Creation a universal technique of predicting performance curves for small-sized centrifugal stages of well oil pump units

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Abstract. The work provided is a continuation the development of research work on the process of pumping high-viscosity liquids and is aimed at creating a universal technique of recalculating the performance curves of dynamic small-sized pumps from water to high-viscosity liquid. The practical need to develop such a technique became relevant after obtaining different results when using existing methods of predicting performance curves for hydrodynamic oil well pumps. The expediency of carrying out studies aimed at clarifying the scope of the technique for calculating the work parameters of a well oil centrifugal pump, which is being developed, is substantiated, and more thorough analysis of the structure of conversion coefficients taking into account the scale factor and flow regime in the hydraulic flow parts of the studied pumps.

1. Introduction

Nowadays, pumping units with small centrifugal impellers (outer diameter – \(D_2=70–80\) mm) are widely used in the oil production complex for produced oil and stratum water. The main feature when working on oil is its viscosity, which varies in very large limits. Based on the analysis the results in the practical use of this type centrifugal pumps in the oil industry, it can be argued that the specified range of oil viscosity is from 15 cSt to 150 cSt. The properties and composition of the pumped medium significantly change both the characteristics of networks and pumping equipment, moreover, these changes during the operation of the well and submersible pump are variable. This requires non-stop monitoring to maintain consistency between the networks work and the pump in the rated operating range. Currently, such a technique, justified by the theory of blade machines, in its qualitative form, is absent. What exists at the moment does not take into account the factor of predicting the performance curves of dynamic submersible pumps pumping fluids with time-varying viscosity values and relatively small overall dimensions of the flow channels. For such working conditions, it became necessary to search for solutions to the creation of a universal technique for predicting the performance curves of small centrifugal pump stages when they operate on a variable viscosity of the pumped medium.

The first works on the problem of recalculating the performance curves of centrifugal pumps pumping highly viscous liquids appeared more than seventy years ago, but the topicality of the problem has not been lost for the time being.

There are several reasons for this. Firstly, by the exceptional complexity and incomplete knowledge of the phenomena occurring in pumps when pumping highly viscous liquids [1]. Secondly, the needs of engineering practice related with the necessity to develop highly efficient centrifugal
pumps for pumping petroleum products, industrial oils, liquid polymers and other viscous liquids. It is these reasons that explain the existence of various methodologies for recalculating performance curves to pumping highly viscous liquids [2-6].

Currently, to recalculate the performance curves of centrifugal pumps, are used experimental correction coefficients that allow the performance curves obtained when the pump is working on water to be converted to the characteristics when working on a highly viscous liquid [2-6]. This leads to the fact that the recalculations results of characteristics by different methods do not match [2]. This is explained by the fact that each method was developed for specific constructional parameters of the working pump stages with a certain range of changes in capacity, pressure, speed, viscosity, etc. It should also be noted that almost all currently existing methods were obtained using experimental data and are usually presented in the form of nomograms or empirical coefficients for recalculating the available performance curves of a pump on water. An analysis of the literature shows [2-6] that the available data mainly cover the region of conventional pumps with medium-sized $D_2 = 150-450$ mm and large-sized $D_2 > 450$ mm impeller diameters, and a fixed speed (usually not more than 3000 rpm). At the moment, there is no acceptable technique for predicting performance curves for small-size centrifugal stages (outer diameter – $D_2 = 70-80$ mm).

The rationale for the appropriateness and actuality of the research in the direction of deciding this problem was to conduct a computational comparative analysis using the most well-known existing methods for recalculating the performance curves of hydrodynamic pumps as applied to a small-size centrifugal stage.

The initial data for conducting a comparative analysis in determining the performance curves of the selected small-sized stage for pumping a highly viscous liquid are its parameters obtained when working on water. Corresponding parameters of the selected pump stage in nominal operating region: capacity – $Q = 50$ m$^3$/day, head – $H = 5.1$ m, pump efficiency – $\eta = 43\%$, rotational speed – $n = 2910$ rpm, impeller outer diameter – $D_2 = 72$ mm. The range of viscosity values of the working fluid was selected to correspond to the oil viscosities, on which work most pumps under studying from the commercially available series of centrifugal oil pumps. Accept the following viscosity values: $\nu = 10$ cSt; 45 cSt; 100 cSt.

The result of a comparative research of existing methods in a visual form is represented by a graphical dependency (Figure 1) of the change in the main energy parameter – pump efficiency $\eta$, % ($Eff.$), from various viscosity values of the pumped liquid $\nu$, cSt.

![Figure 1](image_url)

Figure 1. Comparative characteristics of the pump efficiency values obtained by different methods:
1 – by the method of Y Balakirov; 2 – by the method of P Lyapkov; 3 – by the method of Sulzer; 4 – by the method of Hydraulic Institute Standards; 5 – according to the generalized method of M Aizenshtein, D Sukhanov and A Stepanov; 6 – by the method of Doctor of Science V Boyko.
The conducted analysis confirmed the statement that the obtained results do not correspond between themselves using the existing recalculating methods. To confirm this fact, it was detected that the maximum divergence in the values of efficiency are $32\%$, $63\%$, $76\%$, for viscosities of $10$ cSt, $45$ cSt, and $100$ cSt, respectively. This argumentation objectively proves the statement that each of the considered methods was created for specific conditions for a particular type of pumping units that perform various tasks when pumping highly viscous medium.

The scatter of the obtained data indicates that the issue of the influence and consideration of viscosity requires further research. In this case, it is necessary to take into consider not only the absolute value of the viscosity coefficient, but also the type of liquid (Newtonian, non-Newtonian), the type of pumped medium (homogeneous mixture with gas or another liquid), etc.

Thus, it can be argued that there is no acceptable technique for predicting performance curves for small-size centrifugal stages. Available techniques for the most part are developed for medium and large-sized types of impellers with a flow rate range of $10$ m$^3$/h and high head. Existing methods work quite well for these sizes range. In the analysis of this problem, it was found that the main factor which influence on the accuracy of the obtained results is the scale effect. The physical process of the flow a pumped viscous medium in small-sized hydraulic flow parts has a significant difference from the corresponding process in hydraulic flow parts of large-sized centrifugal pumps [7,8].

In existing methods for predicting the performance curves of dynamic pumps, recalculation coefficients do not always take into account all the necessary information, which leads to contradictions. To decide this problem, it is necessary to uncover the structure of recalculation coefficients, to establish an unambiguous relationship between themselves and flow parameters, which will allow to create a universal technique suitable for recalculation performance curves of blade pumps in a wide range of designs and their work regime. The best approach in this issue – is the use of the apparatus of classical hydromechanics.

2. Developing the recalculating technique

An analysis of the literature showed that to obtain only by theoretical methods, a technique for recalculating the performance characteristics obtained on water to the characteristics for pumping highly viscous liquid it is not possible because of the complex dependence of hydraulic energy losses on the viscosity pumping medium. To find out the structure of recalculation coefficients, it was necessary to study the physical nature of the processes occurring in the inter-blade channel of the enclosed type impeller a normal specific speed with single suction side. The study process found that in all methods for recalculating the performance curves of centrifugal pumps from water to high viscosity fluid, the recalculation coefficients of the pump parameters are presented as functions of the Reynolds number [4-6], but the method for determining the latter in each of these works is different.

In its most general form, the Reynolds number is the ratio of inertia forces to viscosity forces, from the theory of turbomachines, the circumferential velocity at the output of the impeller is adopted as a characteristic velocity $w = u_2 = \omega R_2$, where: $\omega = (\pi n)/30$ – angular wheel speed; $n$ – rotational wheel speed, rpm; $R_2 = D_2/2$ – characteristic geometric dimension, which is the external radius of the impeller, wherein, $D_2$ – impeller external diameter. Accordingly, the Reynolds number is determined by the formula:

$$ Re = \frac{\omega R_2^2}{\nu} \quad (1) $$

In the techniques of M Aizenstein, D Sukhanov and A Stepanov the Reynolds number, in relation to which, as functions, are given the recalculation coefficients of the pump parameters from water to a viscous liquid, is determined by the formula:

$$ Re = \frac{Q_{opt}}{D_{eq} \nu} \quad (2) $$
Q_{\text{opt}} – capacity at best efficiency point in case of work on water, m³/s;
ν – kinematic viscosity coefficient, m²/s;
D_{\text{eq}} – equivalent impeller diameter determined by the formula:

\[
D_{\text{eq}} = (4 \cdot D_2 \cdot b_2 \cdot \psi_2)^{0.5}, \text{m}
\]

D_2 – as mentioned above, impeller external diameter, m;
b_2 – the width of the flow channel at the outlet of the impeller, m;
\psi_2 – the constrain coefficient of the flow at the outlet of the impeller by its blades, which is determined by the formula:

\[
\psi_2 = 1 - \frac{z \delta_2}{\pi \cdot D_2 \cdot \sin \beta_2}
\]

z – number of blades in impeller;
\delta_2 – the width of the blade at the outlet of the impeller, m;
\beta_2 – the angle of the blade at the outlet of the impeller, or the exit angle of the fluid flow from the impeller in the relative system, deg.

From our point of view, the most accurate determination of the Reynolds number is the method according to formula (2), since it contains the main geometric parameters of the impeller outlet as a variable part. For this, it became necessary to determine the formula for transforming into each other the Reynolds numbers determined by formulas (1) and (2). Accordingly, we will also denote them Re₀ and Re.

When pumping a viscous fluid, the flow rate in the curved blade channel of the impeller will be determined mainly by viscous friction forces and inertial forces:

\[
Q = f_1(T, I)
\]

herein: T = f_2(\mu) – friction force, which is a function of dynamic viscosity coefficient \mu, I = f_3(m, a) – inertia force, which is a function of mass m and acceleration a.

Since for viscous fluids T >> I, then, without loss analysis accuracy inertia forces can be excluded, i.e.: Q_{\text{vis}} = f_4(T) = f_5(\mu). Flow of viscous fluid through impeller inter-blade channel Q_{\text{vis}} can be expressed from the Poiseuille formula:

\[
Q_{\text{vis}} = \frac{\pi d^3 \Delta P}{128 l \cdot w \cdot \rho} \cdot \frac{w \cdot d \cdot \rho}{\mu}
\]

d – characteristic impeller diameter, m;
\Delta P – pressure difference between inlet and outlet of the channel, Pa;
l – channel length, m;
w – fluid velocity, m/s;
\mu – dynamic viscosity coefficient, Pa·s;
\rho – fluid density, kg/m³.

Thus, the flow rate of viscous fluid in the channels of the impeller is determined as a function of the Reynolds number:

\[
Re = \frac{w \cdot d \cdot \rho}{\mu}
\]

Given the mentioned above equations in its definition, after a series of mathematical transformations, was obtained a translation formula in the definition of the Reynolds numbers under consideration:

\[
Re = k_g \cdot \frac{\omega \cdot R_2^2}{\nu} = k_g \cdot Re_0
\]

k_g – geometric coefficient, taking into account the basic geometric parameters of the impeller outlet.

Using the velocity triangle at the outlet of the impeller to determine the values of the absolute
velocity and the component of flow rate of the absolute velocity, the formula for determining the geometric coefficient was derived:

\[ k_g = \frac{2 \pi b_2 \psi_2}{d_{eq}} \cdot \frac{\sin \beta_2 \sin \alpha_2}{\sin(180 - (\alpha_2 + \beta_2))} \]  

(7)

\( \alpha_2 \) – angle at which the absolute velocity vector is directed at the outlet of the impeller, deg.;

\( \beta_2 \) – angle at which the relative flow velocity vector is directed at the outlet of the impeller, deg.

After conducting simple mathematical transformations (taking into account the influence of inertia forces), obtain the formula of the recalculation coefficient for the pump capacity:

\[ k_Q = \frac{Re}{a + b \cdot Re} = \frac{1}{a + b \cdot \frac{Re}{Re}} \]  

(8)

Formula (8) discloses the structure of the recalculation coefficient for the pump capacity, since it clearly represents it as a dependence on the number Re (on the speed, the characteristic linear channel size and, most important – viscosity).

Based on the conclusion that it is possible to consider the flow regime in the inter-blade channels of the impellers of the pumps under consideration as laminar – was obtained a system of equations for determining the recalculation coefficients of the main parameters of a centrifugal pump for the case of changing the dimensions of its working parts and the viscosity of the pumped medium. As a result of the calculation analysis, it was determined that the recalculation coefficient for the pump capacity \( k_Q \) is the main recalculation coefficient. Recalculation coefficient for the head \( k_H \), for the power \( k_N \) and for the \( \eta \), \( k_\eta \), are derivative from \( k_Q \). This system of equations has the form:

\[
\begin{align*}
  k_Q &= \alpha^{-1} \\
  k_H &= \alpha^{-2/3} \\
  k_N &= \alpha \cdot k_p \\
  k_\eta &= \alpha^{-8/3}
\end{align*}
\]  

(9)

\( \alpha = a + b/Re \) – structural parameter;

\( a \) and \( b \) – empirical coefficients that are determined in the process of experimental research for a concrete design of the pump impellers and the viscosity range of the pumped liquid;

\( k_p \) – the ratio of the water density and a viscous liquid.

### 3. Experimental study

To determine the indicated empirical coefficients (\( a \) and \( b \)) as applied to centrifugal pumps with small-size impellers (outer diameter \( D_2=70–80 \) mm) was performed a special experimental research on the developed bench. Based on the results of the analysis of the obtained experimental data, were established acceptable values ranges for the viscosity of the pumping medium by the experimental pump at a rotational speed of its rotor \( n = 3000 \) rpm. Three main viscosity ranges are determined for the values of empirical coefficients \( a \) and \( b \) (Tab.1).

| Viscosity ranges (cSt) | \( a \) | \( b \) |
|------------------------|-------|-------|
| (10 – 40)              | 1,1   | 150   |
| (40 – 70)              | 1,1   | 170   |
| (70 – 110)             | 1,1   | 400   |

Applicable for centrifugal pumps of traditional design with medium-sized impellers (outer diameter
and rotational speeds up to 3000 rpm, these coefficients have values – \( a = 1 \) and \( b = 160 \).

Obtained formulas (9) make it possible to use independent experimental data on the issue under consideration, where the recalculation coefficients of the pump parameters are represented depending on \( Re \) calculated by the formula (1).

Taking into account the system of equations (9), the method of recalculating the performance curves of centrifugal pumps, which takes into account the relationship of the geometric parameters of its working parts, the rotation speed and viscosity of the pumped medium, is reduced to using the next system of dependencies (10):

\[
\begin{align*}
Q_v &= k_Q \cdot Q_W \\
H_v &= k_H \cdot H_W \\
N_v &= k_N \cdot N_W \\
\eta_v &= k_\eta \cdot \eta_W
\end{align*}
\]  

(10)

\( Q_W, H_W, N_W, \eta_W \) – parameters of a centrifugal pump from his performance curves on water in the operating range \( 0.7Q_{opt} – 1.2Q_{opt} \).

An experimental verification of this technique was carried out, and the operating bounds of its application are determined, in which calculation errors correspond to the requirements of standards in the field of pump engineering. The developed technique allows with the accepted accuracy to recalculate the performance curves of the pumps under consideration, in the range \( 0.7Q_{opt} – 1.2Q_{opt} \), where \( Q_{opt} \) – pump flow rate at which its maximum efficiency is achieved.

Further use of the obtained technique for recalculating the performance curves of small-sized centrifugal pumps from water to a highly viscous liquid let take results of recalculating that are quite close to experimental data.

Comparative results of calculated and experimental data in ranging from \( 0.8Q_{opt} \) to \( 1.2Q_{opt} \) at rotational speed \( n = 3000 \) rpm, presented on the characteristics (Figures 2, 3 and 4) expressed in a form of dimensionless coefficients: capacity – \( \varphi = \frac{Q}{\pi D_2^2 \bar{u}_2} \), head – \( \psi = \frac{2gH}{\bar{u}_2^2} \) (in these formulas, the velocity flow at the outlet of the impeller \( \bar{u}_2 = \omega \cdot R_2 \)), power – \( \lambda \) and efficiency \( Eff. – \eta \) (designations in figures: 1 – on water; 2 – \( \nu = 50 \) cSt; 3 – \( \nu = 90 \) cSt; dashed line – calculated characteristics, solid line – experimental characteristics).

The results of an experimental research of small-sized stages of oil pumps indicate that the created universal semi-empirical method for predicting the performance curves of pumping units with small capacity and low head pressure gives good results in the flow rate range from 20 m³/day to 200 m³/day at changing the viscosity of the pumped medium from 10 cSt to 110 cSt at rotational speed 3000 rpm. The indicated range of parameters is in good agreement with the real operating conditions of centrifugal pumping stages in the oil production complex.

\[ \text{Figure 2. Dimensionless characteristics of the dependence of the head on the pump capacity at variable viscosity.} \]
4. Common research results

According to the results of an experimental study, it was determined:

1. With low capacity and high viscosity, due to the low efficiency of the pump with a small-size stage, the pumped fluid quickly heats up in the flow channels, and as a result, the viscosity decreases with subsequent reduction of losses in the pump;

2. Losses on disk friction at established angular wheel speed increase to a greater extent than with increasing number of rotational wheel speed, consequently, when pumping viscous liquids in order to increase efficiency, it is necessary to increase the number of rotational wheel speed and apply pump stages with the most possible specific speed;

3. May to speak of an acceptable limit value of viscosity at which the use of the studied working pump stages will be justified. So was established that with an increase in viscosity over 110 cSt, resistance in the flow channels increases to the level of power loss to overcome them comparable (and even more) with the power of the stage itself.

4. With an increase in the viscosity of the pumped liquid, the theoretical head increases...
proportionally (insignificantly) in comparison with the theoretical head obtained on water, due to a significant reduction in vortex loss in the flow channels of the working stage and the leveling of the velocity diagram in the flow structure;

5. In a comparative and evaluative analysis of losses, was considered the impact of the scale effect on their change. As a result of the experimental study, it became a statement that the main influence factors on the value of the change in energy loss in these working parts is the specific speed coefficient and Reynolds number $Re$. Therefore, the complex dependence of losses at pumping a highly viscous liquid becomes apparent.

6. The idling power of the pump unit clearly depends on the viscosity of the pumped liquid and, as a percentage, is on average 40% –50% of the power of the stage itself. This fact became necessary for recommendations in design decisions when creating a drive for pumping units of a standard size range of centrifugal oil pumps. Lack of accounting for idling will lead to a drop in the efficiency of the pumping units in the conditions of its operation in comparison with the calculated efficiency.

It is planned to continue experimental research in this direction in order to verify the created technique for recalculating the characteristics curves of pumps taking into account the factor of high rotor speed.

5. Conclusion
1. Based on the analysis of the physical processes of fluid flow in the inter-blade channels of a small-sized impeller of a centrifugal pump, the structure of the recalculation coefficients of their working characteristics from water to highly viscous liquids is established.
2. Established the connection of recalculation coefficients for volumetric, hydraulic and mechanical efficiency of a centrifugal pump with a structural parameter depends on the Reynolds number.
3. A technique is proposed for recalculating the characteristics curves of small-sized centrifugal pumps when they transition to pumping highly viscous liquids.
4. Comparison of the results of the recalculation proposed technique with the results of an experimental study shows their good agreement, which confirm the appropriateness to using the developed technique for the practical operation of well small-sized oil centrifugal pumps.

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