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Calculation and designing of volutes of rotodynamic pumps

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Abstract. This work deals with improvements of the calculation and design methods of rotodynamic pump volutes taking into account viscous liquid flow. The research was performed with torque flow pump of "Turo" type. The analytical solution for the problem within the one-dimensional model of viscous liquid allowed to obtain a viscous liquid flow pattern in the volute. The law of velocity variation of viscous liquid flowing through the volute was obtained. There are numerical solutions of the problem concerning of viscous liquid flow in the pump with volute designed for traditional model of non viscous and viscous fluid. The research results are given by means of head-capacity and power input curves. To estimate the results of numerical study the tests were performed on the test stand.

1. Introduction

Rotodynamic pumps are energy machines. They are widely used in many branches of the industries and sectors of the national economy. Today's tendency of the development of rotodynamic pumps is connected with the increasing of their efficiency without any change in their mass and dimension. Following the energy balance the part of the hydraulic losses occurs in the volute chamber. The decrease of such losses within the total losses balance of a pump needs the improving of the calculation and design methods of pump volute especially in case of viscous liquid.

One of the most effective methods with regard to minimum energy losses is a volute. It consists of a volute chamber casted integrally with a pump casing and a diffuser nozzle.

Volute shall provide axially symmetric flow downstream of an impeller and thus producing steady flow in the impeller, convert part of the kinetic energy of the flow downstream of the impeller into the pressure energy and direct flow to the discharge nozzle or to the next pump stage.

Volute designing is carried out for the rated conditions of the pump. Therefore, a relatively small circumferential non-uniformity of the flow is observed only within a rather short operating range, close to the optimum condition. Beyond the above mentioned range the non-uniformity of the flow around the impeller increases greatly. If this occurs the hydraulic losses as well as pressure pulsations and radial forces also increase in the volute and impeller. The usage of the volute is economically feasible only at the rated conditions.

The existing methods for calculation of the volutes are based on a number of assumptions which simplify the physics of the liquid flow passing through the volutes. These assumptions include [1]:

- the liquid flow in the volute at the rated condition of the pump is assumed as steady flow;



- the liquid flow at the inlet of the volute is axially symmetrical i.e. flow rate throughout any section of the volute is proportional to angle of center φ ;
- kinetic energy is transformed into potential energy due to the change of the circumferential velocity V_u (radial velocity V_r may be neglected as its value is low relative to the circumferential velocity).

The flow in the volute is determined by constant velocity moment $V_u r = const$ and proportionality of its flow rate to angle of section: $Q_\varphi = Q \frac{\varphi}{360^\circ}$ [1,2].

The principle of the geometry designing of the volute consists of uniformly reducing of its capacity from the design section to the "zero" (section through the cut water of the volute) [2]. The law of circumferential velocity distribution in such volute is parabolic rather than linear, as it is treated in the calculations.

Following [3], two approaches to the designing of the volute geometry are considered: the friction in the volute is not taken into account and it is taken into account. In the equation of the slope angle of the volute, which describes its geometry, the correction for liquid friction against the surface of the volute channel is applied. And upon that we have made an assumption that the additional friction surfaces affect the flow as well as completely closed shell. However, the losses in the volute are characterized not only by the wall friction of the liquid but local losses as well (smooth enlargement in section of the outlet, flow separation and vortex formation, etc.) which are not included in the present calculation.

In the work [4] the flow in the volute is treated as plane flow induced by straight vortex source with infinite length. As the result of this assumption we have received equation for flow line in the polar coordinates, which has a form of the logarithmic spiral and is treated as the wall outlines of the volute. And two approaches for volute designing is considered: the method according which the velocity moment is constant and the method consisting of determining of mean velocity of the flow in the volute. The present calculations do not show the pattern of the velocity distribution for viscous flow in the outlet channel as they are based on the flow of non viscous liquid.

Flow area of the volute chamber designed for models of non viscous liquid will differ from that designed for models of viscous flow. Subsequently, the volume of losses will be different in these volutes. That is why to calculate and study the volutes taking into account the influence of viscosity and determination of decreasing of loss volumes while changing the non viscous liquid into viscous liquid is still a topical problem. It is considered in the present work.

2. Theory of calculation and designing of volutes

The liquid flow in the volute is treated as spiral flow. The flow velocity in the circumferential direction is V_u and in radial direction is V_m . The volute shall provide the axisymmetric liquid flow at the inlet of volute which means that the inlet angle of flow entering the volute α_3 in circumferential direction shall be constant and equal angle [5,7] (Figure 1):

$$\operatorname{tg} \alpha_3 = \frac{V_{m3}}{V_{u3}} \quad (1)$$

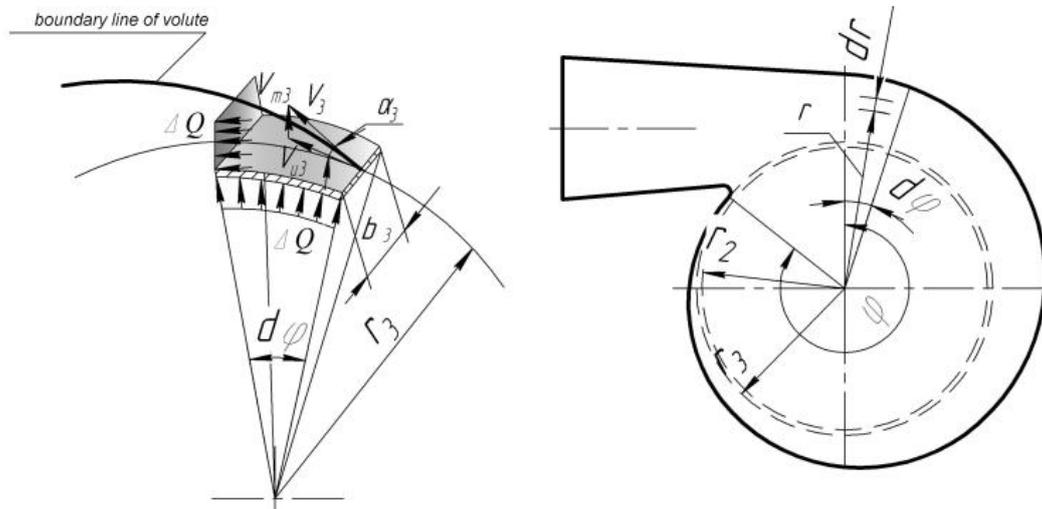


Figure 1. Kinematic parameters of liquid flow at the inlet of the pump volute.

In the first approximation the liquid is assumed as non viscous. Determining the geometry of the volute instead of the traditional approach consisted in building of a finite and therefore intermediate sections we determine the line of volute assuming it as flow line of liquid particle which starts near the bottom of the volute cut water. So we can write the differential equation of the flow line in the form of:

$$\frac{rd\varphi}{V_u} = \frac{dr}{V_r} \quad (2)$$

The circumferential velocity of non viscous liquid flow follows the law of potential vortex $V_u r = K = const$ [2]. The radial velocity is determined by the flow rate equation. For the radial velocity following figure 1, we can write:

$$dQ = V_{r3} b r_3 d\varphi = V_r b r d\varphi \quad (3)$$

Taking into account the forgoing, we obtain from the equation (2) the dependency for building of the flow line:

$$dr = \frac{r}{bK} dQ \quad (4)$$

Equation (4) determines the value of volute line ascent in the circumferential direction which corresponds to the angle of bend φ (Figure 2 (a)).

For outlets with eccentric volute (Figure 2 (b)) the equation takes the form:

$$dl = \frac{r}{bK} dQ \quad (5)$$

where

dl – element of the flow line length of meridian section of the volute.

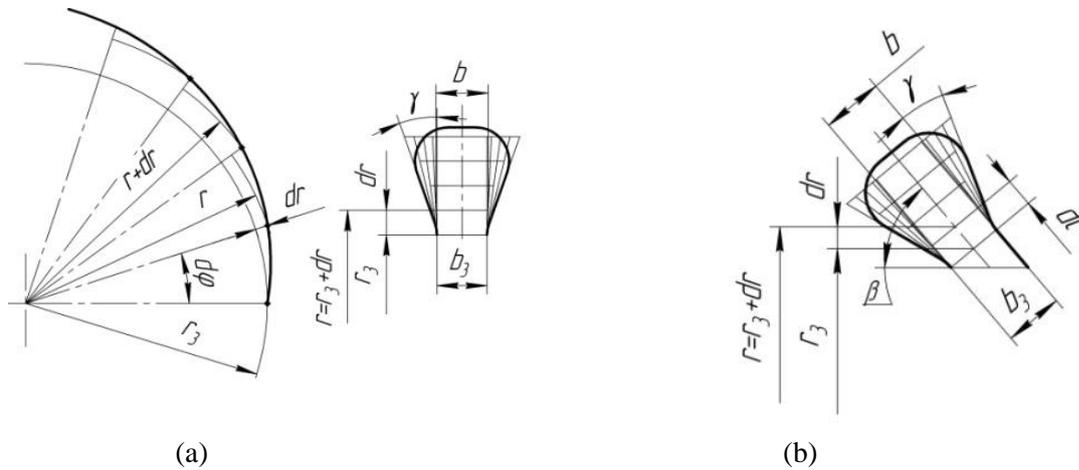


Figure 2. Calculation model of volute: (a) construction of the volute with coaxial cross-section; (b) construction of the volute with eccentric cross-section.

We shall consider the flow of viscous liquid in the volute chamber of the pump.

The motion of liquid particle can be conceived of as a group of stream tubes flowing into volute and moving along spiral flow lines (Figure 3).

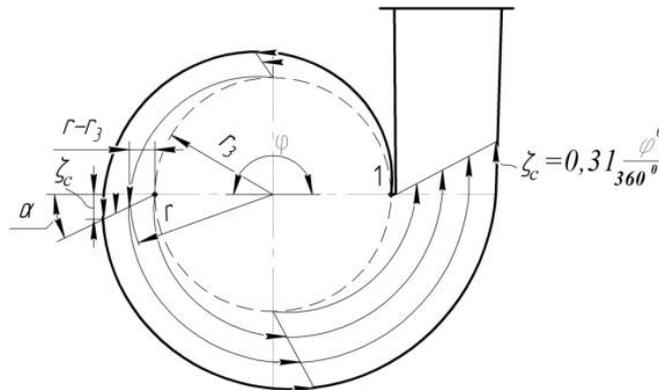


Figure 3. Liquid flow in the pump volute.

For inlet of volute, section 1 and optional section at the angle φ we write Bernoulli equation [6]:

$$z_1 + \frac{P_1}{\gamma} + \frac{V_1^2}{2g} = z + \frac{P}{\gamma} + \frac{V^2}{2g} + h_{1-\varphi} \tag{6}$$

where

$h_{1-\varphi}$ - energy losses along the stream tubes;

$z_1 = z = 0$ - energy of position of liquid particles is neglected.

Energy losses $h_{1-\varphi}$ are assumed as a group of local losses h_M including bend, flow expansion and other as well as losses due to the length h_D :

$$h_{1-\varphi} = h_M + h_D = (\xi_M + \xi_D) \cdot \frac{V^2}{2g} = \xi_c \frac{V^2}{2g} \tag{7}$$

where

ξ_M , ξ_D and $\xi_c = \xi_M + \xi_D$ are coefficients of resistance.

We replace the stream tubes in the form of spiral lines with the stream tubes in the form of concentric circles. The coefficient of resistance is calculated by Welsbach formula [8], obtained empirically:

$$\xi_c = \left[0.31 + 0.16 \left(\frac{d}{r} \right)^{3.5} \right] \cdot \frac{\varphi^0}{90^0} \quad (8)$$

where

d - pipe diameter;

r - bending radius of pipe;

φ - bend angle of pipe.

According to equation (8), the coefficients of resistance of the pipe for stream tube with $d \ll r$ equals: $\xi_c = 0.31 \frac{\varphi^0}{90^0}$.

As a first approximation we assume the law of variation of the coefficient of the resistance along the section of volute as linear (see Figure 3):

$$\xi_c = \frac{\text{tg} \alpha (r - r_3)}{r_0} \quad (9)$$

where

$r_0 = 1$ - a single linear size.

Then calculating the equation (6) with provision for (9), we obtain the dependence of the flow circumferential velocity in the volute on radius:

$$V_u \cdot \left[(a + b \cdot r)^{\frac{a-2}{2a}} \cdot r^{\frac{1}{a}} \right] = \text{const} \quad (10)$$

where

$b = \text{tg} \alpha$; $a = 1 - b \cdot r_3$ - are coefficients.

From equation (10) it follows that when the viscous liquid flows through the volute chamber the velocity moment is lower than the velocity moment of non viscous liquid, flowing through the volute, it can be explained by energy losses. It means that the sections of volute calculated for viscous liquid by equation (10) shall be large than those calculated for non viscous liquid by the condition $V_u r = \text{const}$.

3. Research methods

The check of calculation and designing of volute were performed with the testing torque flow pump 125/28 with eccentric volute [9]. It is overhung pump designed for $Q = 125 \text{ m}^3/\text{h}$, $H = 28 \text{ m}$. The hydraulic flow part is made according to the method [12]. It is shown in Figure 4.

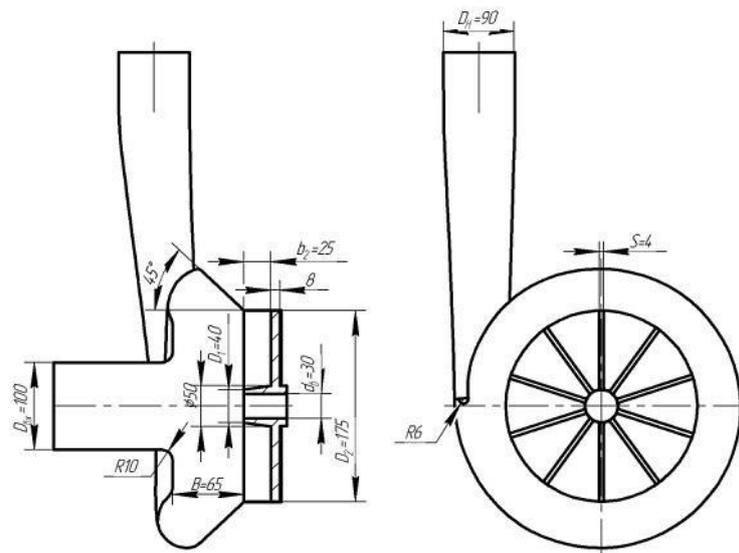


Figure 4. Testing torque flow pump 125-28.

The researches were conducted by means of numeric modeling and physical experimentation.

We used ANSYS CFX version for university to carry out numerical modeling of the liquid flow in the hydraulic flow part of the examined variants of the torque flow pumps. While modeling the operation conditions of the torque flow pump we applied computational domain consisting of pump casing including volute chamber and impeller. Computational mesh of impeller had about 1.63 millions prismatic cells, pump casing – 1.86 millions of cells. Prior to research mesh the independence check was made for each element of the computational domain. The value of the variable Y^+ was in the range from 1 to 20 units, it was complied with the recommendations given in the user manual [10].

Typical Menter SST model [11] was used for turbulence modeling.

Numerical calculations were carried out in nonstationary problem. The outlet boundary of the calculation domain was located at 6 diameters away from the inlet boundary of diffuser nozzle. As boundary condition at outlet of calculation domain we set static pressure. As we assumed the presence of back flow at the outlet of the calculation domain the type of the boundary condition was set as "opening".

To conduct this research we designed two variants of casing of the torque flow pump with dimensioned volute according [13] (Figure 5):

variant No.1 – the absent of losses in the volute, according equation (5);

variant No.2 – the losses in the volute channel is taken into account, according equation (10).

The calculation showed that the finite section of the volute No.2 was higher by $\Delta = 7 \div 10\%$ than the finite section of the volute No.1 due to viscous.

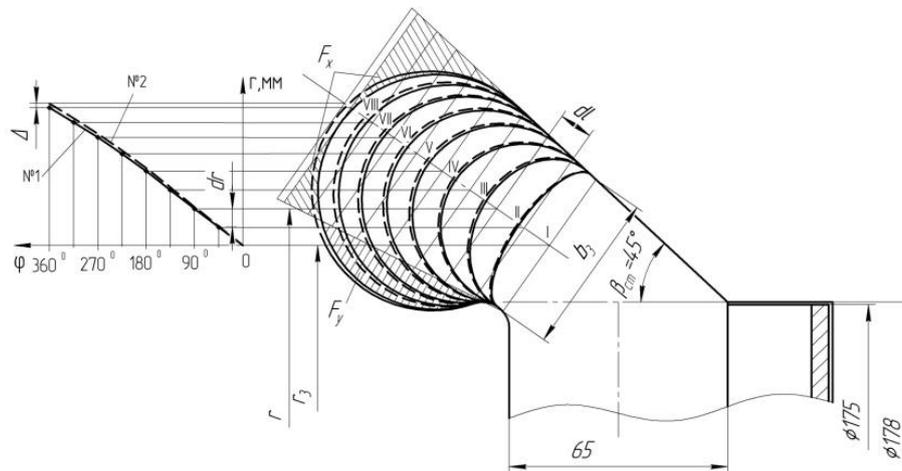


Figure 5. The calculation of casing of torque flow pump 125-28 with eccentric volute.

4. The study results of the volute geometry of the torque flow pump

By means of the numerical study we received a head-capacity and power input curves for volutes of the torque flow pumps (Figure 6). The curves for model hydraulic flow parts were calculated in a capacity range from $0.5 Q_{opt}$ to $1.25 Q_{opt}$.

According to the research results the optimal operating range of the pump with volute No.1 with the highest efficiency 46.4% shifted from the rated capacities towards the area of smaller capacities with the value of $Q=120 \text{ m}^3/\text{h}$ with the head $H= 31.1\text{m}$. The optimal operating range of the pump with volute No.2 corresponds to the rated operating range with efficiency 48.6%: $Q=125 \text{ m}^3/\text{h}$, $H= 30.9\text{m}$. Flow area of the volute No.2 is greater than the flow area of the volute No.1. The optimal operating range of the pump with volute No.2 corresponds to the rated operating range at the lower losses in the volute than the losses in the volute No.2 (Figure 7).

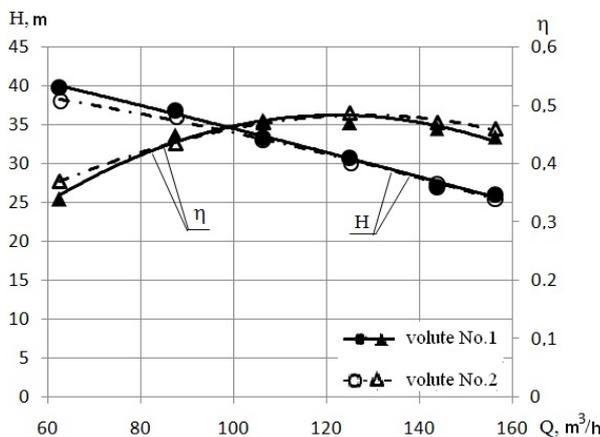


Figure 6. Head-capacity and power input curves for casings of the torque flow pumps.

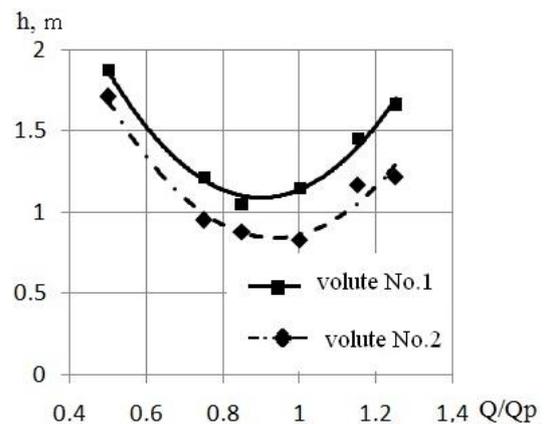


Figure 7. Distribution of the losses in the volutes of the torque flow pumps.

The shift of the optimal operating range in the volute No.1 with decreased flow areas and increased losses can be explained by axially symmetrical distortion of the flow at the inlet of the volute chamber. The volute No.2 is characterized by minimal losses at the design point. Therefore, taking into account the viscous of the liquid during the designing of the volute leads to the axially symmetric flow at the inlet of the volute chamber providing the steady-state flow near the pump impeller thus reducing the hydraulic losses. According to the research results (see Figure 6) efficiency of the pump with volute No.2 calculated with an allowance for viscosity exceeds by 1-2% the efficiency of the pump with

volute No.1 calculated for non-viscous liquid. A good example of ensuring axial symmetry of the liquid flow at the inlet of the volute is the dependence of the flow rate $\frac{Q}{K_2}$ of the volute chamber on the contact angle φ (Figure 8).

The account of the viscosity in the calculation of the volute leads to a linear dependency $\frac{Q}{K_2}(\varphi)$, and hence it provides axially symmetric flow at the inlet of the volute chamber. The reducing of the velocity moment K_2 in the volute No.1 below the calculated value leads to the shift of the rated conditions towards the area of smaller capacities. Volute No.1 is characterized by deviations of the analytical and calculation values of 11%, for volute No.2 – 3.5%.

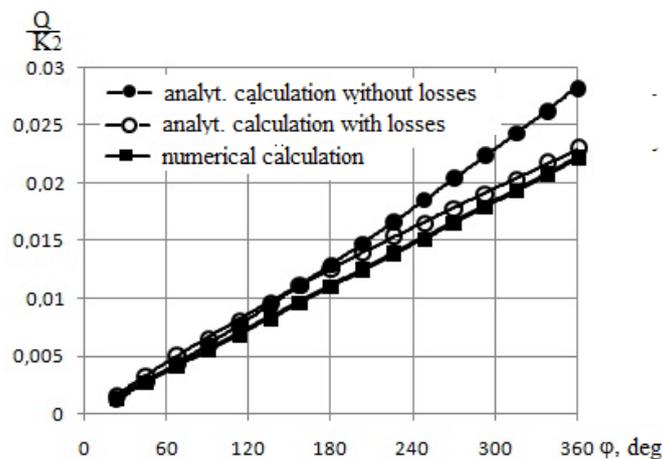


Figure 8. Flow rate of the volute of the torque flow pumps.

In order to confirm the results of the numerical calculation we tested the volute No.2. The testing results are presented by means of head-capacity and power input curves (Figure 9).

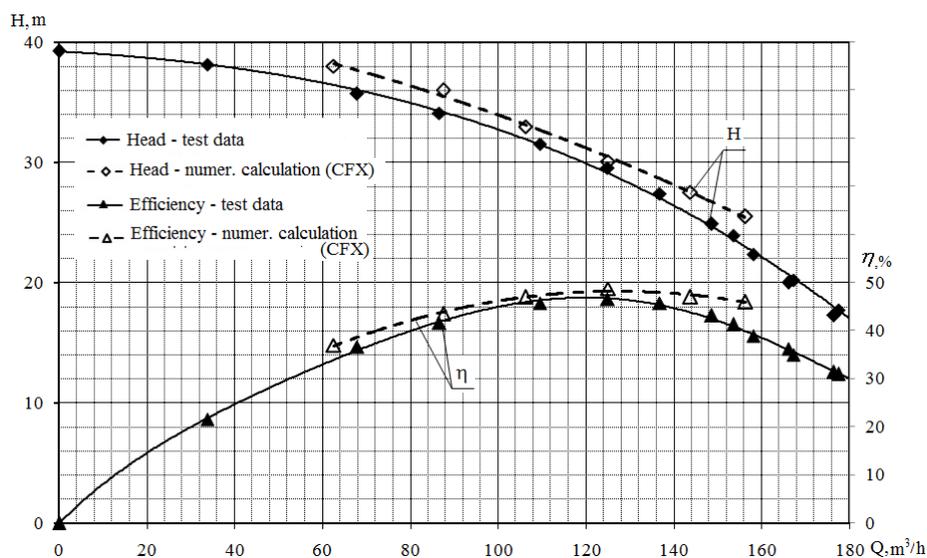


Figure 9. Comparative characteristics of the torque flow pump with dimensioned volute received after numerical calculations and tests.

According to the test results the optimal operating range of the torque flow pump with dimensioned volute No.2 corresponds to its calculated operating range with efficiency 47.8% and head $H = 28\text{m}$. The comparison of the characteristic curves obtained by testing with the results of numerical calculation shows that the head discrepancy is not more than 3% and efficiency discrepancy is about 2%. The most likely cause of such discrepancy of characteristic curves of the torque flow pump is the lack of information about the operation of the torque flow pump.

5. Conclusions

The method for calculating of volute based on the differential equation is presented.

The usage of one-dimensional model for viscous liquid flow allowed to specify the law of circumferential velocity distribution and therefore the velocity moment along the section of the volute.

The volute calculated on the assumption that the velocity moment is constant along the section of the volute chamber $V_{\omega}r = \text{const}$ leads to the shift of the optimal operating range from the calculated value towards the area of smaller capacities. The discrepancy of the velocity moment obtained by numerical calculation of the viscous liquid problem and non-viscous problem for such volute is 11%. The velocity moment discrepancy obtained by calculation performed by proposed dependency with account for viscous allows reduce the velocity moment discrepancy up to 3%.

According to the numerical calculations the volutes of the torque flow pumps calculated by the method presented in this work improved the efficiency of the pump up to 1-1.5%.

References

- [1] Miftahov A A and Zyikov V I 1996 Inlets and outlets of centrifugal compressors (Kazan: Feng (Science)) p 198
- [2] Mashin A N 1980 Calculation and designing of volute outlet and semi-volute inlet of centrifugal pump (Moscow: MPEI Publ.) p 44
- [3] Pfeleiderer K 1960 *Die Kreiselpumpen für Flüssigkeiten und Gase. Wasserpumpen, Ventilatoren, Turbogebläse, Turbokompressoren* (Moscow: GNTI mashinostroitelnoy literatury) p 683
- [4] Kostornoy S D and Chaplygin A A Designing of centrifugal pump volutes of trapezoidal shape with rounded corners *Bulletin of Sumy State University. Series: Engineering* **1** (3) pp 105-115
- [5] Gulich J F 2010 Centrifugal Pumps (second edition) *Springer Heidelberg Dordrecht* (London, New York) p 957
- [6] Barlit V V 1977 *Gidravlicheskie turbiny Hydraulic turbine (Vyshcha shkola)* p 360
- [7] Rudnev S S 1967 Calculation of volute chambers of centrifugal pumps (Moscow)
- [8] Frenkel N Z 1956 *Hydraulics State Energy Publishing* (Mostka)
- [9] Krishtop I V 2015 Designing of hydraulic part of torque flow pump with updated energy efficiency *Eastern-European Journal of enterprise technologies* **2/7(74)** pp 31-37
- [10] ANSYS CFX 10.0 Solver Models. Release 10.0 [Digital source] –Available at: \ www/ URL: <http://ansys.com/> - 2008 p 549
- [11] Krishtop I, German V, Gusak O, Lugova S and Kochevsky A 2014 Numerical Approach for Simulation of Fluid Flow in Torque Flow Pumps *Trans Tech Publicatoinis Inc.: "Applied Mechanics and Materials"* **440** pp 43-51
- [12] German V F, Kovalev I A, Kotenko A I and Gusak A G 2013 Torque flow pumps (Sumy: Sumy State University) p 159