parameters on the pressure and the efficiency, it was sufficient to increase the blade width to the value approaching the upper curve ( \( K_2 = 6.18 \)).

![Figure 3](image_url)

**Figure 3.** The calculated dependences on the change in blade width of the semi-open impeller ▲ - KBHM ○ - KB «Yuzhnoe»

4. Improving the energy transfer processes
One of the real ways to increase the specific power of the pump consists in improving the processes of energy transfer to fluid flow in the flowing part of the pump, which is realized by setting in its cavities the equalization elements in the form of perforated lattices and grids. It should be noted that the equalization elements are widely used in the stabilization of flows in immobile channels. The equalization elements are detailed study on their interactions on the flow in the part of alignment of velocity field and pressure and eliminating fluctuations in the pipelines.

For pumps with a specific supply \( \dot{V}/\omega = (1...5) \cdot 10^{-7} \text{ m}^3 \), the optimal parameters of the grid with the flow cross section coefficient \( f = 0.4...0.56 \) are calculated on the present procedure. The results of comparative tests of pumps with the impellers without the grid and with the grid \( (f = 0.5) \) on the output are shown in figure 4. Due to the alignment of the stream structure the pump pressure is increased by 15 ... 20%. At high speed pumps, nonuniformity is higher and such construction provides increased pressure up to 30%. The presence of the grid at the entrance of the impeller leads to improved cavitation characteristic of the pump. Installation of equalization elements reduces unevenness of fluid velocity field behind the impeller, reduces deviation of the fluid flow from the direction of the blades at the outlet of the impeller and pressure pulsations.

![Figure 4. Energy characteristics of the pump:](image)

1 – the impeller with grid at the output \( (f = 0.5); \)
2 – the impeller without grid

5. Conclusion

Further experimental study of HSCP with the semi-open impeller in a wide range of constructive and regime parameters, as well as analysis of carried out constructions of HSCP received as a result of their repeated improvements (over twenty sizes) have shown the acceptability of dependencies (12) and (13) taking into account lattice density for calculation basic structural relations of HSCP with the impeller of the semi-open type.

In order to generalize the studies carried out, recommendations have been developed for calculating the energy parameters of the HSCP with impellers of semi-open types.
Experimental data of the authors and other researchers confirmed the main theoretical positions. The results of the studies are presented in the form of generalized dependencies, which make it possible to widely use them in engineering practice.

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