Modernization of centrifugal impeller blades

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Abstract. The increase of the energy intensity of the pumping equipment reflects general tendency of technology development: to transmit more energy without significant increase of mass and size parameters of their working bodies. It is represented in the “Directive on Determining the Requirements for the Design of Products Consuming Electricity” that forwards a claim for a 40% reduction of energy consumption in Europe by 2020. Considering the task as for the increase of energy intensity of the pump stage, the possibility of creating a higher head and efficiency increase with the same overall dimensions, flow rates and speed of rotation has been considered as a way of solving the task. This can be achieved by replacing the spatial blade system of a centrifugal impeller of low (n_s = 65) specific speed by cylindrical blades, since such working bodies perform the process of energy transfer from impeller to working liquid in the most intensive way. It will provide for the head rise of impeller and for the simplification of its manufacturing technology.

1. Introduction

To date, the industry faces an urgent problem - high energy intensity during pumping equipment operation. Thus, in various industries pumps consume at about from 25 to 60% of all expended energy. In this case, almost ¾ of energy loss is accounted for by dynamic pumps. In this regard, in 2005 the European Parliament adopted “Directive on Determining the Requirements for the Design of Products Consuming Electricity” which provides for a 20% reduction of total energy consumption in Europe by 2020, and energy consumption by pumping equipment up to 40% [1-5]. Analysis of pump unit life cycle costs [6] shows that the major part is the cost of energy. At the same time the self-cost of the pump as a technical system, and, respectively, the cost of its subsystems is a significant component in the life cycle costs.

In general, considering this issue, we can note that nowadays many industries have a problem concerning the economical efficiency of centrifugal pumps, which provide for the head rise at relatively low flow rates [7]. The solution of such problems should be found in increasing the intensity of energy transfer in the system “pump working body – pumped liquid”. The implementation of this idea can lead to the possibility of usage of a single-stage centrifugal pump instead of multi-stage one or to significant reduction in the numbers of stages of the latter one. Therefore, there is an urgent issue concerning finding of ways to modernize the geometric parameters of the blade grid of the centrifugal pump stage, which would lead to the increase of its integral characteristic, and first of all, of head and efficiency. It should also be noted that the problem of reducing the pump mass and size parameters would be considered as well. In general, the increase of the energy intensity of the
pumping equipment reflects general tendency of technology development: to transmit more energy without significant increase of mass and size parameters of their working bodies.

Thus, the pumps of reservoir pressure maintenance, that are multistage centrifugal pumps of CNS type, play a rather significant role in the oil industry. The need to increase the head of a particular stage of such pumps has been caused by conditions in the oil industry, namely, the reduction of crude oil production level from individual fields. The intensification of oil production on the final stages of development is realized by increasing the reservoir pressure that requires the use of pumping equipment, which provides for the head rise in the network at a steady flow rate. Similar problems are also present in the coal industry. The development of more deep coal seams leads to the need of head rising of pumps that extract groundwater from coalmines. The urgency of rising the head of the pump stage confirms the growing demand of consumers for the modernization and development of pumps with improved characteristics of the head. Therefore, the study of increasing the head of centrifugal pump stage based on a systematic approach, provided for by the modern CAD system capabilities, is relevant and corresponds to the concept of European Pump Manufacturers Organization EuroPump and development of pump building in Ukraine, as well as reflects the general trend in the development of mechanisms and machines.

Regarding the prospects of centrifugal pumps, it can be noted, that the use of piston-, screw-, vortex-type, etc. or any other pump type to solve the aforementioned problems, is neither desirable for reasons of economy, nor impossible at all because of certain features. In addition, it is well known that the blade type pumps have a number of advantages (mass and size parameters, operation and reliability) in comparison with the pumps of other types used in neighbouring fields.

2. Analysis of literature data and target setting

Despite the fact, that the working process of the centrifugal blade grid in the pumping mode is already examined in depth and thoroughly described in many scientific studies, the researches concerning the ways of influencing its head, power input and efficiency characteristic curves are continued. In some cases of pumps operation there appears a necessity to change their parameters, that is caused by the tasks and problems faced by enterprises, in particular in the oil and gas industry that use pumping equipment. As a rule, it concerns to the reduce of the flow rate or head values with a decrease in the debit of oil wells that can be solved by partial or complete replacement of the pump flow passage and does not cause any special difficulties. However, in some cases it is necessary to increase the pump head at a constant flow rate. In this case, the task is significantly complicated, since when replacing the flow passage, it is supposed to be interchangeable, and therefore, the rise of the stage head by increasing the diameter of impeller is unacceptable.

Impeller is the main element of the pump and determines its design and parameters in a great measure. Impellers can be classified by the specific speed \( n_s \), which characterizes the efficiency, the geometry of flow passage, the ratio of geometric parameters (Fig. 1 a) and pump performance test curve format (Fig. 1 b) [8]. In the context of the issue under consideration, the particular attention is paid to the low specific speed impellers. They are the most high-head and therefore, in our view, their operation process should act as the object of study. The subject of the study is the influence of the proposed changes in the geometric parameters of the impeller blade grid on its head-capacity, power input and efficiency characteristic curves at a constant external dimensions.

Special feature of the low specific speed impellers is that the \( b/2/D_2 \) ratio is very small for them. In such case, the impellers are thin and the ratio \( D_2/D_0 \) is quite significant. In this regard it can be assumed that the location of the entrance edge of the spatial blade and the flow deflection in the low specific speed impeller has a much smaller impact on head-capacity, power input and efficiency characteristic curves of impeller and pump as a whole, than in the impellers of higher speed [9].

Based on the above mentioned, it is possible to formulate a scientific and technical problem, which consists in the possibility of using cylindrical blades instead of spatial ones in low specific speed impellers, as well as of identifying the effect of such replacement on head-capacity, power input and efficiency characteristic curves of impeller and of stage as a whole.

The main and most widespread method of centrifugal impeller manufacture used in pump building
up to now is a casting process. At the same time, cast impellers have large mass and low level of surface quality. The manufacture of spatial blade is of the greatest difficulty [10].

The use of cylindrical blades in low specific speed impellers will simplify the process of their manufacturing and increase manufacturability [11, 12]. For the manufacture of impellers for impeller machines, it is possible to use method of precision forging, which is characterised by low complexity, high coefficient of metal use and unmachined surface as well as surface quality level [10].

3. Research purposes and objectives
Based on the above stated, the study purpose can be formulated as follows: revealing the ways for optimization of low specific speed impeller blade grid without lost of economical efficiency.
To achieve this goal, the following tasks have been formulated:
1. To study the possibility of spatial blade replacement by cylindrical one;
2. To find out the optimal variant of new blade grid;
3. To study the characteristic curves of stage with modernized impeller.

4. Research methods
Numerical studies have been performed based on intermediate stage impeller of the pump CNS 180-1900 with the following parameters: \( z = 8, n_s = 65, Q_{nom} = 180 \text{ m}^3/\text{h}, n = 3000 \text{ rpm} \) at \( Q/Q_{nom} = 0.7; 0.9; 1; 1.1; 1.2 \). The ANSYS CFX software of university licenced version was used. The basis of this software is the method of numerical solution of fundamental laws of hydromechanics [13, 14]: viscous fluid flow equation jointly with continuity equation. This is a sufficient condition for the validity of...
It should be noted that ANSYS CFX has been repeatedly tested in solving tasks of pump building; discrepancy between the results of numerical and physical modelling does not exceed 5%. Therefore, this software is applicable for solving the task of the study.

The flow was calculated by numerical solution of combined equations describing the most common case of a liquid medium motion - the Navier-Stokes equation (1) and the continuity equation (2) [15]. Equations are presented in abbreviated form (i, j - 1-3), it is assumed that the summing up of the same indices is performed.

\[
\frac{\partial}{\partial t}(\rho u_i) + \frac{\partial}{\partial x_j}(\rho u_i u_j) = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j}\left[\mu\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right)\right] + f_i, \tag{1}
\]

\[
\frac{\partial p}{\partial t} + \frac{\partial}{\partial x_j}(\rho u_j) = 0, \tag{2}
\]

where \(x_i, x_j\) - coordinate axis;

\(f_i\) - component that expresses the action of mass forces.

This system comprises four equations, and the independent parameters sought are three components of velocity \(u_1, u_2, u_3\) and the pressure \(p\). The density \(\rho\) of liquid is assumed as constant. The flows in rotating working bodies of hydromachines are considered in the incremental measuring system. Element \(f_i\) in the right side of equation (1) expresses the action of centrifugal and Coriolis forces:

\[
\vec{f}_i = -\rho \left(2\vec{\omega} \cdot \vec{u} + \vec{\omega} \cdot (\vec{\omega} \cdot \vec{r})\right),
\]

where \(\omega\) - angular rotational speed, \(1/\ell\);

\(r\) - radius vector the modulus of which is equal to the distance from a given point to the axis of rotation, \(m\).

As boundary conditions, there is set a “non-slip” condition at the solid walls (velocity equals to zero), the distribution of all velocity components at the inlet section, and the equality to zero of first derivatives (in the direction of the flow) of the velocity components in the outlet section.

The flow in the flow passage of hydraulic machines as a rule is turbulent. Its direct modelling by numerical solution of Navier-Stokes equation, recorded for instantaneous velocities, is an extremely difficult task. In addition, the values of time-averaged velocity, as compared to instantaneous ones, are of interest. Therefore, the simulation of turbulent flows was performed using Reynolds equation (3) instead of equation (1):

\[
\frac{\partial}{\partial t}(\rho \overline{u}_i) + \frac{\partial}{\partial x_j}(\rho \overline{u}_i \overline{u}_j) = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j}\left[\mu\left(\frac{\partial \overline{u}_i}{\partial x_j} + \frac{\partial \overline{u}_j}{\partial x_i}\right)\right] + f_i \tag{3}
\]

where \(\overline{u}_1, \overline{u}_2, \overline{u}_3\) - time-averaged velocity values;

\(\overline{u}_i, \overline{u}_2, \overline{u}_3\) - pulsation velocity component.

For the closure of the Reynolds equation, there was used the \(k-\epsilon\) turbulence model. When using this model, the equation system of fluid motion is supplemented by two differential equations that, respectively, describe the transfer of kinetic energy of turbulence \(k\) and the dissipation rate \(\epsilon\) (2), (3).
\[
\frac{\partial}{\partial t} (\rho k) + \frac{\partial}{\partial x_j} (\rho \bar{u}_j k) = \frac{\partial}{\partial x_j} \left( \Gamma_k \frac{\partial k}{\partial x_j} \right) + P_k - \rho \varepsilon ,
\]

(4)

\[
\frac{\partial}{\partial t} (\rho \varepsilon) + \frac{\partial}{\partial x_j} (\rho \bar{u}_j \varepsilon) = \frac{\partial}{\partial x_j} \left( \Gamma_\varepsilon \frac{\partial \varepsilon}{\partial x_j} \right) + \frac{\varepsilon}{k} (C_{\varepsilon 1} P_k - \rho C_{\varepsilon 2} \varepsilon),
\]

(5)

where \( P_k = -\rho \bar{u}_j \frac{\partial \bar{u}_j}{\partial x_j} \) – element that expresses the energy generation \( k \).

\[
\Gamma_k = \mu + \frac{\mu_t}{\sigma_k}, \quad \Gamma_\varepsilon = \mu + \frac{\mu_t}{\sigma_\varepsilon}
\]

Parameters \( \varepsilon \) and \( \mu_t \) are determined as follows:

\[
\varepsilon = \frac{\mu}{\rho} \left( \frac{\partial \bar{u}_i}{\partial x_j} \right)^2, \quad \mu_t = \rho C_{\mu} \frac{k^2}{\varepsilon}
\]

When conducting a numerical study, the following assumptions were made:
- the flow at the inlet to the computational region is axially symmetric;
- the effect of leakages through the impeller seals on the flow in the flow passage is absent.

Numerical simulation within the framework of studies comprised several stages: preparation of liquid 3-D geometric model, construction of computational mesh, entry of input data for calculation and calculations itself. At the first stage, the implementation of which was carried out in SolidWorks software, there have been created geometric models that simulated the liquid volume in channels of flow passage of studied stage and impeller (Fig. 2).

Figure 2. 3-D model of liquid in impeller with spatial blades.

After construction of liquid models, the computational meshes were created by means of ICEM CFD. It allows to adjust the mesh depth by thickening it in the necessary places (for example, at the entrance and outlet exit of blades) and by increasing it in places where the height depth of mesh is not required. This approach allows to save resources of ECM and to obtain sufficient mesh depth in the studied part of computational region.
The value of variable $y^+$, which characterizes the mesh thickening near the walls, was within $20 < y^+ < 100$ units, that corresponds to the recommendation given in the user’s manual [15].

After generation of meshes, the next step of calculations was the creation of a computational region in CFX-Pre. The boundary of entrance into impeller was selected as the limit of entrance into the computational region. The boundary of exit from it was determined by the exit from the impeller and was also located at a distance of one diameter of the entrance to it. Static pressure was determined as a boundary condition at the exit from the operating region. Based on the fact, that all the further study and comparisons were carried out for relative values, the absolute value of pressure was of no importance and was assumed equal to $P_{exit} = 10$ MPa. Type of the boundary condition was determined as “opened” based on the assumption of the presence of reverse flows at the exit from operating region.

By adhering to the “no-slip” conditions, the condition of equality of velocity to zero was set for all impeller walls. Walls roughness was determined as $Ra3.2$. Mass flow rate at the exit was determined as the boundary condition.

Review, processing and analysis of results was carried out by means of CFX-Post software, which has wide opportunities for visualization and evaluation of calculated flow rate characteristics. The result of numerical calculations was to obtain instantaneous velocity and pressure values in each cell of the computational mesh. Mass flow rate averaging was performed in order to determine the integral values.

5. Research results

5.1 Centrifugal impellers with different blade types

Taking impeller with spatial blades to be the base one, we designated it as PK No. 1, and impeller with cylindrical blades we designate as PK No. 2. The studies have been also performed for impeller with cylindrical blades that were removed from the flow deflection zone in impeller (PK No. 3), as well as with blades elongated and turned against the direction of rotation (PK No. 4). Diameter of the blades exit edge setting for schemes PK No. 3 and PK No. 4 is determined by the ratio $D_{set}/D_2 = 0.59$. In this case the angles $\beta_2 = 27^\circ$ of blade setting at the exit of impeller for all schemes of meshes remained unchanged. Figure 3 a to d presents the blade grids of studied impeller schemes PK No. 1 to PK No. 4 respectively.

Distribution of relative velocities for schemes PK No. 1 to PK No. 4 is shown on Fig. 4. Calculation results are presented on Figure 7, changes of such parameters as $H$, $N$, efficiency relative to the base impeller at $Q/Q_{nom} = 1$ are presented in Table 1.

| PK scheme | $\Delta H$, % | $\Delta N$, % | $\Delta \eta$, % |
|-----------|--------------|--------------|-----------------|
| No. 2     | 8.2          | 8.9          | -1.2            |
| No. 3     | 2.7          | 4.8          | -2.5            |
| No. 4     | 6.8          | 6.3          | 0               |

As shown on Figure 4, when using the blade grid, removed from the flow deflection zone in the impeller (PK No. 3), there is observed a significant vortex formation region on the back side of blade. It can be explained by the fact, that the blades of the specified schemes had far too big angle of blade setting at the exit, which evidently exceeded the recommended values ($\beta_1 = 33^\circ$) [8]. Changes in the design of this blade grid were aimed at the matching of the blade position with the direction of flow in impeller after flow deflection zone by removing the vortex formation region (PK No. 4). In this scheme $\beta_1 = 18^\circ$. 

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Figure 3. Spatial (a) and cylindrical (b to d) blade grids.

Figure 4. Distribution of relative velocities.
Analysing the obtained data, we can conclude that, generally, the use of cylindrical blade led to the head rise of impeller. This can be explained by the fact, that the force effect of the blade system and liquid flow are better, and the energy transfer from the impeller to liquid increases. In this case, there is observed a certain decrease in the impeller efficiency, caused by an increase of hydraulic losses due to a change of blade shape.

What calls attention to itself is the fact that the efficiency of impeller of scheme PK No. 4 remained unchanged compared to the base impeller with spatial blades. Based on the combination of changes in the head and efficiency parameters, scheme No. 4 can be called as an optimal one.

It should be also noted, that the simplification (“folding”) of design of machine working body or its element (in our case, it is the impeller of centrifugal pump and its blades) while maintaining or increasing its main useful function (head rise) is fully consistent with the laws of technical development systems [16-18].

Natural development of the idea concerning the simplification of impeller blade grid was the challenging idea to study the possibility and feasibility of using plate-like blades in the low specific speed impeller (Fig. 5). Plate-like blade was set perpendicularly to the radius $R_2$ and based on ratio $D_{set}/D_2 = 0.59$. In this case angle $\beta_1$ actually was equal to zero, аnd $\beta_2 = 52^\circ$.

![Figure 5](image1)

**Figure 5.** 3-D model of liquid in impeller with plate-like blades.

Distribution of relative velocities for scheme PK No. 5 is shown on Fig. 6. Calculation results are presented of Fig. 7, changes of such parameters as H, N, efficiency relative to the base impeller at $Q/Q_{nom} = 1$ are presented in Table 2.

![Figure 6](image2)

**Figure 6.** Distribution of relative velocities.

**Table 2.** H, N, efficiency relative to the base impeller.

| PK scheme | $\Delta H$, % | $\Delta N$, % | $\Delta \eta$, % |
|-----------|---------------|---------------|-----------------|
| No. 5     | 30.5          | 27.2          | 2.5             |
Figure 7. Head-capacity (a), power input (b) and efficiency (c) characteristic curves of studied PKs.
Analysing the obtained data, we can conclude that, generally, the use of plate-like blade led to the significant increase of head of impeller. Such effect was predictable as far as this blade grid had increased angle $\beta_2$. The results of efficiency are of particular interest. Its value for scheme under consideration is greater than for the base impeller. It can be explained by the fact, that the coordination of flow rate and blade setting primarily occurs at the enter.

In general, it should be noted, that the results concerning exclusively the impellers do not reflect the picture of our interest, as far as impellers work as a part of the pump stage. Therefore, a logical continuation of this study will be the research of impellers with modernized grids along with the base diffuser.

5.2. Study of modernized impellers as part of the pump stage

The calculations were made for the following schemes. Intermediate stage of the pump CNS 180-1900 with impeller of scheme PK No. 1 and diffuser to it was defined as the base one (CT No. 1). Impeller of scheme PK No. 4 with cylindrical blades was studied in the scheme CT No. 4, and impeller with blade grid of plate-like form (PK No. 5) was studied in the scheme CT No. 5. It should be noted, that the diffuser was unchangeable for all schemes. Figure 8 presents liquid model of the base pump stage (scheme CT No. 1).

Research in the software application ANSYS CFX was performed according to the same methodology as for the separate impellers. The results of the numerical calculation of the base stage were compared with the results of full-scale study of the specified stage performed on the test stand of JSC “VNIIAEN” (Sumy).

![Figure 8. 3-D model of liquid in base stage.](image)

Calculation results compared with the results of full-scale study of base stage are presented on Figure 9. Changes of such parameters as $H$, $N$, efficiency relative to the base impeller at $Q/Q_{nom} = 1$ are presented in Table 3.

Analysing the integral characteristics obtained at the test stand and comparing them with the results of numerical modelling, we can conclude that the discrepancy in the results, obtained under the working conditions, is less than $\%$ ($\Delta H = -1.9\%, \Delta N = -1.3\%, \Delta \eta = -0.6\%$). In this case, we can say that the obtained discrepancy is acceptable, and the results of numerical experiments can be consider as true. Data obtained as a result of numerical experiments have the necessary accuracy and we may be governed by them without resorting each time to verification at the test stand.

| CT scheme | $\Delta H$, % | $\Delta N$, % | $\Delta \eta$, % |
|-----------|---------------|---------------|------------------|
| No. 4     | -0.9          | -0.6          | 0.0              |
| No. 5     | 14.6          | 22.6          | -6.3             |

Table 3. $H$, $N$, efficiency relative to the base impeller.
Figure 9. Head-capacity (a), power input (b) and efficiency (c) characteristic curves of studied stages.
As to the fact of characteristics difference, it can be assumed, that the most probable reason for the indicated behaviour of characteristic curves is the discrepancy between the geometric model of the blade systems used in the calculations and actually manufactured ones. This is especially concerns to the the shape of enter and exit edges, roughness of flow part, etc.

Analysing the results of study of impeller operation with the proposed blade grids being the part of stage with an unchanged basic diffuser, we can note the following. Concerning the scheme CT No. 4: the obtained values of integral characteristics under design conditions almost do not differ from the parameters of the base stage. Concerning the scheme CT No. 5: in contrast to the particular impeller (scheme PK No. 5) there has been found out that, the head rise is almost three times less and efficiency is decreased.

In such a way we see, that the use of low specific speed impeller with cylindrical blades removed from the flow deflection zone and coordinated with it (schemes PK No. 4 and CT No. 4) is quite justified on the grounds of providing the stage of necessary head-capacity, power input and efficiency characteristic curves. The issue regarding the use of blade grid with further simplifications of the blade (schemes PK No. 5 and CT No. 5) remains open due to a not very significant rise of head along with the distinctive decrease of stage efficiency. In general, we believe that the further studies in this direction are promising. To find out a truly optimal blade design, it would be advisable to use the methods of planning a multifactor experiment and experimental verification.

6. Conclusion
1. Because of numerical study, the possibility of replacing spatial blade with a cylindrical one in low specific speed impellers based on assessment of their efficiency has been fundamentally proved. In this case, it was possible to find out the positive effect of head rise of impeller.
2. The optimal variant of the new blade grid, in which the blades are removed from the flow deflection zone, was revealed; the exit edge is located based on the ratio $D_{sec}/D_2 = 0.59$, angle $\beta_1 = 18^\circ$. There has been also proposed the plate-like blade that was placed perpendicularly to the radius $R_2$. The rise of head increase was achieved with a favourable level of efficiency.
3. The study of the stage characteristics with the modernized first impeller showed that the spatial blade grid in the centrifugal low specific speed impellers could be replaced by more simple, in means of manufacture, cylindrical grid without loss of stage economic efficiency. The discrepancy between the experimental and calculated parameters of a base stage with modernized impeller was not more than 2%.
4. The basis of the scientific novelty of the results obtained is that the possibility and expediency of replacing a spatial blade by cylindrical one in centrifugal low specific speed impellers has been revealed for the first time by means of numerical research methods.
5. The main practical significance of the obtained results is that it has been proved that the use of impeller with modernized blade grid as a part of stage with the base diffuser is proved.

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