Parametric outline of a centrifugal pump as an element complex thermodynamic system

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Abstract.
The stability of the level of parameters of the working medium, which enters the combustion chamber of the heat engine, is realized due to the proper operation of all elements of the fuel supply system. The problems of selection, design and operation of one of the standard sizes of centrifugal pumps of fuel supply systems of heat engines are considered. On the basis of selected models of parameters of the first approximation, proposals are presented for computational and experimental studies of the outline. A method of mathematical consideration of the phenomenon of degradation of individual performance characteristics is proposed.

Keywords:
centrifugal pump
working wheel
efficiency coefficient
speed
model
profiling
divertor device
working fluid
**Introduction.** Centrifugal pumps are widely used in energy transport systems, fuel systems of vehicles and heat engines as pumps for pumping and pumping. In addition, such pumps are used to supply fuel to the afterburner chambers of turbojet engines, and in turbopump units of liquid rocket engines, centrifugal pumps are used as the main ones. The advantages of centrifugal pumps are: smooth and continuous supply of working fluid without pulsations and hydraulic shocks in the pressure line; ensuring high supply pressures and productivity with small dimensions and weight; the possibility of working with a high rotation frequency and the convenience of using a turbine for their drive; reliability and durability. This work presents the results of processing: calculation procedures in the first approximation of the main technical data of a centrifugal heat engine pump; possibilities of taking into account changes in the individual characteristics of the "heat engine - centrifugal pump" system during operation.

Yang and other authors (2012) justify the need to develop more accurate methods for predicting the performance of centrifugal pumps. A theoretical method for predicting pump performance is developed using theoretical analysis and empirical correlation. The authors used CFD to predict the performance of a single-stage centrifugal pump in forward and reverse modes. To obtain a more accurate CFD result, all areas within the control volume of the research object are simulated [1]. The operating characteristics of the modes were obtained, and theoretical and numerical methods for predicting the productivity of the blade machine were developed.

Tan, X., Engeda, A. (2016) used experimental data from a wide range of pumps representing the configuration of centrifugal pumps in terms of specific speed. Based on specific speed and specific diameter, a correlation is developed to predict characteristics at the point of best efficiency of a centrifugal pump in turbine mode [2].

Barbarelli et al. (2016) presented a one-dimensional numerical code evaluating the performance of centrifugal pumps. The program [3] calculates the geometrical components of the pump, using the information provided in the manufacturer's catalogs. Then, once these parameters are derived, losses are calculated and pump characteristics are
determined. The method is verified by comparing theoretical curves with some experimental measurements.

Michele Stefanizzi et al. (2018) propose a one-dimensional model for predicting pump performance, taking into account detailed machine geometric information, hydraulic losses, and the effect of flow deviation relative to the outlet vane angle during operation (Figure 1).

![Figure 1](image)

**Figure 1**
Runner losses: (a) The flow incidence loss; (b) Variation of the radial component of the inlet relative velocity [4]

The review of the literature confirms the relevance of continuing research on the search for concepts for increasing the efficiency of centrifugal pumps of various sizes and areas of application.

**Results and Discussions.** For supplying the working fluid in the amount of \( G \) at the supply pressure \( p_{sup} \), pump suction pressure \( p_{inp} \). \( G_M \) – mass consumption of fuel, which is determined from the thermogas-dynamic calculation of the engine. The volumetric flow rate of the working fluid through the pump is determined by the \( Q = G_M / \rho \). Pump head depends \( H = 10 \cdot \Delta p_{cp} / \rho = 10 \cdot (p_{sup} - p_{inp}) / \rho \). Another important value is the coefficient of speed (specific rotation frequency) which characterizes the rotation frequency of the reference pump, which is geometrically and in terms of efficiency similar to the natural one, but with a single pressure and power. The coefficient of speed is a dimensionless complex arising from the theory of similarity of pumps. We determine
the speed coefficient of the pump

\[ n_s = 3.65 \sqrt{\frac{Q}{H^3}} n \]  

(1)

\( n \) - frequency of rotation of the drive shaft of the centrifugal pump.

Model of losses in the pump due to its full efficiency \( \eta_{CP} \) are characterized by three components – volume \( \eta_{vol} \), hydraulic \( \eta_{hyd} \), mechanical \( \eta_{mech} \)

\[ \eta_{CP} = \eta_{vol} \cdot \eta_{hyd} \cdot \eta_{mech} \]  

(2)

1. Model of volumetric efficiency \( \eta_{vol} \) determines the amount of liquid flowing from the high-pressure cavity back into the low-pressure cavity, and the amount of liquid loss flowing from the high-pressure cavity through the seal

\[ \eta_{vol} = \frac{Q_{CP}}{Q_{vol}} \]. Size \( \eta_{vol} \) depends on the features of the pump design and the pressure level of the liquid being transported.

2. On the basis of the experimental data obtained during the test of a similar pump, we model the hydraulic efficiency coefficient by the flow coefficient \( \eta_{hyd} \) – the ratio of the actual head created by the pump to the theoretical head

\[ \eta_{hyd} = \frac{H}{H_T} \]. This coefficient characterizes the amount of hydraulic head loss in the pump, which consists of losses associated with flow disruption and shock at the inlet to the wheel, diffuser, discharge device and outlet pipe, and losses due to friction of the liquid against the walls of the channels.

3. We model the value of the mechanical coefficient of useful action \( \eta_{mech} \). Mechanical efficiency factor characterizes frictional power losses in bearings, seals, as well as frictional losses that occur when the pump wheel rotates in the liquid (disc friction). The amount of mechanical losses varies depending on the design of the pump.

We find the given diameter at the entrance \( D_{le} = 4.5 \cdot 10^3 \sqrt{\frac{Q}{n}} \).
Determine the power of the pump \( N_{CP} = \frac{QH1000}{75\eta} \). We determine the torque on the pump shaft \( M = 71620 \frac{N_{CP}}{n} \). Determine the diameter of the shaft \( d_{shaft} = \frac{3M}{0.2\sigma} \)

We assume that the diameter of the sleeve \( d_{sl} = 1.25 \cdot d_{shaft} \). We determine the actual flow of fluid through the wheel \( Q_0 = \frac{Q}{n_{CP}} \). We determine the diameter of the entrance to the pump \( D_o \) (Figure 2) \( D_o = 4.5 \cdot 10^3 \frac{Q_0}{n} \). We find the speed of fluid entering the wheel \( C_o = \frac{Q_0}{\pi(D_0^2 - d_{sl}^2)} \). Taking the slanted edge of the shoulder blade, we select \( D_1 = 0.95 \cdot D_o \). We find the width of the wheel at the entrance, assuming it beforehand \( \psi C_1 = C_0 \).

\( b_1 = \frac{Q_0}{\pi D_1 \psi c_1} \)

![Figure 2](image-url)

**Velocity triangles input (1), output (2) of the impeller**

We determine the peripheral speed of the pump impeller on the calculated diameter \( u_1 = \frac{\pi D_1 n}{60} \). We determine the angle of
entry of the working fluid $\beta'_1$ on the wheel without taking into account the compression of the flow by the vanes $\tan \beta'_1 = \frac{c_0}{u_1}$. We determine the additional angle of attack of the blade $\Delta \beta'$ to find the angle of inclination of the scapula. We find the angle of the shoulder blade $\beta_{1b} = \beta'_1 + \Delta \beta'$. For constructive reasons, we set the thickness of the blade at the entrance $\delta_1$, and also previously by the number of blades $z$. We determine the compression ratio of the flow at the entrance to the wheel $\psi_1 = 1 - \frac{\delta_1 z}{\sin \beta_{1b} \pi D_1}$. We determine the actual speed of the flow at the entrance $c_1 = \frac{c_0}{\psi_1}$. We determine the actual angle of fuel entry to the impeller $\beta_1$ and the valid angle $\Delta \beta = \Delta \beta_{1b} - \beta_1$. In this way, the basic data of the input to the centrifugal pump are calculated.

We determine in the first approximation the necessary peripheral speed of the wheel at the exit $H = k\left(\frac{u'_2}{g}\right)^2$; $u'_2 = \sqrt{\frac{gH}{k}}$. We determine in the first approximation the diameter of the wheel at the exit $D'_2 = \frac{60 u'_2}{\pi n}$. For the coefficient of speed of the pump $nS$, we choose the angle of the flow exit $\beta_2$. We check the correctness of the previously selected number of blades, for which we use the formula $z = 13 \frac{D_2 + D_1}{2(D_2 - D_1)} \sin \frac{\beta_1 + \beta_2}{2}$. We determine the angle of compression of the flow at the entrance at the thickness of the blade $\delta_2$

$$\psi_2 = 1 - \frac{\delta_2 z}{\pi D_2 \sin \beta_2} \quad (3)$$

We determine the required width of the wheel at the exit $b_2$ according to the equation

$$c_{2m} = \frac{Q_0}{\pi D_2 b_2 \psi_2} ; D_2 = \frac{Q_0}{c_2 \pi D_2 \psi_2} \quad (4)$$

We accept the width of the wheel at the exit, which can be implemented technologically. We determine the theoretical pressure of the wheel $H_T = H / \eta_{hyd}$. We determine the correction coefficient $\psi$ according to the equation, $\psi = 0.6 + 0.6 \sin \beta_2$.
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\[ p = \frac{\psi}{\pi^2} \left( \frac{1}{1 - \left( \frac{D_1}{D_2} \right)^2} \right)^2 \]  

(5)

The theoretical pressure can be determined using the Euler equation [4]. We find the theoretical pressure with an infinite number of blades \( H_T^\infty = H_T \left( 1 + \frac{1}{P} \right) \). We find the peripheral speed at the exit \( U_2 \) in a second approximation, using equation

\[ U_2 = \frac{C_{2m}}{2tg\beta_2} + \sqrt{\left( \frac{C_{2m}}{2tg\beta_2} \right)^2 + gH_T^\infty} \]  

(6)

We find the diameter of the wheel at the exit in the second approximation \( D_2 = \frac{60U_1}{\pi m} \). Since the difference in the value of \( D_2 \) compared to the first approximation is relatively small, we do not carry out further recalculation of the diameter of \( D_2 \). We perform profiling of the blade of the second circle. We determine the moment of the amount of fluid movement when it exits the wheel

\[ C_{2a} = U_2 - \frac{C_{2m}}{tg\beta_2} \]  

(7)

Profiling of the spiral diverter device is performed. The object of research is a pump with parameters presented in Table 1 and Figures 3, 4, which are used in the process of profiling the output device.

<table>
<thead>
<tr>
<th>№ section</th>
<th>( r, m )</th>
<th>( r = \frac{E_{vz}}{m} )</th>
<th>( C_s ), m/sec</th>
<th>( f_i ), m²</th>
<th>( Q_i ), m³/sec</th>
<th>( \psi_i ), degree</th>
<th>( R_i ), m</th>
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<td>0,0144</td>
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<td>45,5</td>
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<td>0,215</td>
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<td>0,2175</td>
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</table>

To reduce the speed of the liquid, we use a bladeless diffuser with the appropriate height. We take the shape of the
cross-section of the diverter device in the form of a circle, compile a table (table 1) of characteristic dimensions and construct graphs of characteristic dimensions (figures 3, 4).

According to the graphs, we find the necessary dimensions of the lead-off device in its arbitrary cross-section. The designed diverter has very high velocities, which can cause a deterioration of the hydraulic efficiency due to very high...
hydraulic losses. To reduce these losses, it is possible to use a blade diffuser.

We identify the parameters of the outlet pipe of the centrifugal pump. For the object of research, we take the shape of the cross-section of the pipe in the form of a circle and the initial diameter $D_{out}$, which is determined from the output area $F_{out} = \frac{Q}{c_{tube}}$. The length of the outlet nozzle is determined taking into account the corresponding angles $L = \frac{D_{out} - 2\varphi_{out}}{\pi \gamma}.$

Further calculation and experimental studies are carried out with a change in the values of the rotation frequency of the pump drive shaft, the pressurization pressure, and the mass flow rate of the working fluid according to the plan presented in Tables 2, 3.

<table>
<thead>
<tr>
<th>$G$, kg/s</th>
<th>52</th>
<th>54</th>
<th>56</th>
<th>58</th>
<th>60</th>
<th>62</th>
<th>64</th>
<th>66</th>
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<tr>
<td>$n$, rev/min</td>
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<td>4275</td>
<td>4300</td>
<td>4325</td>
<td>4350</td>
<td>4375</td>
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<tr>
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<td>22,3</td>
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<table>
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<td>19,1</td>
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<td>17,9</td>
<td>17,6</td>
<td>17,3</td>
<td>17,1</td>
</tr>
</tbody>
</table>

At the next stage, the possibilities of mathematical modeling of the degradation of the efficiency of the centrifugal pump during operation in the "pump – heat engine" system are considered. On the basis of previous studies, it is proposed to implement this by refining the loss model [5, 6]:

$$
\eta_{CP} = \eta_{vol} \cdot (1 - p_{vol}) \cdot \eta_{hyd} \cdot (1 - p_{hyd}) \cdot \eta_{mech} \cdot (1 - p_{mech}) \quad (8)
$$
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\( p_{\text{vol}}, p_{\text{hyd}}, p_{\text{mech}} \) - state parameters of models of volume, hydraulic, and mechanical losses, which take into account the possibility of changes in the efficiency components of a centrifugal pump during operation, degradation of individual characteristics relative to the average statistical system “pump ↔ heat engine”.

**Conclusions.** The problems of selection, design and operation of centrifugal pumps of fuel supply systems of heat engines were studied. On the basis of theoretical models of parameters of the first approximation of the pump, proposals are presented for calculation and experimental studies of the outline. A method of mathematical consideration of the phenomenon of degradation of individual efficiency characteristics (8) of a centrifugal pump is proposed for more accurate prediction of performance and efficiency characteristics during the life cycle.

**References:**


