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ТРАНСФОРМАЦІЯ СТРУКТУРИ ПОТОКУ ТА ЕНЕРГЕТИЧНІ ХАРАКТЕРИСТИКИ ВІЛЬНОВИХРОВОГО НАСОСА В РЕЖИМАХ ЧАСТКОВОГО НАВАНТАЖЕННЯ (Q = 0,05–0,6 $Q_{ном}$)

У роботі наведено результати чисельного дослідження робочих характеристик вільновихрового насоса СВН 80-32 у зоні знижених подач Q = 0,05–0,6 Qном. Метою дослідження є встановлення закономірностей зміни напірних та енергетичних показників і визначення особливостей трансформації структури внутрішнього потоку при переході від помірного до глибокого недовантаження. Розрахунки виконано в програмному середовищі ANSYS CFX у стаціонарній постановці з використанням моделі турбулентності k–ε. Проаналізовано два конструктивні виконання робочого колеса – одноярусне та двоярусне. Побудовано інтегральні характеристики H(Q) та η(Q), визначено зміни повного напору, споживаної потужності та коефіцієнта корисної дії в досліджуваному діапазоні подач. Встановлено, що в зоні недовантаження напір змінюється незначною мірою, тоді як ККД суттєво знижується через інтенсифікацію внутрішньої рециркуляції та зростання гідравлічних втрат у вільній вихровій камері. Показано, що при Q = 0,05 Qном формується стійка циркуляційна структура, яка забезпечує підтримання напору за одночасної деградації енергетичних показників. Виявлено, що двоярусне робоче колесо забезпечує дещо вищі значення ККД у верхній частині досліджуваного діапазону подач. Отримані результати можуть бути використані для обґрунтування меж раціональної експлуатації та вдосконалення конструкції вільновихрових насосів.

Ключові слова: *вільновихровий насос; вихровий насос; робота при частковому навантаженні; режим малих подач; гідравлічний ККД; CFD-моделювання; модель турбулентності k–ε; структура внутрішнього потоку; рециркуляція; енергетичні показники.*

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FLOW STRUCTURE TRANSFORMATION AND ENERGY PERFORMANCE OF A TORQUE-FLOW PUMP UNDER PARTIAL-LOAD CONDITIONS (Q = 0.05–0.6 Q_{NOM})

The research presents the results of a numerical investigation of the operating characteristics of a torque-flow pump SVN 80-32 in the low-flow range Q = 0.05–0.6 Q_{nom}. The aim of the study is to determine the regularities of changes in head and energy performance indicators and to identify the transformation features of the internal flow structure under partial-load conditions. The simulations were performed in ANSYS CFX using a steady-state formulation and the k–ε turbulence model. Two impeller configurations, single-tier and double-tier, were analyzed. Integral characteristics H(Q) and η(Q) were obtained, and variations of total head, consumed power, and hydraulic efficiency within the investigated flow range were determined. The results show that in the partial-load zone the head remains relatively stable, while the efficiency significantly decreases due to intensified internal recirculation and increased hydraulic losses in the vortex chamber. At Q = 0.05 Q_{nom} a stable circulatory flow structure is formed, maintaining head generation despite severe efficiency degradation. The double-tier impeller demonstrates slightly higher efficiency values in the upper part of the investigated flow range. The obtained results may be used to substantiate rational operating limits and to improve the design of torque-flow pumps operating under variable flow conditions.

Keywords: *torque-flow pump; vortex pump; partial-load operation; low-flow regime; hydraulic efficiency; CFD modeling; k–ε turbulence model; internal flow structure; recirculation; energy performance.*

1. Introduction

Torque-flow pumps (Fig. 1) are widely used for pumping liquids containing solid and fibrous inclusions due to the presence of a free vortex chamber and the reduced risk of clogging of the flow passage.

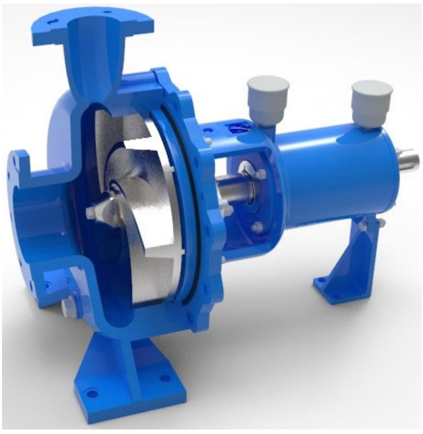


Fig. 1. **Design features of torque-flow pumps of the SVN type**

Classical experimental and analytical studies have shown that the operating process in such pumps is characterized by a combination of blade and vortex mechanisms of energy transfer, while the head characteristic typically exhibits a relatively flat shape over a wide flow rate range [1, 2]. Further generalizations of experimental data confirm the significant influence of the geometric parameters of the impeller and the flow passage on head and energy performance, particularly on the magnitude of hydraulic losses in the vortex chamber [3, 4].

In recent years, considerable attention has been paid to CFD investigations and parametric optimization of pump tiers, which allow analysis of velocity and pressure fields as well as assessment of the influence of design factors on pump performance without conducting a large number of full-scale experiments [5, 6, 7]. For engineering analysis, steady-state formulations and RANS turbulence models are widely applied, providing an acceptable compromise between accuracy and computational cost [8]. Some studies also demonstrate that factors such as wall roughness may significantly affect CFD-based predictions of energy performance indicators [9].

At the same time, most available studies focus on operating conditions close to the nominal regime, whereas in real operating conditions pumping units often function in the partial-load zone due to throttling, variable network demand, or technological constraints. For torque-flow pumps, a reduction in flow rate is accompanied by intensified recirculation phenomena and increased flow non-uniformity, which may lead to a significant decrease in efficiency and alterations in the flow structure within the interblade channels and the vortex chamber [1, 2]. Unsteady effects and the growth of vortex structure intensity in vortex pumps operating far from nominal conditions are actively discussed in recent publications, highlighting the relevance of a detailed analysis of low-flow regimes [10, 11].

The issue of operational reliability under off-design conditions is also of practical importance. Variations in hydrodynamic loading and deterioration of energy performance may be accompanied by increased vibration manifestations and risks of damage to pump components, as confirmed by approaches to failure and reliability analysis of pumping equipment [12, 13]. In addition, for pumps operating over a wide range of regimes, cavitation resistance remains an important aspect, and modern experimental and numerical approaches propose criteria and indicators for assessing cavitation inception from the standpoint of energy efficiency and reliable operation [14, 15].

In this context, the investigation of operating characteristics and internal flow structure of a torque-flow pump in the low-flow range $Q = 0.05–0.6\ Q_{nom}$ is of particular relevance, since a qualitative restructuring of the internal flow occurs and degradation of energy performance becomes apparent in this zone. In the present study, a numerical analysis of performance characteristics and flow fields of a torque-flow pump with two impeller configurations was performed in order to establish the regularities of head and efficiency variation and to compare the influence of impeller design on flow uniformity under partial-load conditions [3, 5, 6, 7].

2. Literature Review and Problem Statement

An analysis of recent studies shows that the majority of research in the field of torque-flow pump implementation is focused on geometric optimization of impellers and improvement of energy efficiency in the vicinity of the nominal flow rate. Parametric investigations of blade wrap angle, axial clearances, number of blades, and configuration of interblade channels demonstrate a significant influence of these factors on head and efficiency [5, 6, 7]. Numerical methods are actively applied for multi-objective optimization and performance prediction of pumps of various types [16, 17, 18].

A separate research direction concerns the improvement of impeller design, particularly through the application of multi-tier or combined blade systems. It has been shown that such configurations may provide a more uniform velocity distribution in the interblade channels and influence the shape of the head characteristic [3, 4]. Within the Ukrainian scientific school, it has also been established that the geometric parameters of curvilinear blades significantly determine the structure of the internal flow and the ratio between blade and vortex mechanisms of energy transfer [19, 20], while the combined operating process may alter the pump's energy performance depending on the operating regime [21, 22].

At the same time, the literature analysis reveals an imbalance in the investigation of different operating regimes. While nominal and overload conditions have been thoroughly examined from the standpoint of CFD analysis and experimental validation, the low-flow region has been studied only fragmentarily. Existing publications are generally limited to a qualitative statement of increased losses and reduced efficiency at decreased flow rates, without a systematic analysis of the evolution of velocity fields in the interblade channels and the free vortex chamber.

Recent studies devoted to the analysis of unsteady processes in torque-flow pumps indicate that deviations from nominal operation are accompanied by increased intensity of vortex structures and hydrodynamic oscillations [10, 11]. However, even in these investigations, the main emphasis is placed on dynamic characteristics, whereas a systematic analysis of energy performance degradation in the deep partial-load range remains limited.

An additional factor complicating the assessment of low-flow regimes is the possible increase in local hydraulic losses and enhanced sensitivity to cavitation phenomena, as confirmed by modern experimental and numerical studies of various pump types [14, 15]. From a practical standpoint, prolonged operation outside the recommended flow range is associated with increased vibration loading and a higher risk of equipment reliability reduction [12, 13], which further emphasizes the necessity of clearly defining rational operating limits.

Thus, despite the availability of numerous studies devoted to geometric optimization and CFD modeling of torque-flow pumps, the issue of regularities in head and energy performance variation in the range $Q = 0.05 – 0.6\ Q_{nom}$, as well as the transformation of the flow structure in this region, remains insufficiently investigated. This is particularly relevant for comparing different impeller configurations in terms of velocity distribution uniformity and the intensity of recirculation zones under deep partial-load conditions.

Addressing this problem requires a systematic numerical analysis using a consistent modeling methodology for a set of operating regimes within the low-flow range, followed by interpretation of the obtained results from the standpoint of hydrodynamic mechanisms responsible for loss formation.

3. Aim and Objectives of the Study

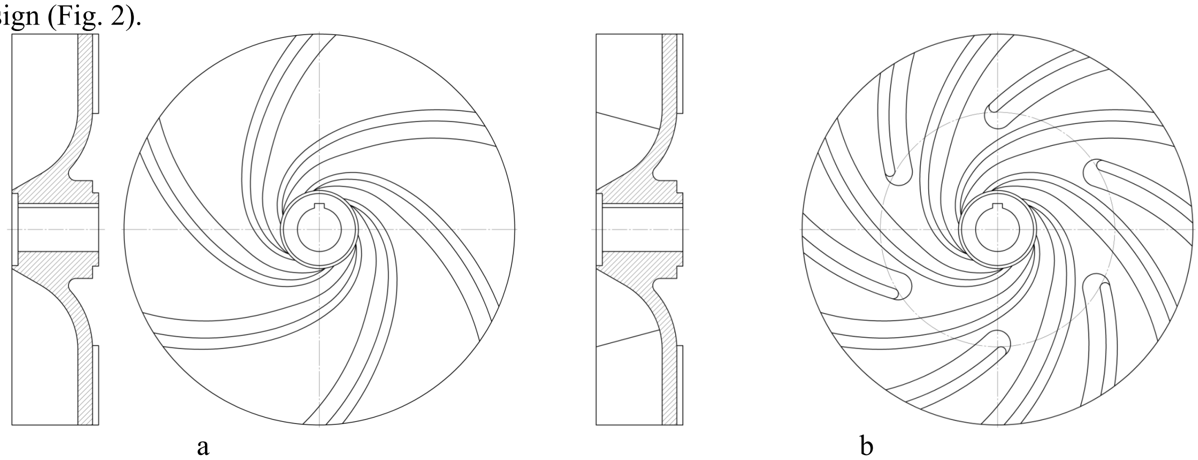
The aim of the study is to establish the regularities of variation of head and energy performance characteristics of a torque-flow pump in the low-flow range $Q = 0.05–0.6\ Q_{nom}$ and to determine the features of internal flow structure transformation when transitioning from moderate to deep partial-load conditions.

To achieve the stated aim, the following objectives were defined:

1. To perform numerical simulation of the operating process of a torque-flow pump in a steady-state formulation for a set of operating regimes within the range $Q = 0.05–0.6\ Q_{nom}$ using a consistent turbulence model.
2. To construct the head characteristic H–Q and the energy characteristic η–Q in the partial-load region and to determine the nature of energy performance degradation with decreasing flow rate.
3. To analyze the distributions of relative and absolute velocities in the interblade channels of the impeller and to identify changes in flow uniformity when transitioning to low-flow regimes.
4. To investigate the influence of impeller configuration (single-tier and double-tier) on the flow structure and the magnitude of hydraulic losses under partial-load operation.
5. To determine the rational operating limits of the pump in terms of maintaining an acceptable efficiency level and stability of the hydrodynamic flow structure.

4. Research Methodology

The object of the study is a torque-flow pump SVN 80-32 (pump capacity 80 m³/h, head 32 m at the design operating point) equipped with an impeller featuring curvilinear blade profiles and a free vortex chamber. Two impeller configurations were considered in the study, namely a single-tier and a double-tier design (Fig. 2).

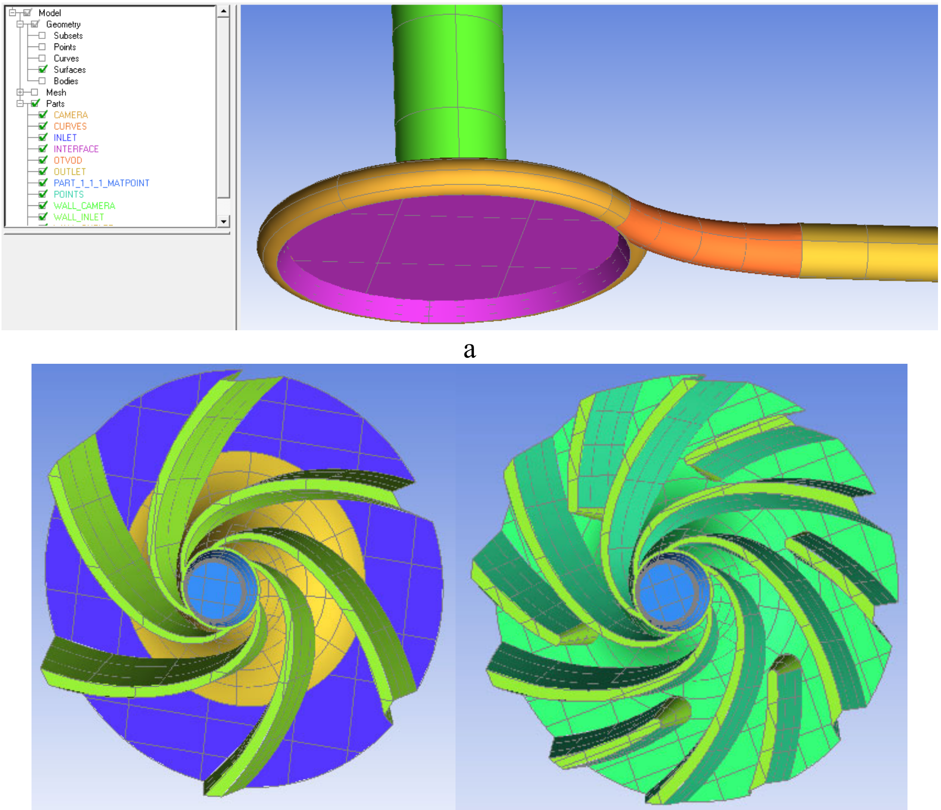


a b

Fig. 2. **Impeller configurations: a – single-tier; b – double-tier**

The geometry of the casing and other elements of the flow passage was identical for both variants, ensuring a correct comparison of the obtained results and allowing assessment of the influence of impeller design features on hydrodynamic processes in the low-flow range.

The three-dimensional model of the flow passage was developed taking into account the inlet nozzle, the impeller region, the free vortex chamber, and the discharge volute. After the geometry was formed, the computational domain was generated for subsequent numerical simulation. Discretization was performed using an unstructured volumetric mesh with local refinement in regions of high velocity and pressure gradients, particularly near the walls of the stator and rotor elements. To ensure accurate boundary layer resolution, prism layers were generated in near-wall regions. Mesh quality was controlled using standard geometric criteria, and the number of elements was selected to ensure grid-independent integral results.



a

b c

Fig. 3. **Three-dimensional models of the flow passage of the investigated SVN 80-32 pump: a – stator element; b, c – rotor element**

Numerical simulations were performed in the ANSYS CFX software environment using a steady-state formulation based on the Navier–Stokes equations for turbulent incompressible flow. A two-equation k–ε turbulence model was applied, which is widely used in engineering practice for the analysis of internal flows in pumping equipment and provides an acceptable balance between computational accuracy and cost. The interaction between the rotating and stationary parts of the model was accounted for using the Frozen Rotor approach. At the inlet of the computational domain, a mass flow rate corresponding to the investigated operating regime was specified, while static pressure was imposed at the outlet. No-slip and impermeability boundary conditions were applied on the walls. The working medium was considered as a homogeneous incompressible fluid with constant physical properties.

The simulations were carried out for operating regimes within the range $Q = 0.05–0.6\ Q_{nom}$. For each regime, the total head, consumed power, and hydraulic efficiency were determined. Convergence of the numerical solution was controlled by monitoring the reduction of residuals and stabilization of integral parameters, in particular the head and shaft torque. The obtained results were used to construct the head characteristic H as a function of Q and the energy characteristic η as a function of Q in the partial-load zone.

To analyze the transformation of the flow structure, visualization of absolute and relative velocity fields was performed in characteristic cross-sections of the flow passage. Particular attention was paid to assessing the uniformity of velocity distribution in the interblade channels and the formation of recirculation zones in the free vortex chamber as the flow rate decreased. The comparison of the two impeller configurations was carried out under identical boundary conditions, allowing an objective assessment of the influence of design execution on hydrodynamic and energy performance indicators under partial-load operation.

5. Research Results

5.1 Integral parameters of the torque-flow pump in the range Q = 0.05–0.6 Q_{nom}

The results of numerical simulation made it possible to determine the integral parameters of the SVN 80-32 pump in the low-flow region for both the single-tier and double-tier impeller configurations. The summarized data are presented in Table 1.

Table 1.

Integral parameters of the SVN pump with single-tier and double-tier impeller

Operating parameters	**Single-tier impeller**				**Double-tier impeller**			
Flow rate, % Q_{nom}	5	20	40	60	5	20	40	60
Flow rate, m³/h	4	16	32	48	4	16	32	48
Total head, m	38.71	38.33	39.23	39.63	38.72	37.82	38.72	40.59
Hydraulic power, W	428.4	1727	3692	5661	424.9	1705	3590	5803.6
Consumed power, W	5751	7539.8	10562	13663	5409	7034	10279	13691
Hydraulic efficiency, %	7.4	22.9	35.0	41.4	7.9	24.2	34.9	42.4

Analysis of the obtained results indicates that in the flow rate range Q = 0.05–0.6 Q_{nom} the total head varies only slightly for both impeller configurations. For the single-tier impeller, the head varies within 38.33–39.63 m, while for the double-tier impeller it ranges from 37.82 to 40.59 m. This behavior confirms the characteristic relatively flat head curve of torque-flow pumps in the partial-load region.

In contrast to the head, the efficiency demonstrates a pronounced dependence on flow rate. At Q = 0.05 Q_{nom}, the hydraulic efficiency for both configurations does not exceed 8 %, indicating intensive internal energy losses under deep partial-load conditions. As the flow rate increases to 0.6 Q_{nom}, the efficiency rises to 41.4 % for the single-tier and 42.4 % for the double-tier configuration. Thus, increasing the flow rate is accompanied by an improvement in flow structure and a reduction in recirculation losses.

The consumed power of the pump increases regularly with increasing flow rate. For the single-tier impeller, it varies from 5.75 kW to 13.66 kW, while for the double-tier configuration it ranges from 5.41 kW to 13.69 kW. At the same time, hydraulic power remains relatively low at small flow rates, which results in low efficiency values in this region.

Figure 4 presents the dependencies of head, efficiency, and consumed power on the relative flow rate for both impeller configurations. Graphical analysis shows that the differences between the designs are most noticeable in the range Q = 0.2–0.6 Q_{nom}. The double-tier impeller provides slightly higher efficiency values in the upper part of the investigated range, which may be associated with a more uniform velocity distribution in the interblade channels.

At the same time, in the deep partial-load region the differences between the configurations become less pronounced, since the dominant factor is the prevalence of circulatory motion in the free vortex chamber.

5.2 Transformation of the Flow Structure in the Low-Flow Region

The hydrodynamic pattern within the pump flow passage was analyzed for four characteristic operating regimes in the range Q = 0.6–0.05 Q_{nom}. Visualization of the absolute velocity fields makes it possible to trace the sequential restructuring of the flow as the discharge decreases and to determine changes in the ratio between blade and vortex mechanisms of energy transfer.

Figures 5–7 present the velocity distribution in the flow passage at Q = 0.6 Q_{nom}.

For this regime, relatively uniform filling of the interblade channels and preservation of directed fluid motion from the inlet to the impeller outlet are characteristic. The vortex chamber participates in the formation of a circulatory flow; however, the blade mechanism of energy transfer remains dominant. Recirculation zones are local and do not occupy a significant volume of the chamber. The hydrodynamic structure is relatively stable, which corresponds to the acceptable efficiency level observed in this regime.

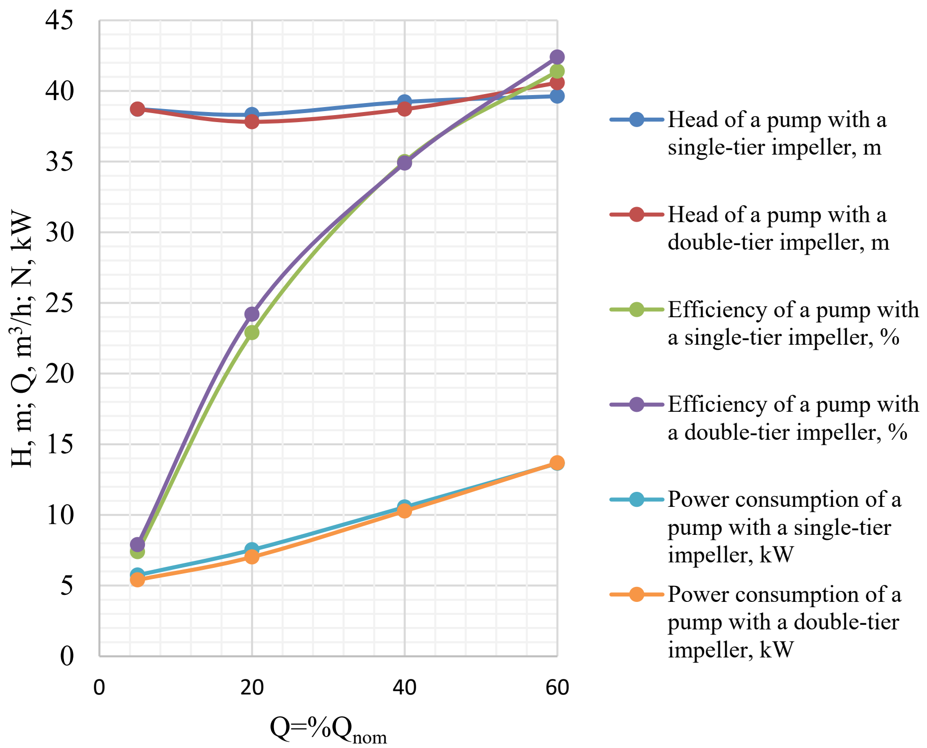


Fig. 4. **Integral parameters of the SVN pump with single-tier and double-tier impellers**

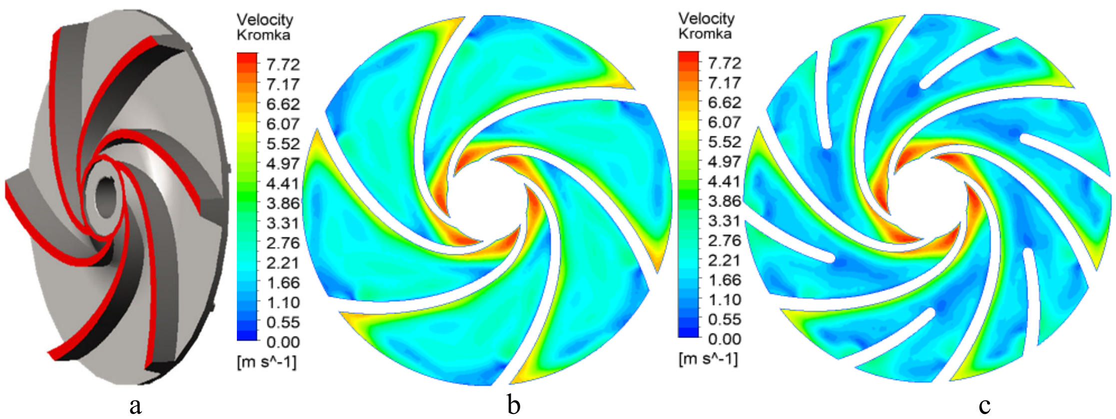


Fig. 5. **Relative velocity in the interblade channels near the leading edge (Q = 0.6 Q_{nom}): a – cross-section; b – single-tier impeller; c – double-tier impeller**

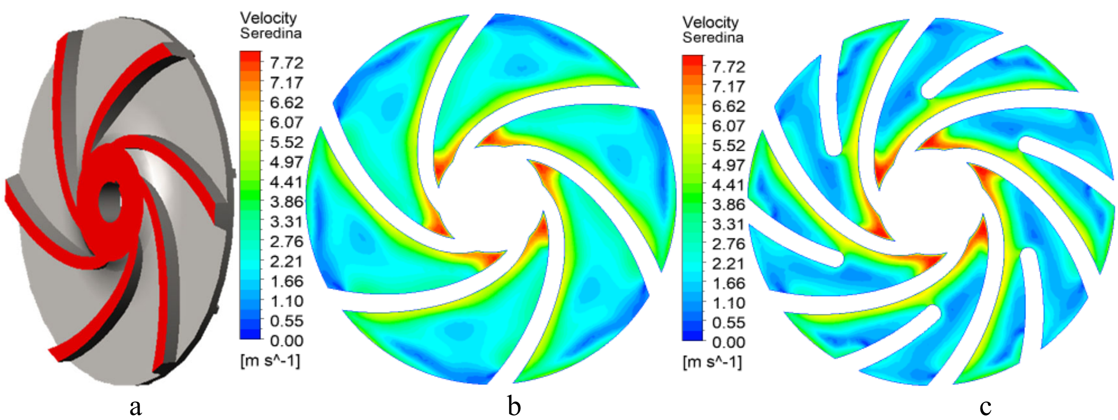


Fig. 6. **Relative velocity in the interblade channels along the midline of the blade passage (Q = 0.6 Q_{nom}): a – cross-section; b – single-tier impeller; c – double-tier impeller**

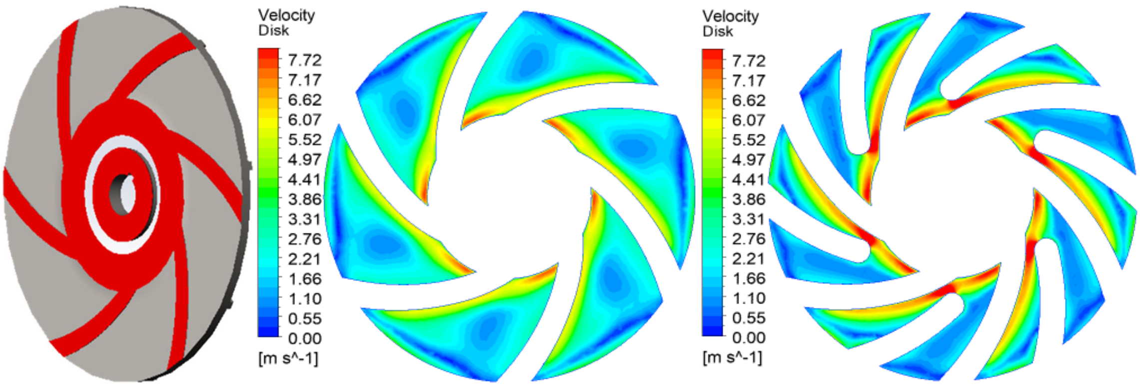


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Fig. 7. **Relative velocity in the interblade channels near the shroud (Q = 0.6 Q_{nom}): a – cross-section; b – single-tier impeller; c – double-tier impeller**

When the flow rate decreases to 0.4 Q_{nom}, as shown in Figures 8–10, an increase in the non-uniformity of velocity distribution in the interblade channels is observed. A portion of the flow begins to deviate from the main direction of motion, and the contribution of circulatory velocity components in the free vortex chamber increases. More extended low-velocity zones appear near the walls, indicating an increase in hydraulic losses. At the same time, the overall flow structure still retains a relatively ordered character.

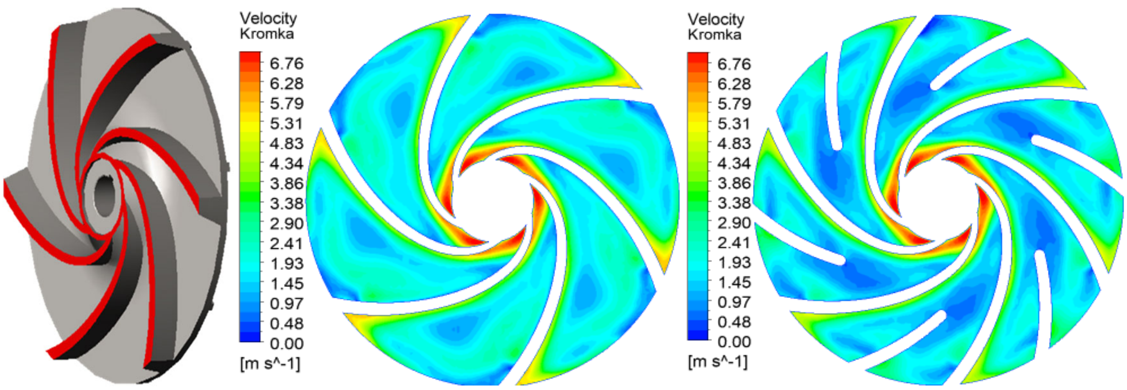

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Fig. 8. **Relative velocity in the interblade channels near the leading edge (Q = 0.4 Q_{nom}): a – cross-section; b – single-tier impeller; c – double-tier impeller**

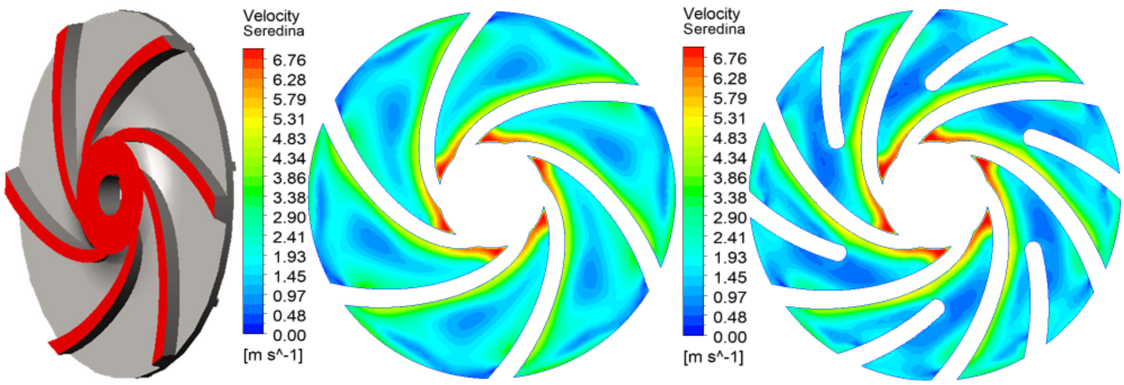

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Fig. 9. **Relative velocity in the interblade channels along the midline of the blade passage (Q = 0.4 Q_{nom}): a – cross-section; b – single-tier impeller; c – double-tier impeller**

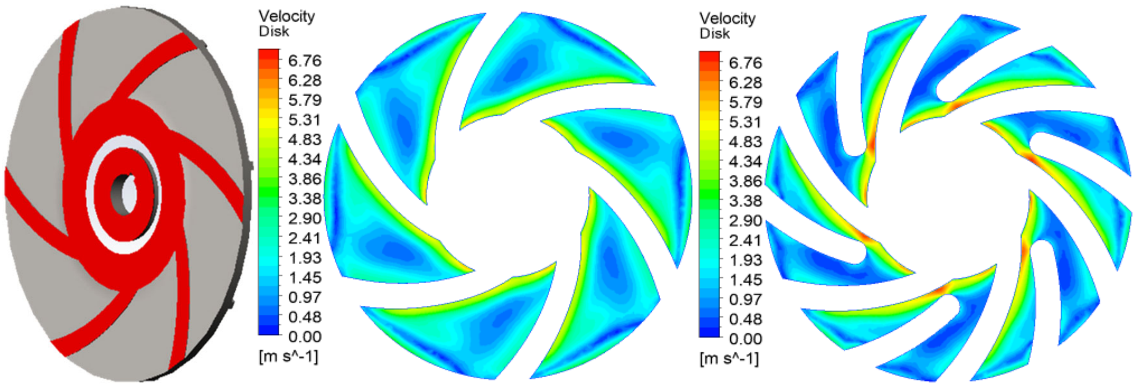

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Fig. 10. **Relative velocity in the interblade channels near the shroud (Q = 0.4 Q_{nom}): a – cross-section; b – single-tier impeller; c – double-tier impeller**

Figures 11–13 show the velocity distribution at Q = 0.2 Q_{nom}. In this regime, a substantial restructuring of the flow occurs. The non-uniformity of the velocity field in the interblade channels increases sharply, and part of the vortex chamber volume becomes occupied by recirculation zones. The tangential velocity component increases, indicating an intensification of the vortex mechanism of energy transfer. The influence of the blades decreases, and energy transfer is predominantly achieved through circulatory motion within the chamber.

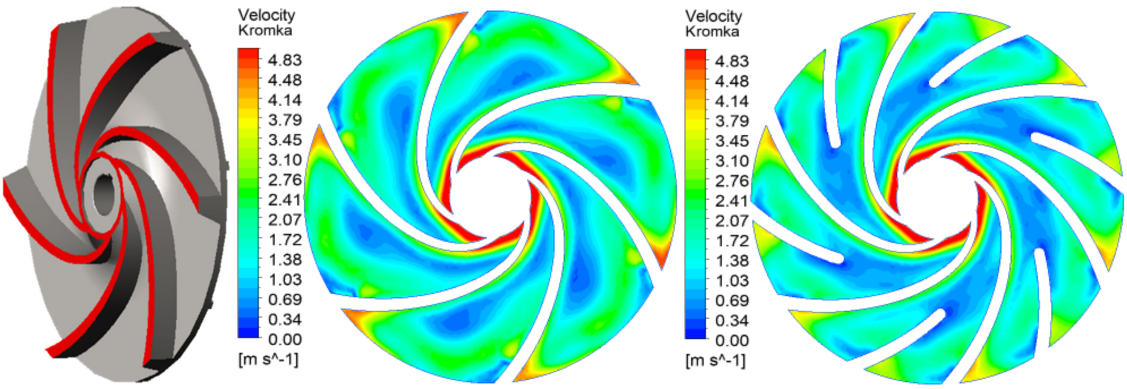

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Fig. 11. **Relative velocity in the interblade channels near the leading edge (Q = 0.2 Q_{nom}): a – cross-section; b – single-tier impeller; c – double-tier impeller**

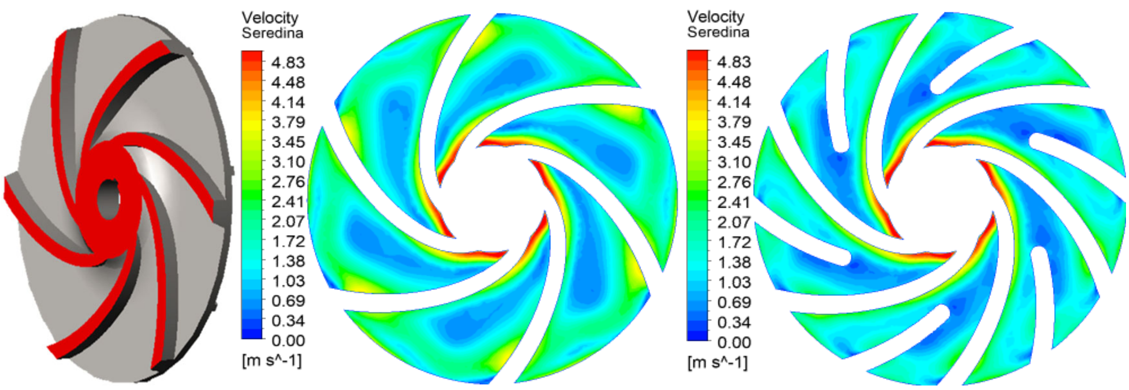

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Fig. 12. **Relative velocity in the interblade channels along the midline of the blade passage (Q = 0.2 Q_{nom}): a – cross-section; b – single-tier impeller; c – double-tier impeller**

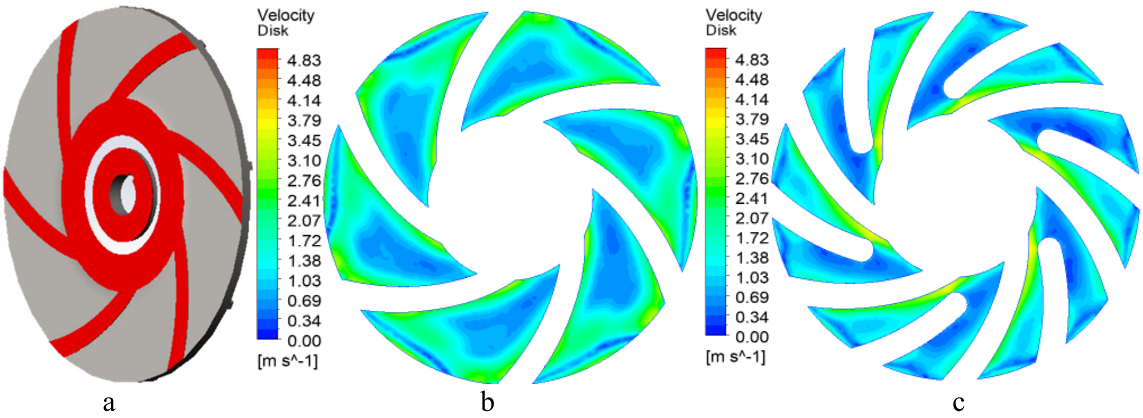

Fig. 13. **Relative velocity in the interblade channels near the shroud ($Q = 0.2\ Q_{nom}$): a – cross-section; b – single-tier impeller; c – double-tier impeller**

It is at this tier that a sharp decline in energy performance indicators is observed, which is associated with intensified internal losses and partial flow separation.

The most pronounced changes in flow structure are recorded at $Q = 0.05\ Q_{nom}$, as shown in Figures 14–16. The interblade channels are filled non-uniformly, and a significant portion of the fluid participates in internal recirculation. A stable toroidal vortex structure forms in the free vortex chamber, which effectively determines the nature of fluid motion within the pump. The blade mechanism of energy transfer loses its dominant role.

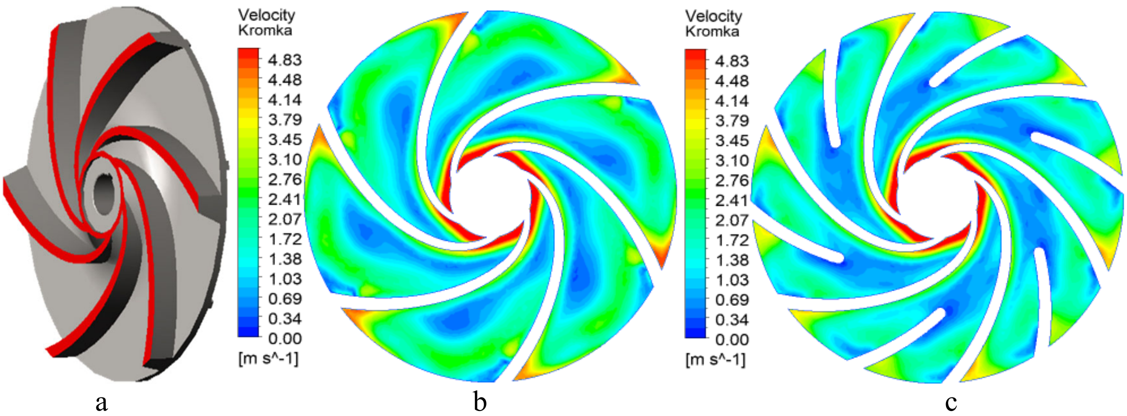

Fig. 14. **Relative velocity in the interblade channels near the leading edge ($Q = 0.05\ Q_{nom}$): a – cross-section; b – single-tier impeller; c – double-tier impeller**

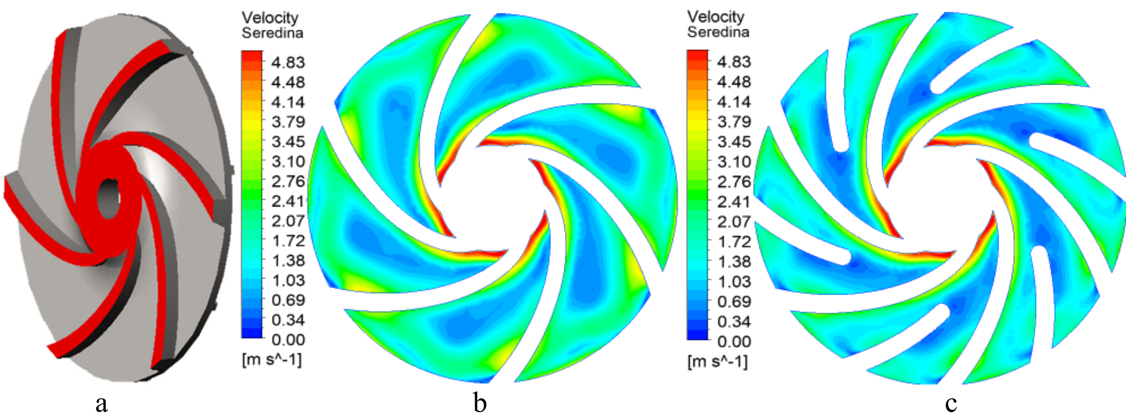

Fig. 15. **Relative velocity in the interblade channels along the midline of the blade passage ($Q = 0.05\ Q_{nom}$): a – cross-section; b – single-tier impeller; c – double-tier impeller**

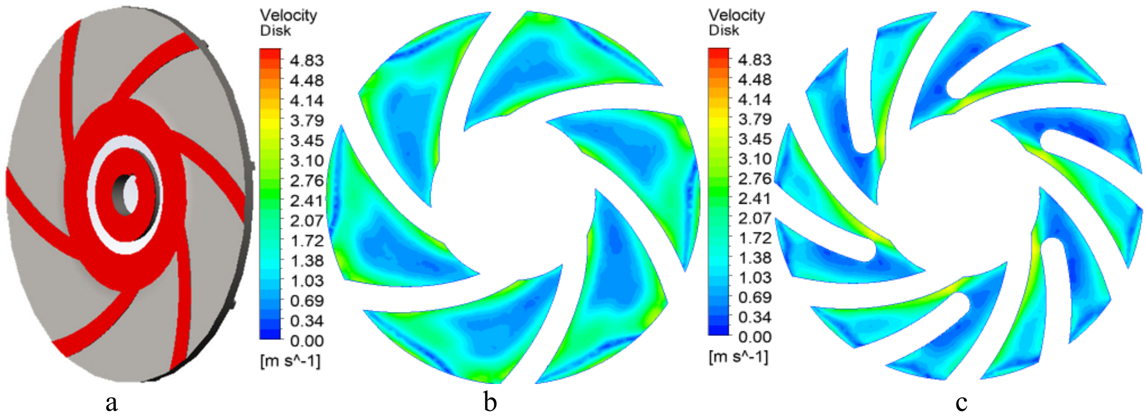

Fig. 16. **Relative velocity in the interblade channels near the shroud ($Q = 0.05\ Q_{nom}$): a – cross-section; b – single-tier impeller; c – double-tier impeller**

Despite the relatively stable head, the efficiency of the operating process decreases significantly due to the increase in the volume of low-velocity zones and intensified energy dissipation.

6. Discussion

The obtained results confirm that, for a torque-flow pump operating in the low-flow region, relative stability of the head characteristic is maintained while energy performance indicators degrade significantly. A similar behavior has been reported in classical studies devoted to vortex pump analysis [6, 23], where the relatively flat shape of the H(Q) curve was emphasized even under substantial deviation from the nominal regime. In the present study, it was established that within the range $Q = 0.05–0.6\ Q_{nom}$, the head variation does not exceed several percent, indicating the dominance of the circulatory pressure generation mechanism in the free vortex chamber.

At the same time, the efficiency demonstrates a pronounced dependence on flow rate. Under deep partial-load conditions, a developed system of internal recirculation is formed, leading to increased dissipative losses and intensive redistribution of kinetic energy within the chamber. Similar trends of efficiency reduction under partial-load operation have been reported in CFD studies of vortex and combined pumps, where the role of increased tangential velocity components and non-uniform filling of interblade channels has been emphasized.

The comparison of single-tier and double-tier impellers showed that structural modification does not have a decisive influence on head in the deep partial-load region; however, it affects the uniformity of the velocity field and the magnitude of hydraulic losses in the upper part of the investigated range. This is consistent with studies of multi-tier and combined blade systems, where additional blade elements were shown to contribute to more uniform velocity distribution and reduced local diffusivity within the channels. In the present case, the double-tier configuration provides slightly higher efficiency at $Q \geq 0.4\ Q_{nom}$, which can be explained by stabilization of blade inflow conditions and reduction in the scale of local flow separation.

Particular attention should be paid to the observed preservation of head at a substantial decrease in efficiency under the regime $Q = 0.05\ Q_{nom}$. This indicates a fundamental shift in the pressure generation mechanism of the torque-flow pump under deep partial-load conditions. While near-nominal operation is governed primarily by blade-induced momentum exchange, deep partial-load regimes are characterized by volumetric vortex-dominated pressure generation within the free chamber. In this case, the impeller plays a secondary role, mainly sustaining circulation rather than directly transferring energy through blade loading. The formation of a stable toroidal vortex structure within the chamber creates a circulation loop that maintains the head level but is accompanied by significant energy losses due to turbulent dissipation.

From a practical standpoint, the obtained results are important for defining rational operating limits. Although the pump is capable of maintaining head even at 5 % of the nominal flow rate, prolonged operation under such conditions is energetically inefficient and potentially undesirable from a reliability perspective. Increased flow non-uniformity, local low-velocity zones, and intensified circulatory structures may create prerequisites for elevated vibration loading and local fluid overheating, which is consistent with approaches used in reliability assessment of pumping equipment.

At the same time, certain limitations of the present study should be noted. The simulations were performed in a steady-state formulation using the k–ε turbulence model, which does not fully capture unsteady pulsation phenomena characteristic of deep partial-load regimes. Further research may be directed toward the application of unsteady models or experimental validation of the obtained dependencies.

Overall, the results deepen the understanding of hydrodynamic mechanisms governing the operation of torque-flow pumps in the low-flow region and complement existing studies that are primarily focused on nominal regimes. The established regularities of flow structure transformation may be used to improve impeller design and to develop recommendations for the permissible operating range of pumps of this type.

7. Conclusions

A comprehensive numerical analysis of the operating process of the torque-flow pump SVN 80-32 in the low-flow range Q = 0.05–0.6 Q_{nom} was carried out with comparison of two impeller configurations. The obtained results made it possible to establish the regularities of variation of integral characteristics and to identify the features of internal flow structure transformation under partial-load conditions.

1. It was established that the torque-flow pump SVN 80-32 in the range Q = 0.05–0.6 Q_{nom} maintains a relatively stable head level, while energy performance indicators demonstrate a pronounced dependence on flow rate. The variation of total head within the investigated interval is insignificant, confirming the relatively flat shape of the H(Q) characteristic in the partial-load region.

2. It was shown that under deep partial-load conditions, a sharp degradation of efficiency occurs due to increased internal hydraulic losses and the formation of developed recirculation zones in the free vortex chamber. At Q = 0.05 Q_{nom}, the hydraulic efficiency does not exceed 8 %, indicating the dominance of dissipative processes in the flow structure.

3. A regular transformation of the internal flow with decreasing discharge was revealed, from relatively ordered blade inflow at Q = 0.6 Q_{nom} to the formation of a stable toroidal vortex structure in the chamber at Q = 0.05 Q_{nom}. In this case, the contribution of the tangential velocity component increases, while the role of the blade mechanism of energy transfer decreases.

4. It was established that the double-tier impeller provides a more uniform velocity distribution in the interblade channels and slightly higher efficiency values in the upper part of the investigated flow range. Under deep partial-load regimes, the differences between the configurations decrease, since the dominant factor becomes the nature of circulatory motion in the free chamber.

5. The obtained results make it possible to outline the limits of rational pump operation. Although head maintenance is possible even at 5 % of the nominal flow rate, from energy and operational perspectives it is advisable to operate at Q ≥ 0.4 Q_{nom}, where an acceptable efficiency level and a more stable flow structure are ensured.

6. The study demonstrates that torque-flow pumps exhibit a dual-mechanism operating behavior, where the relative contribution of blade and vortex energy transfer mechanisms depends strongly on the flow regime. The identified transition zone provides a theoretical basis for optimizing impeller design and defining rational partial-load operating limits.

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