

ANALYTICAL AND EXPERIMENTAL ASSESSMENT OF SCREW CENTRIFUGAL PUMP AT IMPROVING ITS DESIGN

Purpose. Development of analytical and experimental assessment of screw centrifugal type pump at a design stage, which permits defining more precisely its power and cavitation characteristics.

Methodology. To achieve the above-mentioned purpose, the technique that includes the following possibilities was developed: make the list of changes which increase the efficiency of the existing pump; quickly and with high precision to estimate influence of certain constructive and/or regime changes on efficiency of the existing pump without difficult flow calculations; determine the need to develop a new pump, if all the changes of the existing pump did not give a positive result; determine changes influence on cavitation properties of the existing pump; calculate the efficiency dispersion during hydraulic tests of the modified pump.

Findings. During the research, an analytical and experimental technique which permits determining increase in the pump efficiency quickly and with high precision at the change in pump design and/or operating mode was developed. The developed technique determines how all changes in the existing pump influence its anti-cavitation properties. The above mentioned technique also allows calculating the efficiency dispersion during hydraulic tests of the modified pump. The technique determines the need to develop a new pump, if all the changes in the existing pump did not give a positive result. The use of the presented calculation technique, at a design stage, allows estimating the efficiency of the developed pump more precisely. The developed technique allows increasing the efficiency assessment accuracy and cavitation characteristics of low-flow high-speed screw centrifugal pump.

Originality. During the research, new empirical dependences were obtained that permit defining more precisely power and cavitation characteristics of low-flow high-speed screw centrifugal pump.

Practical value. The presented technique at a design stage allows estimating the developed pump efficiency more precisely. Due to this, it is possible to reduce the time for pump development and its development tests.

Keywords: *pump efficiency, cavitation speed coefficient, low-flow high-speed pump*

Introduction. The current stage of space rocket technology is characterized by an increased level of competition. To win this competition, it is necessary to reduce the cost and development time of products as well as to improve the design operability.

The most effective way to attain the objectives is the use of high-reliability elements and units of TP, whose operability has been repeatedly tested as a part of engine [1]. This makes it possible to reduce the cost and development time of TP and the engine as a whole. However, the efficiency of existing units (pumps) often does not meet modern requirements for energy perfection. To increase the existing pump efficiency, it is necessary to change its design and/or operating mode. It follows from the foregoing that the technique is needed which permits to determine the increase in the existing pump efficiency quickly and with high precision at the change in the pump design and/or operating mode [2]. Introduction the autonomous power supply in this process also is very important [3]. In addition, it is necessary to estimate influence of each change on the cavitation characteristic of the existing pump.

Literature review. Results of the retrospective analysis showed that the work on increase in the energy performance is highly relevant [4, 5]. This question is investigated in all industries and for all types of pumps [6, 7]. Energy properties increase is performed in the following areas: development of the new mathematical models [8, 9] describing physics for each type of the pump in more detail, and the use of modern software such as ANSYS [10, 11]. The modern sources do not con-

tain any information about the increase in performance of a low-flow high-speed screw centrifugal pump of LRE (liquid rocket engine) energy. There is no integrated approach to increase the energy performance of small-sized high-speed screw centrifugal pump of LRE. It is necessary to perform the experimental and theoretical study and as a result, to develop a new technique that permits increasing low-flow high-speed screw centrifugal pump efficiency quickly and with high precision.

Purpose. At the engine RD861K development, the oxidizer pump should ensure the highest standards of energy performance. Its efficiency has to be 5 % higher than at the RD861 engine oxidizer pump (the pump prototype).

To increase the efficiency of the RD861K engine oxidizer pump, which is a prototype of RD861, the design of the existing pump should be changed. After changes in the design, the required pump efficiency was achieved. However, it took considerable material resources. The pump development took longer than planned, and continued for over 7 years.

Considering the above-mentioned, the presented research is necessary to create a new technique that includes the following possibilities:

- to propose the list of changes which can increase the efficiency of the existing pump;
- quickly and with high precision to estimate influence of certain constructive and/or regime changes on efficiency of the existing pump;
- to determine the need to develop a new pump, if all the changes in the existing pump did not give a positive result;
- to determine influence of changes on cavitation properties of the existing pump;

Initial data for analysis

Parameter	Symbol	Unit
Existing pump efficiency	η_v	—
Existing pump rotor speed	n_1	rpm
Developed pump rotor speed (if it is differ from n_1)	n_2	rpm
Existing pump rotor power consumption	$N_{r.v.}$	kW
Existing pump thrust	H	J/kg
Existing pump mass flow rate	G_N	kg/s
Existing pump internal efficiency	η_i	—
Existing pump disc efficiency	$\eta_{D.v.}$	—
Existing pump hydraulic efficiency	η_G	—
Existing pump volume efficiency	η_V	—
Existing pump mechanical efficiency	η_m	—
Developed pump efficiency (required)	$\eta_{D.r.}$	—

- to calculate the efficiency dispersion during hydraulic tests of the modified pump.

Methods. To fulfill all of the above requirements, an analytical and experimental technique was developed. This technique permits determining the pump efficiency value at the change in the design or/operating mode without difficult flow calculations. The list of changes is presented in the above-mentioned technique.

This technique is intended to determine the efficiency value of low-flow high-speed screw centrifugal pump of LRE TP. Characteristics of the pump are presented below.

This technique can be used for the increase in the pump efficiency with characteristics that are outside the ranges indicated below. In this case, it is necessary to take into consideration the fact that the calculated result may greatly differ from the experimental one. The proposed technique does not consider the change in the pump thrust. To ensure the required thrust pump, either the diameter of the impeller is reduced or the rotational speed of the rotor is increased. In case when rotor speed cannot be increased, the internal diameter of the impeller should be increased.

The proposed analytical and experimental technique is based on the empirical dependences obtained during in-depth study and comprehensive analysis of the energy performance of five low-flow high-speed screw centrifugal pumps and two high flow low-speed screw centrifugal pumps of TP LRE.

For low-flow high-speed screw centrifugal pump

- ω from 2600 to 4400 1/s;
- Q from 0.003 to 0.017 m³/s;
- H from $6 \cdot 10^3$ to $21.6 \cdot 10^3$ J/kg;
- n_s from 22 to 79.

For high flow low-speed screw centrifugal pumps

- ω – 1351 and 2019 1/s;
- Q – 0.053 and 0.166 m³/s;
- H – 8142 and $29.8 \cdot 10^3$ J/kg;
- n_s – 76.5 and 70.0.

Results. Below the technique that permits determining the pump efficiency at the following changes in the design is presented:

1. To reduce the clearance between floating ring and impeller shoulders.
2. To increase the pressure behind the floating ring installed at the shoulder of impeller’s back disk, up to 340 % of the initial value.
3. To use the variable pitch screw instead of constant pitch.
4. To reduce the impeller blades thickness at the outlet.
5. The presence of the bypass holes in the main impeller disk.
6. To reduce the diameter at the impeller inlet.
7. To reduce the clearance between impeller and bend.
8. To increase the number of impeller blades.
9. To increase the rotor speed.
10. To reduce the impeller external diameter.

Initial data for calculation according to the above mentioned technique are presented in Table 1.

1. To reduce the clearance between floating ring and impeller shoulders. It should be noted that the clearance between a floating ring and impeller shoulders could be reduced up to the value which ensures the pump operability. The further decrease in the clearance will provoke the impeller blocking and as a result, the pump destruction.

To determine the pump efficiency by reducing the clearance between a floating ring and impeller shoulders, it is necessary to select the pump whose value n_s would be very near this one, according to Fig. 1. According to the dependences presented in Fig. 1, determine the efficiencies for both existing and developed pumps at reducing the clearance between a floating ring and impeller shoulders ($\Delta slot$). Value $\Delta slot$ in millimeters is determined according to formula, which takes into consideration the geometry of the groove seal

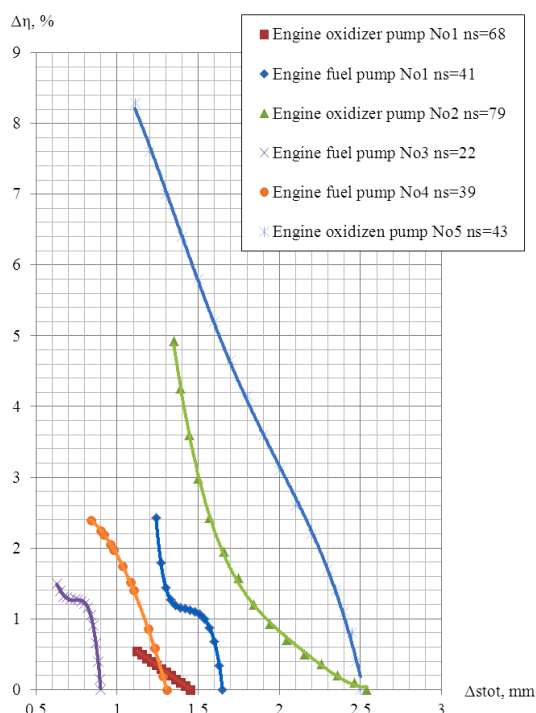


Fig. 1. The efficiency increasing per $\Delta slot$ decreasing

$$\Delta slot = \frac{\delta_{slot} \cdot D_{slot}}{l_{slot}}$$

where δ_{slot} is diameter clearance between floating ring and impeller shoulder; D_{slot} is a diameter on which the impeller shoulder is located; l_{slot} is the length of the slot in the axial direction.

According to formula, determine the developed pump efficiency increase, at decrease in the $\Delta slot$ value

$$\eta_{trun} = (\eta_2 = f(\Delta slot2)) - (\eta_1 = f(\Delta slot1)),$$

where η_2 is the efficiency of the developed pump, determined according to Fig. 1; η_1 is the efficiency of the existing pump, determined according; $\Delta slot2$ is $\Delta slot$ for the developed pump; $\Delta slot1$ is $\Delta slot$ for the existing pump.

It should be noted that the dependencies shown in Fig. 1 can also be used to determine the pump efficiency decrease if during its development it is necessary to increase the clearances between a floating ring and impeller shoulders.

The parameter to determine the efficiency of the developed pump is η_{turn} (4).

2. To increase the pressure behind the floating ring installed at the shoulder of impeller's back disk, up to 340 % of the initial value. The increase in the pressure behind the floating ring installed at the shoulder of impeller's back disk reduces leakage through the floating ring, but at the same time provokes the change in the pump axial force value and influences the bearings operability. It is necessary to pay special attention to this change. To determine the pump efficiency increase, at the pressure increase behind the floating ring installed at the shoulder of impeller's back disk, a formula is used.

$$\eta_{p.f.r.} = 0.00127887 \cdot P_{\text{seal}}^2 - 0.0315047 \cdot P_{\text{seal}} + 45.997452, \quad (1)$$

where P_{seal} is pressure behind the floating ring installed at the shoulder of impeller's back disk, kgf/cm².

The developed pump efficiency increase is determined according to the formula

$$\Delta\eta_1 = (\eta_{p.f.r.2} = f(P_{\text{seal}2})) - (\eta_{p.f.r.1} = f(P_{\text{seal}1})),$$

where $\eta_{p.f.r.1}$ is existing pump efficiency according to the (1); $\eta_{p.f.r.2}$ is developed pump efficiency according to the (1); $P_{\text{seal}1}$ is existing pump pressure, behind the floating ring installed at the shoulder of impeller's back disk; $P_{\text{seal}2}$ is developed pump pressure, behind the floating ring installed at the shoulder of impeller's back disk.

Parameter for determine the efficiency of the developed pump is $\Delta\eta_2$ (4).

3. To use the variable pitch screw instead of constant pitch. The variable pitch screw instead of the constant pitch increases the pump efficiency up to 1.3 %. This proposition is fair for the pumps with n_S value in a range from $n_S = 22.4$ to $n_S = 55.9$ (experimental data). If value n_S of the developed pump is outside the range indicated above, the efficiency of the pump with the variable pitch screw instead of the constant pitch will increase up to 1 %.

The parameter to determine the efficiency of the developed pump is $\Delta\eta_3$ (4).

4. To reduce the impeller blade thickness at the outlet. The impeller blade thickness could be reduced up to the value, which ensures the impeller strength. The further thickness decrease will provoke the damage or failure of the impeller, and as a result, the pump destruction.

The efficiency of the screw centrifugal pumps of TP LRE could be increased up to 1.8 %, at decreasing the impeller blades thickness at the outlet up to 40 % from their initial thickness.

The parameter to determine the efficiency of the developed pump is $\Delta\eta_4$ (4).

5. The presence of the bypass holes in the main impeller disk. The bypass holes in the main impeller disk influence the values of the pump axial force. It is necessary to pay special attention to this change.

Bypass holes increase the pump efficiency up to 3 % if the following recommendations are met:

- ratio between flow rate trough bypass holes and pump flow rate should be in range from 0.012 to 0.019;
- ratio between bypass holes speed and surface speed on the bypass circle should be in the range from 0.0775 to 0.097.

The parameter to determine the efficiency of the developed pump is $\Delta\eta_5$ (4).

6. To reduce the diameter at the impeller inlet. At the decrease in the diameter coefficient from $K_{D0} = 6.85$ to $K_{D0} = 6.31$ (decrease the diameter at the impeller inlet), the pump efficiency increases up to 3 %.

The parameter to determine the efficiency of the developed pump is $\Delta\eta_6$ (4).

7. To reduce the clearance between the impeller and the bend. It should be noted that the clearance between the spiral collector tongue and impeller could be reduced up to the value that ensures the pump operability. The further decrease in the clearance will provoke the impeller contact with the pump

body, and as a result, damage of the pump flow section, and its possible destruction. The impeller touching the pump body of the oxidizer pump would cause it to catch fire. The increase in the clearance between the impeller and the tongue of the spiral collector, for low-flow high-speed pump (the $\delta_{opt} = 0.09$ to $\delta_{opt} = 0.05$) increases its efficiency on average by 3 %.

$$\delta_{opt} = \frac{\delta p}{R_{c.imp.}},$$

where δp is radial clearance between impeller blades and the spiral collector tongue; $R_{c.imp.}$ is the radius at which impeller blades end.

The limits for δ_{opt} and the optimal clearance between a bend and the impeller, providing minimum level of energy losses in a bend, were calculated by formula

$$\left(\frac{\delta p}{R_{c.imp.}} \right)_{opt} = (0.8 \dots 1.0) \cdot n_S. \quad (2)$$

However, this (2) cannot be used for low flow high-speed pumps. This is demonstrated above.

The pump efficiency increase at the decrease in the clearance between blades of the impeller and the tongue of the spiral collector is determined by formula

$$\eta_v = 145783314 \cdot \delta_{opt}^5 - 46393718 \cdot \delta_{opt}^4 + 5781991 \cdot \delta_{opt}^3 - 354075 \cdot \delta_{opt}^2 + 10657 \cdot \delta_{opt} - 123. \quad (3)$$

The developed pump efficiency increase at decrease δ_{opt} value is determined by formula.

$$\Delta\eta_8 = (\eta_{v.2} = f(\delta_{opt2})) - (\eta_{v.1} = f(\delta_{opt1})),$$

where $\eta_{v.1}$ is the existing pump relative efficiency value, is determined by the (3) for δ_{opt1} ; $\eta_{v.2}$ is the developed pump relative efficiency value, is determined by the (3) for δ_{opt2} ; δ_{opt1} – for the existing design; δ_{opt2} – for the developed design.

The parameter to determine the efficiency of the developed pump is $\Delta\eta_8$ (4).

8. To increase the number of impeller blades. The impeller extra blades can increase the pump efficiency by 1.5 %.

The parameter to determine the efficiency of the developed pump is η_9 (4).

9. To increase the rotor speed. It should be noted that the increase in the main characteristics (thrust and flow rate) of LRE pump, for example, during the engine, boosting, always provokes the increase in the rotor speed.

This calculation can be performed only when the rotor speed of the pump increases, but the flow rate and pump thrust remain as in the existing pump. Power consumption of the developed pump rotor is determined by formula (at increase in the pump rotor speed)

$$N_1 = N_{r.v.} \cdot \left(\frac{n_2}{n_1} \right)^3.$$

The useful power of the existing pump is determined by the formula

$$N_n = \frac{H \cdot G_N}{10^3}.$$

The pump power of the existing pump is determined by formula

$$N_{p.I.} = \frac{N_n}{\eta_i} + N_{r.v.}$$

The developed pump power is determined by formula (at increase of the rotor speed)

$$N_{p.1} = \frac{N_n}{\eta_i} + N_1.$$

The pump efficiency decrease, when only the rotor speed increases, is determined by formula

$$\Delta\eta_m = \left(\frac{N_n}{N_{P.I.}} - \frac{N_n}{N_{P.1}} \right) \cdot 100.$$

It is necessary to note, that in section 9 only calculated mechanical component influence on the general efficiency of the pump, is considered. To perform this calculation, all other components of the overall efficiency are taken as in the existing pump. Actually, at increase in rotation speed of the pump rotor and conservation of values of thrust and flow rate which correspond to initial rotation speed, total efficiency of the pump can both increase and decrease, and can remain unchanged.

The parameter to determine the efficiency of the developed pump is $\Delta\eta_m$ (4).

10. To reduce the impeller external diameter. When only the impeller external diameter decreases to 9 % of its nominal diameter, the disk pump efficiency increases. Disk pump efficiency is determined according to the chart presented in Fig. 2, where $\eta_D = f(\Delta D_{D.imp.})$; η_D is the value of disk efficiency increase at decrease in impeller external diameter; $\Delta D_{D.imp.}$ is the impeller external diameter in % by which it decreases.

If not only impeller, external diameter is changed, but also other parameters and elements of pump's design, it is necessary to make the calculation correction-using diagram shown in Fig. 2. The value of this correction is determined using the technique for disk efficiency calculation indicated in [2, 12].

The increase in pump efficiency at decrease in the impeller external diameter is determined by formula

$$\Delta\eta_D = (\eta_G \cdot \eta_V \cdot \eta_m \cdot \eta_D) - (\eta_G \cdot \eta_V \cdot \eta_m \cdot \eta_{D.v.}).$$

It is necessary to note that the dependence shown in Fig. 2 can also be used to determine the decrease in the disk efficiency, if during pump development it is necessary to increase the impeller external diameter.

The parameter to determine the efficiency of the developed pump is $\Delta\eta_D$ (4).

To determine the developed pump efficiency it is necessary to add or subtract the value by which it was changed from the efficiency initial value, when introducing or excluding some design elements or operating mode parameters.

Formula 4 will always be different, because it depends on the number of changes in the existing pump. This formula can be corrected depending on obtained experimental data.

Formula 4, which takes in consideration all of the above-mentioned changes in the design and the operating changes, is shown below. The plus sign means that after the changes in the design, the developed pump efficiency will increase. The sign \pm means that that after the changes in the design the developed pump efficiency can both increase and decrease. This is due to the fact that at certain values of some parameters, the specified design changes can increase the pump efficiency. The minus

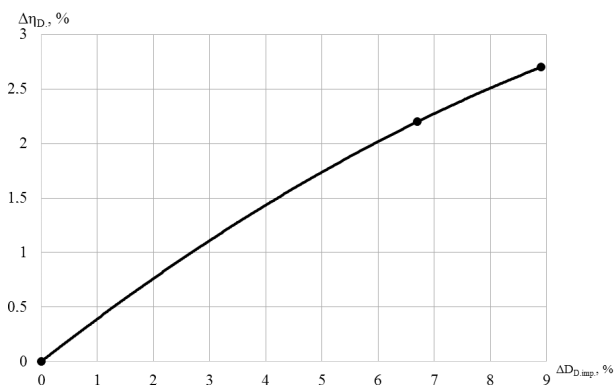


Fig. 2. Dependence of disk efficiency increase on impeller external diameter

sign means that after the changes in the design the developed pump efficiency will decrease

$$\eta_1 = \eta_v \pm \eta_{lum} \pm \Delta\eta_1 + \Delta\eta_3 + \Delta\eta_4 \pm \Delta\eta_5 + \Delta\eta_6 \pm \Delta\eta_8 + \Delta\eta_9 - \Delta\eta_m \pm \Delta\eta_D. \quad (4)$$

The condition 5 should be met

$$\eta_1 \geq \eta_{D.r.} \quad (5)$$

If condition 5 is not met, the new pump should be developed.

The analysis of the obtained results, based on the proposed technique and its comparison with the experimental data corresponding to the engine RD861K pump development showed that the calculated absolute efficiency of the pump is 0.2 % (4) less than experimental one. This fact shows that the data, calculated by proposed technique are valid.

Further, it is shown which design changes influence the decrease in the efficiency dispersion during hydraulic tests of the pump.

The following factors influence the decrease in the efficiency dispersion during the pump hydraulic tests:

1. The measurement error of the stand measurement equipment.
2. The tolerances on the dimensions of the pump flow section.
3. The specificities of pump flow section.

Points 1 and 2 are well known in the scientific world. The influence of specificities of pump flow section on the efficiency dispersion was determined during this research. This fact is demonstrated in Figs. 3 and 4.

The changes 6, 7 and 8 are presented above. The change in the meridional section and the angles of impeller blades at the inlet and at the outlet, allowed increasing the dispersion of efficiency from ± 1.5 to ± 1 % of the absolute value. All other changes do not influence the value of efficiency dispersion during hydraulic tests of the pump.

To determine the value of cavitation coefficient of pump specific speed C_{cr} at design changes specified above, on the ba-

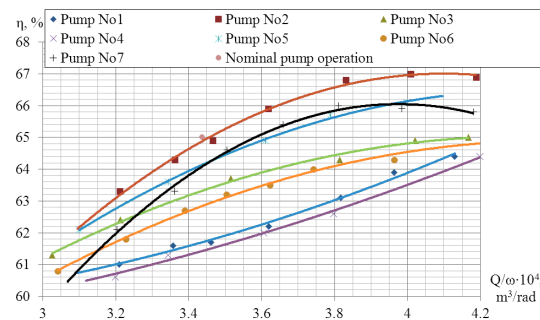


Fig. 3. Efficiency dependences of RD861 engine oxidizer pump on the volumetric flow rate corresponded to the rotational speed

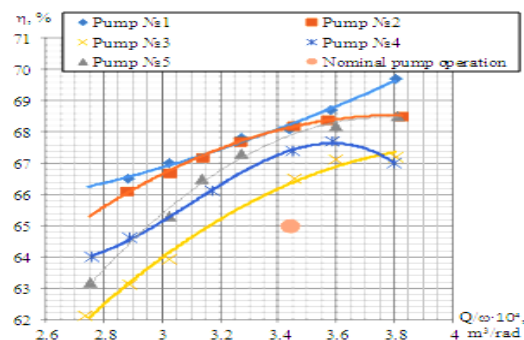


Fig. 4. Efficiency dependences of RD861K engine oxidizer pump on the volumetric flow rate corresponded to the rotational speed

sis of deep comprehensive analysis of experimental data, the analytical and experimental technique was created. It is intended for determining value of C_{cr} for high-speed low-flow screw centrifugal type pumps of LRE TP whose parameters are presented in the subsection “Research methodology”. This proposed technique can be used for the pumps, whose parameters are outside the range according to subsection “Research methodology”. However, it is necessary to take into account the fact that the calculated result can be different from the experimental data.

Below the technique for determination of C_{CR} value is presented. The existing pump can be changed as follows:

1. To reduce the clearance between a floating ring and the impeller shoulders.
2. To reduce the diameter at the impeller inlet.
3. To change the design of the groove seal at the impeller shoulder front disk.

Initial data for analysis is the value C_{CR} for existing pump.

To reduce the clearance between a floating ring and the impeller shoulders. When the clearance between a floating ring and the impeller shoulders decreases, the value C_{CR} increases. To determine the C_{CR} value it is necessary:

1. To select the pump prototype whose value n_s would be very near to the developed (Table 2).
2. If there is no appropriate prototype, select the existing pump according to Table 2, whose value n_s would be very near to the developed.
3. According to formula, determine the increase in ΔC_{CR1} when the value of $\Delta slot$ is decreased.

$$\Delta C_{CR1} = (\Delta C_{CR} = f(\Delta slot2)) - (\Delta C_{CR} = f(\Delta slot1)),$$

where ΔC_{CR} is determined according to the equation presented in Table 2.

The parameter to determine the cavitation coefficient of pump specific speed for the developed pump is ΔC_{CR1} (6).

To reduce the diameter at the impeller inlet. When the diameter at impeller inlet decreases from $K_{D0} = 6.85$ to $K_{D0} = 6.31$, the value C_{CR} decreases from 2.0 to 13.9 %.

Parameter for determine the cavitation coefficient of pump specific speed for the developed pump – ΔC_{CR2} (6).

To change the design of the groove seal at the impeller shoulder front disk. The design of the groove seal that is shown in Fig. 5 in comparison with other design variants, which are not presented in the following research permits increasing the cavitation coefficient of pump specific speed from 7.8 to 16.1 %.

Parameter for determining the cavitation coefficient of pump specific speed for the developed pump is ΔC_{CR3} (6).

The other design and operating mode changes, presented above do not influence the value C_{CR} .

To determine the value C_{CR} for developed pump, it is necessary to add or subtract the value by which it was changed

Table 2

Equations for determination of cavitation coefficient of pump specific speed

Pump prototype	Value n_s	Equations
Oxidizer pump	79	$\Delta C_{CR} = -1831 \cdot \Delta slot^4 + 14\,418 \cdot \Delta slot^3 - 41\,515 \cdot \Delta slot^2 + 50\,978 \cdot \Delta slot - 21\,662$
Oxidizer pump	39.4	$\Delta C_{CR} = -752.54 \cdot \Delta slot + 713.68$
Oxidizer pump	67.5	$\Delta C_{CR} = -334.19 \cdot \Delta slot + 483.68$
Fuel pump	40.5	$\Delta C_{CR} = -212.87 \cdot \Delta slot + 374.16$
Fuel pump	38.7	$\Delta C_{CR} = -6008.7 \cdot \Delta slot^2 + 8108.1 \cdot \Delta slot - 1871.9$

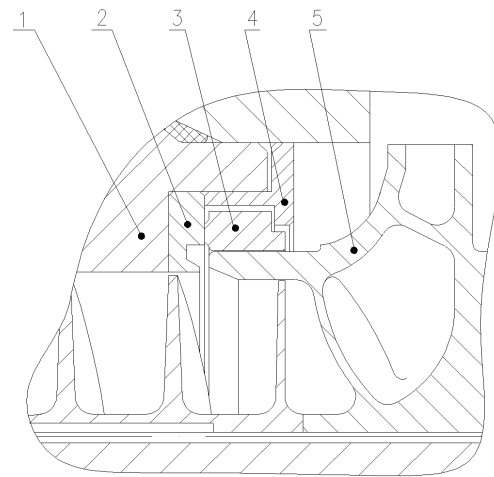


Fig. 5. Design of the groove seal with the best anti-cavitation properties:

- 1 – inlet brunch; 2 – stop ring; 3 – floating ring; 4 – cover; 5 – impeller

from C_{CR1} . initial value, after the developed pump change. Formula 6 will always be different, because it depends on the number of changes in the existing pump. This formula can be corrected depending on the obtained experimental data.

Formula 6 presented below takes in consideration all the above-mentioned design and operating mode changes which influence the C_{CR1} value. The sign \pm means that after the changes the value C_{CR1} at the developed pump can both increase and decrease. This is due to the fact that at certain values of some parameters the specified design changes can increase the value C_{CR1} .of the developed pump. The minus sign means that after the changes in the design the value C_{CR1} of the developed pump will decrease

$$C_{CR1} = C_{CR} \pm \Delta C_{CR1} - \Delta C_{CR2} \pm \Delta C_{CR3}. \quad (6)$$

The analysis of the obtained results based on the proposed technique and its comparison with the experimental data corresponding to the engine RD861K oxidizer pump development show that the calculated values of C_{CR} is 2 % (6) less than experimental one. This fact shows that the data calculated by proposed technique are valid.

Conclusions.

1. During the research an analytical and experimental technique was developed. This technique permits:

- quickly and with high precision to determine the pump efficiency at certain constructive and/or operating changes;
- determining how all changes in the existing pump influence on its anti-cavitation properties;
- calculating the efficiency dispersion during hydraulic tests of the developed pump;
- determining the need to develop a new pump, if all the changes in the existing pump have not given a positive result.

2. The presented technique permits determining more precisely the efficiency value of the developed pump, at the design stage. Due to that it is possible to reduce the duration and volume of the pump development tests, which in turn will decrease the design price.

3. The presented technique permits increasing the accuracy of estimation of cavitation characteristics and efficiency of low-flow high-speed screw centrifugal pump of LRE.

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Розрахунково-експериментальна методика визначення характеристик шнековідцентрового насоса при вдосконаленні його конструкції

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Мета. Розробка розрахунково-експериментальної методики, за допомогою якої ще на стадії проектування можна більш точно визначити енергетичні й кавітаційні характеристики шнековідцентрового насоса при вдосконаленні його конструкції.

Методика. Для досягнення вищезазначеної мети був усесторонньо проаналізований великий обсяг експериментальних даних по маловитратним високооборотним шнековідцентровим насосам. Аналізувався вплив конструктивних змін цих насосів на їх енергетичні та антикавітаційні характеристики.

Результати. У ході дослідження була розроблена розрахунково-експериментальна методика, за допомогою якої можна швидко та з високою точністю визначити величину приросту ККД насоса при зміні його конструкції і/або режиму роботи. Також розроблена методика визначає, яким чином усі зміни, внесені до конструкції насоса, впливають на його антикавітаційні властивості. За допомогою вищезгаданої методики можна визначити розрахунковим шляхом величини розкиду ККД при проведенні гідравлічних випробувань насоса, до конструкції якого були внесені зміни. Методика визначає необхідність розробки нового насоса, якщо всі конструктивні і/або режимні зміни, внесені до існуючого насоса, не дали позитивного результату. Використання представленої розрахункової методики на стадії проектування дозволяє більш точно оцінити величину ККД насоса, що розробляється. Розроблена методика дозволяє підвищити точність оцінки ККД і кавітаційних характеристик маловитратних високооборотних шнековідцентрових насосів.

Наукова новизна. У ході дослідження були отримані нові емпіричні залежності, за допомогою яких можна точніше визначити енергетичні й кавітаційні характеристики маловитратних високооборотних насосів.

Практична значимість. Використання представленої методики на стадії проектування дозволяє більш точно оцінити величину ККД насоса, що розробляється. За рахунок цього можна скоротити час на відпрацювання насоса та об'єм його доводочних випробувань.

Ключові слова: ККД насоса, кавітаційний коефіцієнт швидкохідності, маловитратний високооборотний насос

Recommended for publication by V. O. Habrinets, Doctor of Technical Sciences. The manuscript was submitted 27.02.21.