Calculation and optimization of parameters in low-flow pumps

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Abstract. The materials on balance tests of high-speed centrifugal pumps with low flow rate are presented. On the bases of analysis and research synthesis, we demonstrate the rational use of impellers of semi-open and open types providing high values for energy parameters of feed system of low-flow pumps.

Currently, work is continuing on the improvement of pumps for various purposes, especially centrifugal pumps with a semi-open impeller, where impeller type is selected mainly for constructional reasons and only sometimes for technology. Impellers of high-speed centrifugal pumps (HSCP) have small size and are well assembled with a high-speed small-sized axis gas turbine. This predetermines their widespread use in energy installations, primarily in turbo pump assemblies of liquid fuel rocket engines for systems of orientation, stabilization and brake propulsion.

Because the theoretical methods for calculating the jet-vortex flow around lattices of half-open impellers are developed for the rational design impellers for exactly such centrifugal pumps, so when we select the best constructive solutions for an entire pump, paramount importance until now belongs to experiments that allows to specify the basic theoretical principles.

HSCP with small flow rate \(\dot{V}\) at the angular velocity of the rotor \(\omega\) up to 10 000 rad/s are widely used as a part of turbopump assembly of liquid fuel rocket engines with low thrust and in energy installations of a spacecraft. This is due to a wide range of changes in pump mode parameters. For example, when the angular velocity is from 3000 to 10000 rad/s, the value \(\dot{V}/\omega\) is usually less than \((\dot{V}/\omega) = 1 \cdot 10^{-6}\) m\(^3\) which is the maximum permissible for centrifugal pumps with a closed impeller [1].

However, in the design of low-flow centrifugal pumps (LFCP) for aerospace applications, techniques and principles for optimization of flowing part are commonly used that are recommended for full-size centrifugal pumps, usually with a closed-type impeller [2]. This significantly limits mode parameters of pump units.

Hydrodynamic analysis [3] of viscous fluid flow around a lattice of LFCP profiles shows a significant displacement of the flow core from the channel walls and narrowing of the flow cross section in interscapular channels up to 80%. This leads to sharp decline in energy parameters of LFCP down to disc pump parameters [1]. Such blade pumps have a low value of the specific speed ratio:
\[ n_s = 193.3 \frac{\omega \sqrt{\dot{V}}}{H^{0.75}} \]  

where \( \omega \) is angular velocity of the rotor, rad/s; 
\( \dot{V} \) is volume flow rate of the working fluid, m\(^3\)/s; 
\( H \) is pressure of the working fluid, J/kg.

The value \( n_s < 50 \) in pumps with low flow rate levels (50 ... 300 cm\(^3\)/s) is mainly determined by the angular velocity and pump pressure. Such blade pumps, compared to pumps with high values \( n_s \), have an increased proportion of hydraulic, mechanical and consumable losses. The proportion of these losses in the total energy balance is great and can be up to 70% of power consumption [1].

When designing flow part of a centrifugal impeller with \( n_s < 50 \), small width can be obtained at the output of impeller (sometimes less than 1 mm), which is technologically difficult to implement and often designer increases the channel width of the impeller considering the boundary layer thickness. Such an increase in the channel width leads to a high degree of diffusivity \( \overline{W} \) and a significant reduction in the energy parameters of the pump [4]. With increasing of viscosity, addition to reducing efficiency and impeller pressure, there is displacement of the greatest value of the efficiency to the region of lower flow rate. Occurs as though locking effective section of the impeller channel due to the growth of the boundary layer thickness, which is consistent with the calculation model presented in [3; 4].

In [5] on the basis of generalization of data presented in the works on HSCP with low flow rate in various organizations of aerospace industry, authors noted the need for a detailed study of HSCP with semi-open impellers as a promising high-speed pump at a value \( n_s < 50 \) and \( \dot{V}/\omega < 5 \cdot 10^{-7} \) m\(^3\). In improving the flow part of HSCP with low flow and optimizing constructive solutions, great importance is experiments with carrying out the balance tests and analysis of the results.

The famous technique of S.S. Rudnev for balance testing large sized centrifugal pumps have allowed with some interpretations in a number of studies to realize the carrying out of balance tests with HSCP with impellers of closed, semi-open and open type according to the patent [6].

Analysis of the energy balance of HSCP with semi-open and open impellers has allowed to reveal features of the balance of energy loss in a pump with allocation the components of certain types of losses. Along with the balance tests, the energy characteristics of a centrifugal pump were taken with change in a wide range of constructive and mode parameters that are of practical and scientific interest. For example, the end clearance between the blades and the pump body were varied in different combinations of impellers from 0.25 mm to 3 mm. For purity of experiments, testing HSCP with different types of impellers were carried out with the simplest sealing system without the use of impellers, recirculation systems and so on, the value of the flow rate was discrete changing from 0 to 1.5 \( \dot{V}_a \) through each 30 ... 50 ml/s. A nominal value of flow rate was equal to a value corresponding to the largest total efficiency of the pump.
Comparative tests and assessment of the balance of energy losses in HSCP with impellers of different types showed bias of mode of the most hydraulic pump efficiency to the region of lower flow rate. It is thus noticed the redistribution of certain types of energy loss. Thus, along with decreasing mechanical efficiency $\eta_m^k$ of an impeller, consumable efficiency $\eta_p$ was increased. For a semi-open impeller, that is identical to closed impeller by flowing parts, all this generally led to change of the efficiency of the pump from 0.535 to 0.525. When $\omega = 1047 \text{/s}$ for the mode with the greatest efficiency of the pump with semi-open impeller, efficiency components were as: full $\eta_a = 0.519$; hydraulic $\eta_g = 0.699$; consumable $\eta_p = 0.924$; mechanical impeller $\eta_m^k = 0.828$; mechanical parts of the stator $\eta_m^s = 0.969$.

The absence of coating disc and front gap seals significantly reduced leakage rate. At the same time mechanical losses of the impeller increased due to friction of end bladed surface. The hydraulic efficiency slightly decreased down to 0.699 due to energy loss in friction of fluid flowing in the channels of the half-open impeller. It should be noted that the value of the complex $(K, \eta_g)$ for a HSCP with the half-open and open impeller at $\beta_z$ near to $90^\circ$ remains constant with a change in the flow rate.

Hydraulic characteristics of a centrifugal pump with an open and semi-open impeller have the best value in the region of low flow rate. The hydraulic efficiency of an open centrifugal pump is 0.75 - 0.78 in the entire range of low feed

$$\frac{\dot{V}}{\omega} \leq 1.5 \cdot 10^{-7} \text{ m}^3,$$

with a substantial increasing consumable efficiency up to 0.997 (there are practically no leakage). However, due to friction on the end surfaces of blades of the impeller, there are observed a significant loss, the value of which has increased up to the value $\eta_m^k = 0.7$ because of removal the part of the second disk. Herewith, the average level of energy parameters, which are similar for HSCP with closed impellers, at

$$\frac{\dot{V}}{\omega} \leq 1 \cdot 10^{-7} \text{ m}^3$$

was not only reached, but also was surpassed.

Losses in the drainage devices was determined integrally over the entire channel of drain in HSCP as the amount of loss in the spiral collector $L_s$ and the conical diffuser $L_{c,d}$. When testing various types of impellers, the hydraulic efficiency of drainage devices in centrifugal pumps had only little dependence on the type of the impeller and in a greater extent was determined by the operating mode of the pump. And this efficiency was decreases with increasing flow rate [1]. In the analysis of the energy characteristics of drainage in HSCP, the methodology of B.I. Borovsky [7] is accepted and the coefficient of drain losses is calculated.
The value of this coefficient for a wide range of pump flow rate \( \dot{V} \) equals to \( \xi_{\alpha,p} \) = 0.2 ... 0.3, which is consistent with the data for high-flow pumps [7] in the region of low feed and exceeds their values at nominal and high flow rate. Herewith, \( \xi_{\alpha} = 0.25 ... 0.3 \) for calculated mode, and that was accepted for further study of LFCP. It should be noted that for all tested impellers the minimum for coefficient of drain loss coincides with the maximum for hydraulic efficiency.

In pump theory, a complex issue is determination of the angle of flow deflection from the direction of the blades at the output of the impeller \( \delta = \beta_{2t} - \beta_2 \) and the associated rate of influence of a finite number of blades \( K_z \) [7].

\[
K_z = \frac{H_T}{H_{Fp}}
\]  

(2)

Directly on the characteristics of the pumps tested, the product

\[
K_z \eta_p = \frac{H}{H_{Fp}}
\]  

(3)

can be determined. And conversely, it is easily to determine pump pressure from the known value of product \( K_z \eta_p \).

The results of generalization of known data on the value \( K_z \eta_p \) for a centrifugal pump with various types of impellers allowed to accept the value \( K_z \eta_p = 0.62...0.68 \). Herewith the materials of work [7] are used, that summarizes the tests of high-flow pumps with a closed impeller \( (\beta_{2t} < 90^\circ, n_s = 60...180; z = 8...12) \) under changing

\[
\overline{D}_1 = \frac{D_1}{D_2} = 0.8...0.3.
\]

And the tests of pumps with \( n_s < 50 \), including HSCP with low flow rate in systems of thermal control in a spacecraft [2]. It should be noted the identity of the recommendations on the results of all work about the values \( K_z \eta_p \) which is constant at \( \overline{D}_1 = 0.3...0.55 \).

A large number of semi-empirical methods and dependencies are known, that allow to estimate the value \( K_z \) for a given class of pumps. However, they consider only closed impellers and are not applicable for calculation of pumps with semi-open impellers. Balance testing of
centrifugal pumps with semi-open and open impellers allows to analyze the impact of design and operational factors individually on the value $K_z$ and the hydraulic efficiency of the pump.

With a decrease in $\bar{D}_l < 0.2$, the complex $K_z \eta_g$ decreases only for HSCP with a closed impeller due to the increase in hydraulic losses at flow in the channels [7]. It should be noted that with the growth of the working fluid viscosity, this value comes at a larger value $\bar{D}_l$ [3]. That is determined by increasing the thickness of the boundary layer displacement at flow in narrow channels of a closed impeller.

From the foregoing it follows that the rational application of HSCP with a half-open impeller is acceptable at $\bar{D}_l = 0.1 ... 0.55$. It should be considered dependence ($K_z \eta_g$) from the angle $\beta_{zl}$. For an impeller with $\beta_{zl}$ increased up to 90°, a value ($K_z \eta_g$) is high.

The peculiarity of the working process in channels of semi-open and open impellers at vortex interaction with the blades of radial fluid flow causes a stable value $K_z \eta_g$ over a wide range of variation of angular speed of the rotor of the pump (Figure 1). For impellers of semi-open and open types in the whole region of change the value $\bar{D}_l < 0.55$, there is a consistently high value $K_z \eta_g = 0.62 ... 0.68$. That determines preferential using these impellers in HSCP at $\bar{D}_l < 0.25$. And it is typical for high-pressure pumps in power installations of a spacecraft.

![Figure 1](image)

**Figure 1.** Influence of the angular velocity on the change of the complex $K_z \eta_g$ for impellers of semi-open and open types:

- tests obtained in SibSAU
- tests obtained in other researches
Experimental data for open and semi-open types of an impeller are approximated at \( \beta_{2i} = 90^\circ \) by dependence of the form:

\[
K_z = 0.2 + 0.7(1 - \bar{D}_1^{12})
\]  

(4)

or, taking into account the value \( \beta_{2i} \), the dependence has been obtained:

\[
K_z = 0.2 + 0.67\beta_{2i}^{0.15}\left[1 - \left(\frac{\bar{D}_1}{\sin\beta_{2i}}\right)^{12}\right]
\]

(5)

where \( \beta_{2i} \) is tilt angle of the blade at the outlet, in radians.

The impact of the relative diameter \( \bar{D}_1 \) on the value \( K_z \) is consistent with similar data for closed impellers [7] and occurs at low density lattices. At the usual number of blades \( z = 4 \ldots 8 \) the impact of the relative diameter of an impeller occurs in area \( \bar{D}_1 > 0.6 \ldots 0.7 \). Increasing the value \( K_z \) can be achieved by increasing the number of blades \( z \). However, it clutters through passage section of the impeller, and this leading to increased hydraulic loss. The optimum number of blades providing the largest value \( K_z \) can be found from the dependence [5; 8]

\[
z = \frac{\pi \tau \sin\left(\frac{\beta_{1i} + \beta_{2i}}{2}\right)(1 + \bar{D}_1)}{\left(1 - \bar{D}_1\right)},
\]

(6)

where \( \tau \) is density of the lattice of profiles.

The analysis of the balance tests for centrifugal pumps with semi-open impellers confirmed the admissibility of dependence (5) in calculation of the value \( K_z \) for the optimized (on density) circular lattice of profiles at the flow part in the channels of the impeller and also in calculation of its design features. It should be noted that the results obtained from the dependence of the coefficient \( K_z \) on the angle \( \beta_{2i} \), the values \( \bar{D}_1, \tau \) and \( z \) are acceptable for operation mode of HSCP at

\[
C_{2m}/U_z \leq 0.1.
\]

That corresponds to the parameter

\[
\dot{V}/\omega \leq 5 \cdot 10^{-7} \text{ m}^3.
\]
The analysis of the balance tests for HSCP with half-open and open impellers has shown the value \( \eta_g = 0.8...0.85 \). That meets the modern requirements for energy excellence of pumps (Figure 2). For comparison, there the experimental data are presented for HSCP with closed impellers. These data are described by the dependence taken from the work [1]

\[
\eta_g = 0.38 \cdot n_s^{0.21}.
\]  

(Figure 2) Dependence of the hydraulic efficiency for HSCP with various types of impellers on speed coefficient: 

- \( \text{̂} \) semi-open and open impeller; \( \bigcirc \) closed impeller.

Thus, at the low value \( n_s \), and therefore, at the value \( D_1 < 0.25 \), in order to obtain a high pump efficiency, it is recommended to apply an impeller of semi-open or open types because its power characteristics exceed those of pumps with closed impellers.

Special attention in the study of HSCP with the semi-open and open impellers was given to assessing the impact of the value \( a_i \) of a lateral gap between the impeller and the pump body on change the energy characteristics of the pump to optimize sizes of gaps, providing not only the highest settings, but also processability of HSCP.

The tests with analysis of the impact were conducted on a number of pumps with impellers with different geometry at various angular rotor speeds from 500 to 4400 rad/s. The value \( a_i \) of the end gap changed from 0.15 to 3 mm on both sides simultaneously, and also at a fixed value of one. Furthermore, we tested impellers with variable width of the blade at the outlet of the impeller for the series of constant gap values \( (b_2 = 2.7...9.5 \text{ mm}) \). That has allowed to experimentally verify the condition to minimize blade width at the output of the impeller according to the theoretical relation in the work [8] from the condition of conservation of the flow core in the channels of the impeller by the oncoming jets from the end edges of the blades.
Dispersion of the experimental points for dependence (8) is $\sigma = 0.023$ and for dependence (9) is $\sigma = 0.00455$, which is quite satisfactory. In the result of testing the limit values $a_i/b_2$ are verified. HSCP with flowing part parameters, which do not corresponding to the received ratios, are characterized by low energy parameters.

We consider the rational use of centrifugal pumps. It is shown that the centrifugal high-speed pumps with closed impellers at $\overline{D}_1 < 0.25$ have low efficiency due to the high proportion of hydraulic losses. That allows us to recommend their use in regimes

$$\dot{V}/\omega \geq 1.5 \cdot 10^{-7} \text{ m}^3.$$

Experimentally we studied how the main design parameters of pumps with semi-open and open impellers effect on the efficiency and the pressure coefficient. Taking into account the theoretical studies, recommendations have been obtained on the selection of optimal parameters providing improvement the overall level of energy performance of a centrifugal pump in the area of unit feed

$$\dot{V}/\omega \leq 5 \cdot 10^{-7} \text{ m}^3$$

down to zero.

Recommendations were developed for the calculation of energy parameters of HSCP with impellers of semi-open and open types. Experimental data confirmed the basic theoretical principles. The results of studies are presented in form of generalized dependencies that allow to widely use them in engineering practice.

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