Investigation of the operating process of a high-pressure centrifugal pump with taking into account of improvement the process of fluid flowing in its flowing part

V Y Kondus¹, O G Gusak¹, J V Yevtushenko¹

¹Department of Applied Hydro- and Aeromechanics, Sumy State University, 2, Ryms’kogo-Korsakova, Sumy 40007, Ukraine

E-mail: vladislav.kondus@meta.ua

Abstract. Centrifugal pumps are used in most industries. In particular, they are effectively used in the technological processes of oil production to create reservoir pressure at oil fields. Due to increasing demand for petroleum products, there is a need to improve such pumps to increase reservoir pressure by increasing their pressure. The research considered the possibility of improving the centrifugal pump impeller design by changing its blade system. It is theoretically possible to increase the pump head by changing the blade outlet angle at the impeller outlet $\beta_2$ to $75^\circ$. The design of additional wedge-shaped blades of the second tier of the impeller was developed to reduce some additional hydraulic losses zones. They made it possible to stabilize the fluid flow in the impeller intervane channels, to reduce additional hydraulic losses. In the research, the impeller head was increased up to 26.7% for the proposed design impeller compared to the standard one. The research was performed with a comprehensive analysis of the pump installation life cycle cost. Thus, the development of new ones and the upgrading of pumping installations rooted in the industry with the aim of increasing the head is possible while achieving the minimum pump installation life cycle cost.

1. Introduction
Centrifugal pumps are used in various industries. Pumps of this type are mainly used for the transportation of pure liquids, liquids with low inclusions content, gas, liquids with low viscosity, etc.

Depending on the required parameters, centrifugal pumps are divided into single-stage and multi-stage pumps.

The design of a flowing portion of a typical centrifugal pump consists of three main elements: a suction nozzle, an impeller and a pressure nozzle. A typical design diagram of a centrifugal multistage pump is shown in Fig. 1 a.

Depending on the required pump parameters, the impellers (Fig. 1b) are divided into centrifugal low specific-speed, centrifugal normal specific-speed, centrifugal high specific-speed, diagonal and axial ones.
Figure 1. Typical scheme of a centrifugal multistage pump (a) and the design of a low specific-speed impeller (b).

The main indicator of the pump head, depending on its flow capacity and rotational frequency is the specific speed. It is determined by the dependence of:

\[ n_s = \frac{3,65n\sqrt{Q}}{H^{3/4}} \]

where \( n \) - rotational frequency of the pump, rpm, \( Q \) - pump flow capacity, \( m^3/s \), \( H \) - pump head, m

Thus, the high head is achieved in low specific-speed centrifugal pumps.

Nowadays, a number of industries require pumping installations meeting the increased pressure requirements of systems. That is, they need to increase the head of the pumping equipment.

In particular centrifugal pumps used in the oil production to create a reservoir pressure by feeding there of process water. At the same time, there is a need to increase the head of pumping installations to increase oil production.

Using of multi-stage pumps with a capacity of 1.5 MW or more is characteristic of this field. Thus, the replacement of pumping equipment with new, high-pressure equipment requires considerable investment costs. In the final case, this significantly increases the total pump installation life cycle cost [1]. Therefore, it is not a rational way of upgrading the pumping equipment.

The urgency of the research is to find ways to increase the centrifugal pumps head. In this case, a comprehensive approach to this issue was taken in view of the total pump installation life cycle cost. This allows to the maximum increase of the centrifugal pump head with the minimum investment cost and the minimum cost of electricity. In practice, this means achieving maximum pump efficiency by minimizing hydraulic losses in its flowing part.

2. Analysis of literature data and statement of research issues

In the paper [2], the influence of the passage channels widths of the flowing part of the low specific-speed pumps on its parameters was investigated. It is established that when their width increases, hydraulic losses in the pump flowing part decrease.

In research [3], increasing the head of the centrifugal pump is achieved by increasing the rotational speed. However, the modernization of implemented into industry pumps requires additional investment to replace the motor or frequency converters. This greatly increases the total pump installation life cycle cost [1], which is not rational for most consumers.

The study [4] considered the increase of the centrifugal pump parameters by reducing the hydraulic losses in its flowing part. It has been determined that the largest hydraulic losses in multistage centrifugal pumps are observed in the guide vanes diffuser channels. Authors proposed the thick vanes construction in the diffuser channels of guide vanes. It reduced hydraulic losses up to 8%. As a result, the head of the pump increased. However, such structural features lead to decreasing of static head and
increasing of dynamic head. This is not acceptable, since in this case further conversion of the dynamic head into a static one occurs with the increasing of hydraulic losses in the pipeline.

The influence of the number of blades on the fluid flow character in the centrifugal pump stage was determined in the study [5]. While using an impeller with a larger number of blades, it is found that there is some greater pressure fluctuations in the guide vanes.

In the study [6], the fluid flow character in the flowing part of a low specific-speed pump was determined. The existence of vortex formation zones behind the impeller at the backside of the blade is established. This phenomenon leads to decreasing of the head and the efficiency of the centrifugal pump.

In the paper [7] the influence of the discharge nozzle design on the characteristics of centrifugal pumps was investigated. It is established that radial discharge nozzles of round section allow reaching the maximum head and efficiency of the pump.

In the studies [8, 9] the influence of the gap between the impeller and the discharge device on the operating parameters of the diagonal pump is determined. It is established that increasing of the pump head is achieved by reducing the gap between them. However, while the gap decreases, an unsteady stream enters the discharge nozzle, which leads to increasing of noise, vibration, and increasing of the wearing of the corresponding surfaces.

In studies [10, 11] investigated using of two-tier vane systems for increasing the head of the centrifugal pump stages. It is determined that this increases the pump head to 12%.

In the paper [12], promising ways of improving the design of the impeller are proposed. The prospects of using impellers with two and three additional blades are considered. It allows increasing the centrifugal pump head. However, this design involves the use of small blades. It leads to difficulties in the manufacturing process, as well as in the operation of impellers when transporting liquids with solid (especially abrasive) inclusions. The durability of additional blades in such operating conditions is lower than in the main ones. This results in a rapid deterioration of the pump's operating parameters during its operating.

In the study [13], it was found that the operation of low specific-speed pumps at flow capacities less than nominal was due to the presence of declining zones of their Q-H characteristics. Such specific speed factor characterized in particular for impellers with high blade outlet angle $\beta_2$ [14]. The operation of pumps at non-calculated modes can lead to their unstable operation [15]. The main causes of presence of Q-H characteristics declining zones are identified. Some ways to reduce hydraulic losses in these modes are also suggested. This avoids the occurrence of declining zones and stabilize the operation of pumping equipment in such modes.

In the research [16], the influence of the impeller blade outlet angle $\beta_2$ of the centrifugal pump on the character of its Q-H characteristic was considered. It is established that as the impeller outlet angle $\beta_2$ increases, the gentles of the Q-H characteristic decreases.

Given the literary analysis, we can state the following. Most ways of increasing the centrifugal pump head require increasing of the rotational frequency of the pump, or its overall dimensions. Thus, the first case requires the replacement of the drive machine with a similar with higher rotational frequency (in most cases asynchronous motor). The second case requires the replacement of the pump as a whole for the pumping installations implemented in the industry. Both ways significantly increase the total pump installation life cycle cost due to the high investment costs of the proposed modernization measures.

The possibility of increasing the pumps pressure by changing the design of the impeller blade system was established. This minimizes investment costs. At the same time, for some industries it is characteristic the rapid wearing of the elements of the pump flowing part (impeller, guide vanes). In this case, the upgrade of the pumping equipment is possible during the scheduled repair process by replacing the flowing part elements. As a result, the total pump installation life cycle cost is lower than in the previous two cases. However, the achieved pump head increasing indicators are insufficient to satisfy industry requirements.
Thus, work of further improvement of the impeller blade system without changing its overall dimensions looks promising. It will increase the pump head while ensuring the minimum pump installation life cycle cost.

3. The purpose and objectives of the research

The purpose of the study is to increase the centrifugal pump head by improving its flowing part while ensuring the minimum pump installation life cycle cost.

To achieve this goal identified the following key objectives:

- theoretical substantiation of the possibility of increasing the centrifugal pump head;
- determination of the main ways of centrifugal pump head increasing while ensuring the constant overall dimensions of its flowing part;
- studying of ways to minimize hydraulic losses to achieve high efficiency of a centrifugal pump.

4. Theoretical basis of the research

From the course of turbomachine theory, it is known, that the theoretical head of the centrifugal pump impeller is determined by the dependence:

\[ H_t = \frac{\omega}{g} (v_{t2} R_2 - v_{t1} R_1), \]  

(2)

where, \( \omega \) - angular velocity of the pump impeller, \( v_{t1}, v_{t2} \) - tangential (circumferential component of the absolute velocity) at the inlet and outlet of the impeller, \( R_1, R_2 \) - impeller radius at the inlet and outlet.

Thus, the pump impeller theoretical head is influenced by the following factors. Increasing the angular velocity (rotational frequency) of the impeller increases the pump impeller theoretical head. Increasing the radius at the outlet and reducing the radius at the inlet of the impeller also increases the pump impeller theoretical head. However, changing these parameters requires significant investment costs. The first case requires an asynchronous motor to be replaced by a similar one with an increased rotational frequency. The second case requires changing the overall dimensions of the flowing part. In practice, this means the need to completely replace the installed pump units with upgraded ones.

Increasing the pump impeller theoretical head is also achieved by increasing the tangential velocity outlet \( v_{t2} \) and reducing the tangential velocity inlet \( v_{t1} \) to the impeller. In formula (2) these parameters are included in the products \( v_{t2} R_2 \) and \( v_{t1} R_1 \). Since for slow specific-speed impellers of centrifugal pumps, the radius of the impeller outlet \( R_2 \) is generally 2-3 times larger than the radius of the impeller inlet \( R_1 \), the effect of the tangential velocity of the impeller outlet \( v_{t2} \) on the theoretical head is much greater than the tangential velocity of the impeller inlet \( v_{t1} \).

Figure 2 shows the velocity triangles at the inlet (Fig. 2 a) and the outlet (Fig. 2 b) of the impeller.

![Figure 2. Velocity triangles at the inlet (a) and outlet (b) of the impeller.](image)

The absolute velocity \( v_i \) (\( i = 1, 2 \)) in the velocity triangle is decomposed into two components: the tangential \( v_t \) and the radial \( v_r \). The value of the radial velocity \( v_r \) is defined as the ratio of the flow
capacity passing through the impeller to the area of the corresponding cross section of its intervane channels. The tangential speed is directly influenced by the impeller blade angle $\beta$ in the corresponding cross-section of the impeller intervane channels. In this case, increasing of this angle leads to increasing of the value of the tangential velocity $v_t$.

Most centrifugal pump impellers are designed with low blade inlet angle ($\beta_1 < 25^\circ$) and blade outlet angle ($\beta_2 < 35^\circ$). Designing the impeller with a higher blade outlet angle ($\beta_2 = 70$-$80^\circ$) is a promising way of increasing its theoretical head. However, the unresolved issue in practice is the increasing of hydraulic losses due to increasing of diffusivity and rapid rotation of the fluid flow in its intervane channels. The development of a high-pressure impeller of low specific-speed ($n_s < 65$) while providing a high efficiency is the subject of study of this research.

5. The results of the research of the fluid flow in the centrifugal pump flowing part

5.1. Research methodology

The experimental study was performed by conducting a numerical investigation using Ansys CFX software. Flowing parts of the impeller of standard and advanced design were tested. The operating parameters of the pump are as follows: flow capacity $Q = 180$ m$^3$/h, the rotational frequency of the pump is 3000 rpm.

Operating environment - water at 25˚ C. Operating mode is turbulent. For the closure of the Reynolds equations, the standard k-ε turbulence model is used.

The design area (Fig. 3) is the rotational part of the pump (impeller). An unstructured calculation grid was built for it. The total number of cells in the calculation area is 1.1 million cells.

As a boundary condition at the inlet of the calculating area, the flow capacity of the centrifugal pump impeller was set at 50 kg/s (180 m$^3$/h).

A pressure equal to 10 MPa was set as the boundary condition at the outlet of the pump impeller. Due to the possibility of backflows at the outlet of the calculation area, the boundary condition "Opening" is set.

To ensure a steady flow in the impeller intervane channels at the inlet surface, an elongation of 120 mm and at the outlet surface an elongation of 50 mm is constructed.

Determination of the impeller total head was performed using the function "Total Pressure" by the formula:
(massFlowAve (Total Pressure in Stn Frame) @outlet – massFlowAve (Total Pressure in Stn Frame) @inlet) / (997,1 [kg/m^3] · g),

where, massFlowAve - function of flow averaging by weight, Total Pressure in Stn Frame - full pressure (static and dynamic), @inlet, @outlet - arrangement of place of pressure measurement at the inlet and outlet of impeller, 997,1 [kg/m^3] - density of water at 25˚ C, g = 9,81 m/s^2 - acceleration of free fall.

The total pressure is a function, which defined as the pressure that would exist at a point if the fluid was brought instantaneously to rest such that the dynamic energy of the flow converted to pressure without losses. This is an explanation of this function from the Ansys CFX solver theory guide [17]. Thus, this function corresponds to the total head of the impeller without taking into account the losses of converting the dynamic energy of the flow into pressure without losses (hereinafter referred to as "total head").

5.2. Designing of the high-head impeller construction of low specific-speed pump

The standard design impellers of the centrifugal pumps are designed with the blade outlet angle β₂ < 35˚ (N0) in order to avoid additional hydraulic losses. These loses can be attributed to the large diffusivity of the blades and local hydraulic losses due to the separation of the flow from the wall surfaces of the blades and impeller wheels. Thus, properly designed impellers are characterized by low hydraulic losses due to the successful fluid flow in its intervane channels (Fig. 4).

As a result of the numerical investigation, it was established that the fluid flow in the impeller intervane channels occurs with minimal areas of flow separation, which confirms its high efficiency.

According to the results of the research, it is determined that the total head of this impeller is 158 m, which is a low indicator. This confirms the opinion of the significant influence of the blade construction on the value of the total head.

During the study, the impeller (N1) was developed with an increased blade outlet angle β₂ = 75˚ (Fig. 5). To ensure minimum head losses at the inlet of impeller, the blade inlet angle did not change, β₁ = 24˚.

**Figure 4.** Flow diagram of the fluid flow (a) and the relative velocity w distribution in the impeller intervane channels N0 with the blade outlet angle β₂ = 25˚ (b) as a result of numerical investigation.
Figure 5. Flow diagram of the fluid flow (a) and the relative velocity w distribution in the impeller intervane channels N1 with the blade outlet angle $\beta_2 = 75^\circ$ (b) as a result of numerical investigation:
1, 2, 3 - the main areas of local hydraulic losses.

The N1 impeller allows increasing the total head to 178.6 m, i.e. by 13%. However, in this construction significant areas of flow separation are formed. It results in significant hydraulic losses, which results that the pump head and its efficiency do not reach the maximum value.

In this case, zones 1 and 3 are associated with the rapid change of flow direction in the impeller intervane channels. The blade angle changes from 24° at the inlet ($\beta_1$) to 75° at the outlet ($\beta_2$) of the impeller. In this case, in zone 1, the force interaction of the blade with the fluid flow is minimal. This results in minimal energy transmitted by the blades to the fluid flow over a wide range of a blade length. The flow separation zone 2 is associated with the high diffusivity of the N1 impeller intervane channels. In addition to increased hydraulic losses, this zone is dangerous by backflows near the impeller outlet. In this case, they increase when the pump flow capacity decreases. As a result, in the zones of reduced pump flow capacities there is a decrease of the head. It observes a declining zone of the pump Q-H characteristic.

The character of the fluid flow in the impeller N1 lets implying the following hypothesis. Using of a two-tier impeller design can stabilize fluid flow in its intervane channels. It will reduce the flow separation zones, reduce hydraulic losses and increase the total head. This hypothesis underlies the design of impeller N2 (Fig. 6). Thus, the second tier blades are the shortened blades identical to the main.

As a result of performing an additional row of blades in the impeller N2, a significant reduction of the flow separation zone 1 is achieved. It could be explained by the greater correspondence of the fluid flow direction and the blade construction.

An additional row of blades divides the intervane channel into two separate channels with less diffusivity. From this point of view, the hypothesis of the reduction of the flow separation zone 2 was formulated and confirmed by the numerical investigation. However, due to the presence of an additional row of blades, two new zones of flow separation zones, 3 and 4, arises. Zone 3 is conditioned by the presence of a finite thickness of the additional row of blades, as well as the need to flow around them. Zone 4 from the backside of the shortened blade arises because of the diffusivity of
the channel, as well as slightly less pressure from the backside of the blades than from the operating ones. In general, zones 3 and 4 are small.

**Figure 6.** Flow diagram of the fluid flow (a) and the relative velocity \( w \) distribution in the impeller intervane channels N2 with the blade outlet angle \( \beta_2 = 75^\circ \) (b) as a result of numerical investigation:

1, 2, 3 - the main areas of local hydraulic losses.

These changes make it possible to increase the impeller's total head to 196.7 m. This is 24.5% higher than of the N0 impeller and 10.1% higher than of the N1 impeller.

Taking into account the positive experience of implementing of the two-tier impeller design with an increased value of the blade outlet angle \( \beta_2 \), the hypothesis about the importance of making changes of the design of a blade additional row was made. It is proposed to perform it using straight blades in which the blade outlet angle \( \beta_2 \) corresponds to the same angle of the main blades. This allows some reduction of the hydraulic losses of the flowing the additional row of blades, since in this case the direction of such blades will correspond to the fluid flow direction. The proposed changes formed the basis of the N3 impeller (Fig. 7).

Performing additional blades of straight form allows coordinating their shape with the direction of fluid flow. It results in decreasing of the separation flow zone 1 and more coordinating of fluid flow and the impeller main blades. It should be noted that the separation flow zone 2 also slightly decreases. It takes place due to decreasing of the diffusivity of the intervane channel part between the backsides of the additional blades and the operating one of the main blades.

Measures of improving the N3 impeller increased the pump's total head to 197.9 m. This is 25% higher than of the N1 impeller and 0.6% higher than of the N2 impeller.

The presence of the separation flow zone 2 makes it possible to hypothesize the possibility of improving this design by performing an additional series of blades in a "wedge" shape. It allows reducing hydraulic losses by reducing the diffusivity of the channel between the backside of the additional blade and the operating side of the main blade. These changes are implemented in the model of the impeller N4 (Fig. 8).

Because of the numerical investigation, the reduction of the hydraulic losses zone has been practically established. It is due to decreasing of the diffusivity of the channel that described above. However, the hydraulic losses zone due to the frontal flowing the edges of the additional blades still cannot be eliminated.
Figure 7. Flow diagram of the fluid flow (a) and the relative velocity $w$ distribution in the impeller intervane channels N3 with the blade outlet angle $\beta_2 = 75^\circ$ (b) as a result of numerical investigation: 1, 2 - the main areas of local hydraulic losses.

Figure 8. Flow diagram of the fluid flow (a) and the relative velocity $w$ distribution in the impeller intervane channels N3 with the blade outlet angle $\beta_2 = 75^\circ$ (b) as a result of numerical investigation: 1 - the main area of local hydraulic losses.

Structural changes of the N4 impeller increased the impeller’s total head to 200.2 m. This is 26.7% higher than of the N0 impeller and 1.2% higher than of the N3 impeller.

5.3. Discussion of the research results
The ways of improving the impeller design of a centrifugal pump to increase its total head are investigated in the research. It is established that the design of its blade system allows increasing the pump’s total pressure up to 26.7% (table 1).
Table 1. Results of the research.

| Parameter                              | Impeller N0 | Impeller N1 | Impeller N2 | Impeller N3 | Impeller N4 |
|----------------------------------------|-------------|-------------|-------------|-------------|-------------|
| Total Head, m                          | 158         | 178.6       | 196.7       | 197.9       | 200.2       |
| Total Head versus the impeller N0, %   | 0           | +13         | 24.5        | 25          | 26.7        |
| Total Head versus the impeller N1, %   | -13         | 0           | 10.1        | 10.8        | 12.1        |

The proposed impeller design (Fig. 9) is promising in view of the pump installation total life cycle cost. Since this option does not require a change of the pump rotational frequency, it eliminates the necessity of replace the motor of industrial rooted pumping installations.

![Advanced N4 impeller design](image)

Figure 9. Advanced N4 impeller design.

One of the approach while conduction the research was the inability to increase the overall dimensions of the impeller. It reduces the design work cost of new pumps by using old housing parts. Upgrading of industrial rooted pumps is possible by replacing only the impeller without the necessity to replace the housing parts. In the case of using multistage centrifugal pumps, the cost of the housing parts may be several times greater than the cost of the flowing part (Fig. 1a). The proposed upgrade option can significantly reduce investment costs.

The prospective industry for implementing the results is primarily oil production. The technological processes in this field are associated with the creation of excess reservoir pressure in the oil fields by supplying them with technical water by centrifugal multistage pumps. The working fluid contains inclusions (including abrasives), which significantly shorten the life of the pump flowing part. In these conditions, the process of upgrading the pump units by replacing standard impellers with advanced N4 impellers can be carried out during the scheduled repair process. In this case, the process of upgrading the pumping installations does not require additional investment costs.

As a result of the numerical investigation, the following fact is established. The average relative flow velocity in the intervane channels of the standard impeller is approximately 12–15 m/s (Fig. 4b). In the advanced impeller N4, it is approximately 10–12 m/s (Fig. 8b). This is due to the smaller blade coverage angle $\varphi$ due to a more rapid increase of the blade angle $\beta_i$ in the direction of fluid flowing.
Thus, the authors hypothesize that the wearing of impeller intervane channels surfaces can be reduced when transporting liquids with abrasive inclusions. With this in mind, the duration of the overhaul cycle using the advanced impeller N4 is slightly increases compared of using a standard impeller. Thus, operating and repair costs also reduces.

It should be noted that the wedge-shaped form of the second tier blades less prone to rapid wearing than the standard additional blades of the second tier (Fig. 6a). This is due to its more streamlined shape, as well as its greater thickness near the exit of the impeller.

Thus, the improvement of centrifugal pumps to increase their head can be done by changing the design of the impeller blade system. In the research, the total head is increased up to 26.7%. At the same time, the improvement of the pump design is achieved with the achievement of the minimum pump installation life cycle cost for the design and manufacture of new pumps, as well as the modernization of the pump installations rooted into the production.

The study of the influence of the proposed design impeller on the character of the pump head Q-H and the energy η-H characteristics of the centrifugal pump is the subject of a separate research. It is explained of the necessity to determine the mutual influence of the structural elements of the rotor part (impeller) and the stator part (diffusor and return channels of the guide vanes) on the character of the fluid flow in the pump flowing part.

6. Conclusions
1. The theoretical basis for the possibility of increasing the head of the centrifugal pump impeller is substantiated in the research. Such requirements for pumping equipment are characteristic firstly for the field of oil production, where multistage centrifugal pumps are used to create reservoir pressure at oil fields. It is established that the design of the blade system has a direct effect on the total head created by the impeller.

2. In order to increase the total head of the centrifugal pump, a scheme of the impeller blade system is proposed, which characterized by an increased blade outlet angle (β2 = 75°). The formed intervane channels made it possible to increase the pump's total head to 13%. However, rapid change of fluid flow direction and high diffusivity of the impeller intervane channels resulted in appearance of some zones of additional hydraulic losses. This decreased the efficiency of the impeller.

3. The design of impeller additional blades of the second tier was developed in the research. This made it possible to stabilize the fluid flow in its intervane channels and to reduce additional hydraulic losses. As a result of the study, the total head of the advanced impeller N4 was increased up to 26.7% compared to the standard one.

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