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Article

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RESEARCH AND DEVELOPMENT OF THE STRUCTURE OF A VORTEX HEAT GENERATOR BY THE METHOD OF MATHEMATICAL MODELING

The object of research is a mathematical model of a new design of a vortex heat generator with translational-rotational flow in a variable geometry working space.

One of the most problematic areas in the development of new and promising designs of heat generators by the method of physical modeling is the search for its optimal operating-technological and instrumental-design parameters. The implementation of a preliminary analysis of such structures by the method of mathematical modeling will significantly reduce the time and material costs for the development of promising designs of heat generators.

The studies of the design of the new vortex heat generator, carried out by the method of mathematical modeling, made it possible to determine the range of its operation, to evaluate the operating-technological and hardware-design parameters that affect the efficiency of work. Studies of the hydrodynamics of the translational-rotational motion of a viscous fluid flow in the working space of a new vortex heat generator with a variable geometry of the working space made it possible to determine the critical velocity and pressure, the influence of the geometric parameters of the device on the generation of vortices that promote cavitation. Model studies were carried out in the range of fluid load changes in the range from 0.001 m^3/s to 0.01 m^3/s. The study of changes in the velocity field in the channels was carried out for the geometry of the channel with a taper angle γ from 0° to 25°. The width of the working channel of the space W_n varied in the range of 130, 70 and 40 mm.

It has been established that a good axial symmetry and smoothness of the coolant flow in the vortex zone along the swirler screw provides the coolant inlet through a nozzle with a rectangular cross-section. The dependence of the influence of the flow area of the nozzle for introducing the coolant into the vortex zone on the energy efficiency of the vortex apparatus as a whole is found experimentally.

The research carried out makes it possible to design vortex heat generators with geometric parameters that meet modern energy efficiency requirements. The geometry of the swirler screw is determined, which increases the efficiency of the heat generator by 35 % in comparison with similar designs of vortex heat generators given in the literature.

Keywords: *vortex heat generator, translational and rotational flow, thermal energy, electrical energy, critical speed, mathematical model, cavitator.*

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1. Introduction

One of the energy efficient, environmentally friendly devices that convert the energy of the vortex motion of a liquid into thermal energy using cavitation is a vortex heat generator [1, 2].

A distinctive feature of a vortex heat generator is the use of a device of a certain geometry as a swirler of the fluid flow [3]. A high degree of swirling of the coolant flow ensures the organization of vortex flows in the working channel of a vortex heat generator with an increasing speed and a decrease in pressure to exit from it [4, 5]. This provides low coefficients of hydrodynamic resistance of the working channel of the vortex heat generator and contributes to the generation of cavitation processes.

Cavitation heat generators have a number of advantages over other heaters [6, 7]:

- installation of a heat generator does not require permits;
- cavitator works in autonomous automatic mode;
- an environmentally friendly source of energy, has no harmful emissions into the atmosphere;
- complete fire and explosion safety;
- efficiency of individual vortex heat generators, according to various estimates, approaches the efficiency equal to 0.9 and higher;
- water in the system does not form scale, no additional water treatment is required;
- can be used for both heating and hot water supply;
- characterized by low metal consumption, easily fit into the heating network.

At the moment, vortex heat generators are effectively used for heating residential buildings, industrial premises and agricultural complexes [4, 6]. Vortex cavitators are distinguished by the design of the working process into tubular [8, 9] and disk. With the design of one or another vortex heat generator, questions arise about the influence of the design features of the cavitator on the energy efficiency of its operation [10, 11].

In works [7, 12], the approach was investigated and recommendations were developed for the design of disk heat generators. These works were carried out in order to obtain critical parameters (pressure and velocity of the coolant) that ensure an effective cavitation process (formation and collapse of cavities, with the conversion of collapse energy into heat).

At the moment, studies of promising designs of vortex heat generators are devoted to finding the influence of geometric characteristics on the resulting energy efficiency [13]. Consequently, the study of the influence of the geometry of the new design of the vortex heat generator on its energy efficiency is an urgent task.

Thus, *the object of research* is a mathematical model of a new design of a vortex heat generator with a translational and rotational flow in a working space of variable geometry. And *the aim of research* is analytical verification of the influence of the design parameters of the vortex heat generator on the energy efficiency of its operation.

2. Methods of research

The motion of «Newtonian» liquid media provides for a special instrumental-constructive design of the hydrodynamic process of translational-rotational motion [14].

In the proposed work, a fundamentally new type of vortex hydrodynamic device for generating heat based on the rotational-translational motion of the medium, which contributes to the occurrence of the cavitation effect, has been developed, manufactured and investigated. As a result, the liquid heats up to a temperature of 60–80 °C.

On this basis, experimental studies have been carried out and a mathematical model has been developed for the influence of the design parameters of the device on its hydrodynamic performance.

A distinctive feature of the developed hydrodynamic heat generator is that the generator body is made in the form of a Venturi tube, inside the converging part of which there is a conical swirler screw with a decreasing pitch. The developed and manufactured vortex heat generator was installed in an experimental stand equipped with thermal control devices, on which studies of its hydrodynamic parameters were carried out (Fig. 1).

For optimal operation of the unit, it is necessary to have a mathematical model that takes into account the influence of the geometric dimensions of the working bodies on its hydrodynamic performance, depending on the properties of the working environment.

The design scheme for mathematical modeling of hydrodynamic motion inside the working space of the apparatus is shown in Fig. 2 [15].

In Fig. 2, designation V_1 shows the direction of movement of the medium, while the index number one corresponds to the entry of the medium into the channel, and the index two corresponds to the exit of the medium from the channel. In this case, the flow will occur in a negative direction relative to the radial axis, therefore, the velocity value must also be with a minus sign.

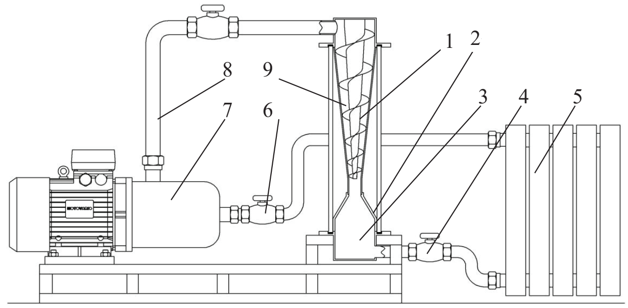

Fig. 1. General view of vortex cavitation device:
1 – main body (convergent channel); 2 – diffuser; 3 – brake chamber; 4 – outlet pipeline; 5 – heating system; 6 – return pipeline; 7 – pump; 8 – supply pipeline; 9 – screw-swirler flow

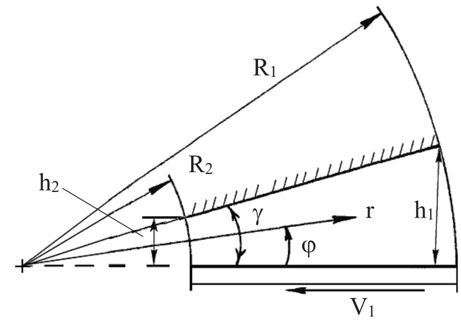

Fig. 2. Design scheme for modeling the speed mode of vortex cavitation device: h_1, h_2 – channel width at the inlet and outlet of the flow; R_1, R_2 – determining the size of the inlet and outlet of the convergent channel; γ – convergent channel opening angle; φ, r – coordinate axes; V_1 – direction of movement of the coolant

Assuming that there are no secondary flows in the working gap, the axial and angular velocity components can be equated to zero. Let's take into account only the radial velocity component:

$$V_r = V_r(r,\varphi); \quad V_y = V_z = 0. \tag{1}$$

The coordinate z is directed perpendicular to the diagram in Fig. 2.

In a stationary process, the equations of motion, taking into account the accepted assumptions and neglecting the change in physical quantities along the axial coordinate, have the form:

$$-\frac{\partial P}{\partial r}+\frac{1}{r}\cdot\frac{\partial}{\partial r}(r\cdot\sigma_{rr})+\frac{1}{r}\cdot\frac{\partial\sigma_{r\varphi}}{\partial\varphi}-\frac{\sigma_{\varphi\varphi}}{r}=0, \tag{2}$$

$$-\frac{1}{r}\cdot\frac{\partial P}{\partial\varphi}+\frac{1}{r^2}\cdot\frac{\partial}{\partial r}(r^2\sigma_{r\varphi})+\frac{1}{r}\cdot\frac{\partial\sigma_{\varphi\varphi}}{\partial\varphi}=0, \tag{3}$$

where $\partial P/\partial r$ – pressure change along the coordinate r; r – coordinate; σ_{rr} – stress tensor of deformation of the radial component of the fluid velocity along the coordinate r; $\sigma_{r\varphi}$ – stress tensor of deformation of the radial component of the velocity along the coordinate r, taking into account the angle φ; $\sigma_{\varphi\varphi}$ – stress tensor of the deformation of the radial component of the velocity along the coordinate φ.

In this case, the continuity equation looks like:

$$\frac{\partial V_r}{\partial r}+\frac{V_r}{r}=0. \tag{4}$$

In general, the radial velocity component is a function of two variables: radial and angular coordinates. Using the continuity equation (4), let's introduce the following replacement:

$$V_r(r,\varphi)=\frac{f(\varphi)}{r}, \tag{5}$$

where $f(\varphi)$ – function that depends only on the coordinate φ.

The components of the strain rate tensor, other than zero, for the considered scheme will have the form:

$$d_{rr}=\partial V_r/\partial r,\quad d_{\varphi\varphi}=V_r/r,\quad d_{r\varphi}=1/(2\cdot r)\cdot\partial V_r/\partial\varphi.$$

To pass from the stress tensor components in the equations of motion (2) and (3) to the velocity characteristics, let's use the rheological equation of state for a Newtonian fluid in the form:

$$\sigma_{ij}=2\cdot\eta\cdot d_{ij}, \tag{6}$$

where η – viscosity coefficient; i, j – indices, in this case corresponding to the coordinate axes r, φ.

After making the appropriate substitutions in equations (2) and (3), let's obtain:

$$-\frac{\partial P}{\partial r}+\frac{\eta}{r^3}\cdot\frac{\partial^2 f}{\partial\varphi^2}=0, \tag{7}$$

$$-\frac{\partial P}{\partial\varphi}+\frac{2\cdot\eta}{r^2}\cdot\frac{\partial f}{\partial\varphi}=0. \tag{8}$$

To get away from the pressure gradients in equations (7) and (8), let's differentiate equation (7) along the coordinate φ, and equation (8) – along the coordinate r. After substitution and transformation, let's obtain:

$$\frac{\partial^3 f}{\partial\varphi^3}+4\cdot\frac{\partial f}{\partial\varphi}=0. \tag{9}$$

The solution to equation (9) has the form:

$$f=\frac{C_1}{2}\sin(2\varphi)-\frac{C_2}{2}\cos(2\varphi)+C_3. \tag{10}$$

To determine the radial velocity component, let's use the substitution (5), and obtain:

$$V_r=\frac{1}{r}\frac{C_1}{2}\sin(2\varphi)-\frac{1}{r}\frac{C_2}{2}\cos(2\varphi)+\frac{C_3}{r}. \tag{11}$$

Equation (11) includes three constants of integration. Let's find two integration constants from the following boundary conditions:

$$V_r=0 \quad \text{at} \quad \varphi=0, \tag{12}$$

$$V_r=0 \quad \text{at} \quad \varphi=\gamma. \tag{13}$$

Let's find the third constant of integration from the equation of continuity in integral form:

$$W_n\cdot\int_0^\gamma r\cdot V_r\cdot d\varphi=Q_b, \tag{14}$$

where W_n – channel width; Q_b – performance of a device for supplying working fluid, for example, a pump.

Performing the appropriate substitutions and transformations, let's obtain expressions for the integration constants:

$$C_1=\frac{4\cdot Q_b}{W_n}\cdot\frac{\sin(\gamma)^2}{\sin(2\cdot\gamma)\cdot\gamma-2\cdot\sin(\gamma)^2};$$

$$C_2=C_1\cdot\frac{\sin(2\cdot\gamma)}{\cos(2\cdot\gamma)-1};\quad C_3=\frac{C_2}{2}. \tag{15}$$

The study of the operation of the vortex cavitator was carried out by the method of mathematical modeling on a previously developed mathematical model. The values of the initial parameters of the initial data for the calculation were taken as follows: $Q_b=2.66\cdot10^{-4}$ m^3/s; $\gamma=15°$; $W_n=80$ mm; $R_1=800$ mm; $R_2=70$ mm. For these values, the widths of the channels at the inlet and outlet were: $h_1=R_1\cdot\sin(\gamma)=0.207$ m; $h_2=R_2\cdot\sin(\gamma)=0.018$ m. The surface of the radial component values $V_r(r,\varphi)$ (11) was obtained for the ranges of variation: $r=R_2...R_1$ and $\varphi=0{,}1°...\gamma$.

3. Research results and discussion

The results of the performed calculations are shown in Fig. 3.

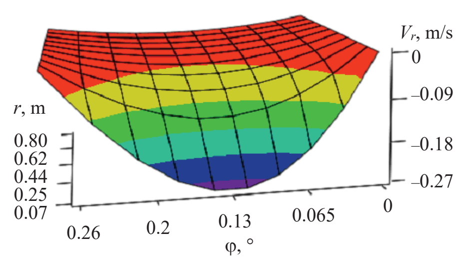

Fig. 3. Volumetric graph of the change in the speed V_r with the selected parameters

As seen from Fig. 3, with the selected parameters, the high-speed mode of movement of the coolant along the vortex cavitation device does not approach the critical speed of sound, which contributes to the occurrence of the cavitation process.

Fig. 4 shows the dependences of the variation of the geometric and pressure characteristics of the device in accordance with the developed mathematical version.

Fig. 4, *a* shows the change in the speed mode at different values of productivity ($V1_r$ at $Q_1=1\cdot10^{-3}$ m^3/s; $V2_r$ at $Q_2=3\cdot10^{-3}$ m^3/s; $V3_r$ at $Q_1=10\cdot10^{-3}$ m^3/s), and the unchanged parameters adopted when constructing the dependencies in Fig. 3.

As seen from Fig. 4, and at certain ratios of the parameters, there are areas of intense movement of the medium (the area between the curve $V1_r$ and $V3_r$), where the appearance of critical velocities preceding cavitation is observed.

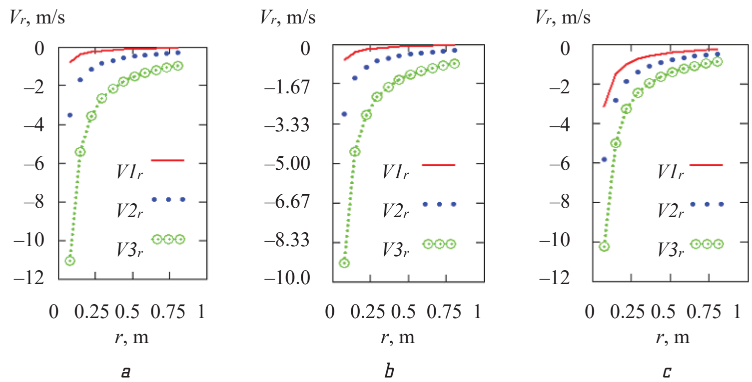

Fig. 4. Graphs of speed change V_r: *a* – at different values of pump flow rates Q_i; *b* – at different values of the inclination angles; *c* – for different values of the channel width W_n

Fig. 4, *b* shows the change in the speed mode V_r at different angles of inclination ($V1_r$ at $\gamma=25°$; $V2_r$ at $\gamma=11°$; $V3_r$ at $\gamma=5°$) and productivity values $Q_2=3\cdot10^{-3}$ m/s. From the analysis of the dependences it follows that the change in the angle of inclination has a significant effect on the speed regime in the working space of the heat generator.

In Fig. 4, *c*, the results of assessing the effect of the channel width W_n ($V1_r$ at $W_n=130$ mm; $V2_r$ at $W_n=70$ mm; $V3_r$ at $W_n=40$ mm) on the speed mode Vr are presented. In this case, the following parameters were taken: $\gamma=9°$, $Q_b=3\cdot10^{-3}$ m^3/s. The rest of the parameters have values corresponding to the construction of Fig. 4, *b*.

With an increase in the inclination angle of the wedge gap, the velocity V_r increases, which is associated with a decrease in the action of the friction force.

It should be noted that the graphs in Fig. 4 correspond to the middle of the channel, i. e., they are obtained at $\varphi=\gamma/2$.

4. Conclusions

A fundamentally new type of vortex heat generator with variable geometry of the working space has been developed, manufactured, and experimentally tested. A mathematical model with a simplified design scheme has been developed that simulates the movement of a medium inside a vortex channel with fixed elements. The influence of the main geometrical and pressure parameters of the device on its hydrodynamic performance was investigated on a mathematical model. The resulting model shows the critical regions where the most intense cavitation zones are possible.

The optimal angle of inclination of the convergent channel is determined, which corresponds to $\gamma=7°$, while the high-speed mode of movement of the coolant is the most effective from the standpoint of the device's heat generation. The article presents the first research results. Such parameters of the vortex cavitation device as: length; diameter; ratio of length and diameter; the angle of attack of the vortex device; the geometry of the nozzle for injecting the coolant is not taken into account in the presented model. Experimental testing and refinement of the presented model taking into account the specified geometric parameters is the object of further research.

The research results can be used in thermal and hydrodynamic calculations of vortex heat generators, in the development of the design of these devices with specified energy-technological indicators.

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