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Original Study Article

Study of the potential use of hydrodiodes to enhance the volumetric efficiency of a centrifugal pump

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ABSTRACT

BACKGROUND: For centrifugal pumps, especially those with low specific speed coefficients, the volumetric efficiency is a very important parameter that largely determines the overall efficiency of the pump. Meanwhile, the amount of leakage in the flow part of the pump depends on the shape and size of the slot seals on the impeller. In this paper, the attempt to apply the well-known operational principle of a hydrodiode is made in order to reduce volumetric losses in the pump through a reduction in the flow rate coefficient of the slot seal, whose surface is profiled according to the principles of a hydrodiode.

AIM: Analysis of the possibility of utilizing the hydrodiode-like grooves on the surface of a slot seal in order to reduce the flow rate of liquid through the seal based on the computational fluid dynamics methods.

METHODS: The computational fluid dynamics method based on the solving of discrete analogs of the basic hydrodynamic equations is used in this paper.

RESULTS: The parameters of liquid flow in the slot seals with smooth surfaces, concentric grooves, and proposed profiled hydrodiodes in various sizes and shapes have been calculated. The flow rate coefficients for each type of seal have been determined, and comparative graphs have been built.

CONCLUSION: Based on the findings of this study, it can be stated that, overall, the use of hydrodiodes does not give significant advantages over the concentric groove with significantly increased complexity in manufacturing.

Keywords: centrifugal pump; numerical modeling; volumetric efficiency; slot seals; hydrodiode.

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Оригинальное исследование

Исследование возможности применения гидродиодов для повышения объёмного КПД центробежного насоса

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АННОТАЦИЯ

Обоснование. Для центробежных насосов, особенно с низкими коэффициентами быстроходности, очень важным параметром, в значительной мере определяющим общий КПД насоса, является величина объёмного КПД. В свою очередь, величина перетечек в проточной части насоса зависит от формы и размеров щелевых уплотнений на рабочем колесе. В данной работе сделана попытка применить известный принцип работы гидродиода для снижения объёмных потерь в насосе за счёт уменьшения коэффициента расхода щелевых уплотнений, поверхность которых спрофилирована по принципу гидродиода.

Цель работы — подтвердить или опровергнуть на основе методов гидродинамического моделирования возможность использования гидродиодных канавок на поверхности щелевого уплотнения для снижения расхода жидкости через него.

Методы. В данной работе применяется метод численного моделирования, основанный на решении дискретных аналогов базовых уравнений гидродинамики.

Результаты. Рассчитаны параметры течения жидкости в щелевых уплотнениях с гладкой поверхностью, с концентрическими канавками, а также с предложенными профилированными канавками в форме гидродиодов с различными размерами и формой. Определены коэффициенты расхода для всех типов уплотнений, построены сравнительные зависимости.

Заключение. На основании результатов статьи можно утверждать, что в целом применение гидродиодных канавок не даёт существенного преимущества над концентрическими канавками при существенно большей сложности их изготовления.

Ключевые слова: центробежный насос; численное моделирование; объёмный КПД; щелевые уплотнения; гидродиод.

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INTRODUCTION

Modern methods of hydrodynamic modeling are usually used to optimize the flow paths of centrifugal pumps to increase their energy efficiency. However, an important element of the flow path, which is not always given due attention, is the slot seals of the impeller and their geometry. In centrifugal pumps with low speed factors, volumetric losses in the slot seals have a significant impact on the overall efficiency and energy efficiency of the pump.

One method for increasing the volumetric efficiency of the pump is reducing the flow coefficient of the slot seal by making additional elements on its surface, such as concentric grooves, screw threads (Fig. 1), and some other configurations of gaps.

It was suggested that profiled grooves could be used whose shape follows the Tesla hydrodiode shape, on the seal surface [1] (Fig. 2). For testing this assumption, hydrodynamic modeling of various types of grooves in the slot seal was performed, whose initial geometry was taken from a real pump with known test results.

DESCRIPTION OF THE MATHEMATICAL MODEL

There are various methods for determining the flow rate through a slot seal of a centrifugal pump, one of which is the method described in [2] and [3], the essence of which is as follows:

1. Calculating the pressure drop on the slot seal H_s;
2. Calculating the Reynolds number Re in the slot;
3. Calculating the friction resistance coefficient of the slot λ_s;
4. Calculating the slot discharge coefficient μ_s;
5. Calculating the leakage through the slot Q_y.

The given method is semiempirical and is used together with the method of successive approximations, since a certain value of the slot discharge coefficient μ_s is specified first, and then steps 2–5 are repeated until the leakage value Q_y at the new iteration differs from the value from the previous one by no more than 1%.

However, due to the complex nature of the flow in the side cavities of the pump and the slot seal, whose geometry varies greatly depending on the pump, the technique described does not always provide results with acceptable accuracy. In this regard, a more advanced method based on computational fluid dynamics is required, which can be used not only to calculate the leakage value more accurately but also to analyze the flow nature.

MATHEMATICAL APPARATUS APPLIED

Hydrodynamic modeling is based on solving discrete analogs of the basic equations of hydrodynamics [3], [6]. In the case of noncompressible fluid, the Navier–Stokes equation is applied, which in vector form has the following form:

$$\frac{\partial \bar{v}}{\partial t} + \left(\bar{v} \cdot \nabla\right)\bar{v} = -\frac{1}{\rho}\nabla p + \vec{f} + \nu \Delta \bar{v}, \qquad (1)$$

where $\bar{v}$ is the velocity vector; p is the pressure; $\vec{f}$ is the vector of mass forces; t is time; ρ is the fluid density; ν is the kinematic viscosity of the fluid; Δ is the Hamilton operator; and ∇ is the Laplace operator.

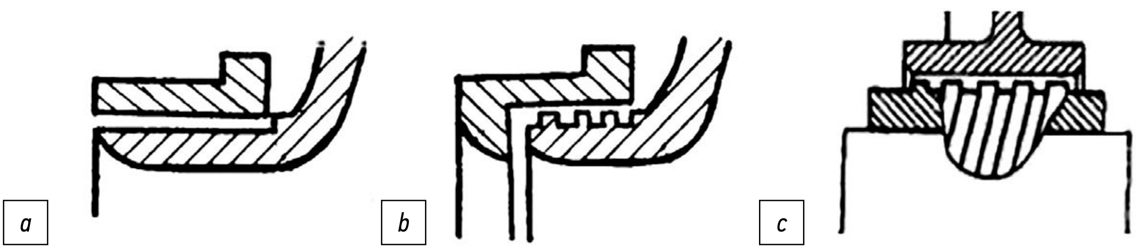

Fig. 1. The shape of the slot seal without additional elements (*a*), with concentric grooves (*b*) and with screw thread (*c*).
Рис. 1. Форма щелевого уплотнения без дополнительных элементов (*a*), с концентрическими канавками (*b*) и с винтовой нарезкой (*c*).

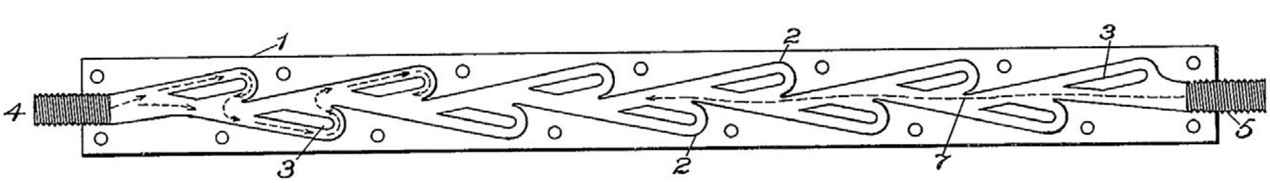

Fig. 2. The Tesla Hydrodiode.
Рис. 2. Гидродиод Тесла.

The continuity equation can be expressed as follows:

$$\nabla \cdot \vec{v} = 0 . \quad (2)$$

Reynolds averaging is performed to obtain discrete analogs of these equations [6]. As a result equations (1) and (2) take the following form (all equations are written using Einstein's method):

1. Navier–Stokes equations (Reynolds averaged):

$$\rho\left(\frac{\partial U_i}{\partial t}+U_i\frac{\partial U_i}{\partial x_j}\right)=-\frac{\partial P}{\partial x_i}+\frac{\partial}{\partial x_j}\left(T_{ij}^{(v)}-\rho<u_iu_j>\right), \quad (3)$$

where u_i is the instantaneous value of the projection of the vector velocity $\vec{v}$ onto the i th axis ($i=1,2,3$); U_i is the averaged value of the velocity u_i over the averaging period T; P is the averaged value of pressure; $T_{ij}^{(v)}=2\mu S_{ij}$ is the viscous stress tensor for an incompressible fluid; $S_{ij}=\frac{1}{2}\left(\frac{\partial U_i}{\partial x_j}+\frac{\partial U_j}{\partial x_i}\right)$ is the strain rate tensor; and $\rho<u_iu_j>$ are Reynolds stresses.

2. Continuity equation can be expressed as follows:

$$\frac{\partial U_j}{\partial x_j}=0 . \quad (4)$$

Reynolds averaging the Navier–Stokes equation makes the system of equations "unclosed," since in addition to the four unknowns (u_i, $i=1,2,3$ and p) another six unknowns are added in the form of Reynolds stresses $\rho<u_iu_j>$. As a result, a system of four equations with 10 unknowns can be obtained. Therefore, to close this system of equations, additional equations of turbulence models are introduced.

To solve the problem in this paper, a semiempirical model, namely, the shear stress transport (SST) turbulence model $k-\omega$ [9], was used, which introduces additional equations.

1. The turbulence kinetic energy transfer can be expressed in the following equation:

$$\frac{\partial k}{\partial t}+U_i\frac{\partial k}{\partial x_j}=P_k-\beta k\omega+\frac{\partial}{\partial x_j}\left[\left(\nu+\sigma_k\nu_{т}\right)\frac{\partial k}{\partial x_j}\right]. \quad (5)$$

2. The relative dissipation rate of this energy can be expressed as follows:

$$\frac{\partial \omega}{\partial t}+U_i\frac{\partial \omega}{\partial x_j}=\alpha S^2-\beta\omega^2+\frac{\partial}{\partial x_j}\left[\left(\nu+\sigma_\omega\nu_{т}\right)\frac{\partial \omega}{\partial x_j}\right]+ \\ +2\left(1-F_l\right)\sigma_{\omega 2}\frac{1}{\omega}\frac{\partial k}{\partial x_j}\frac{\partial \omega}{\partial x_j}, \quad (6)$$

where k is the kinetic energy of turbulence; P_k is the sum of the terms of turbulence energy generation taking into account the nonlinearity; α,β,γ are the closure coefficients; ω is the relative dissipation rate; ν is the kinematic viscosity; σ_k,σ_ω are the semiempirical model coefficients of the SST turbulence model $k-\omega$; $\nu_{т}$ is the turbulent viscosity specified using the SST turbulence model $k-\omega$ $\left(\nu_{т}=k\frac{\gamma}{\omega}\right)$; P_ω is the sum of the terms of generation of specific dissipation and cross-diffusion.

As a result, this study obtained a closed system of equations.

METHODS OF MODELING THE FLOW IN THE CLEARANCE OF A SLOT SEAL

Considering the complex geometry of the flow part of the pump, hydrodynamic modeling was first performed in a full formulation. The slot seal in this case was a conventional annular slot (Fig. 3).

Then, the computational mesh has the form presented in Fig. 3, where it is evident that the slot seal is represented by a prismatic directed mesh comprising four layers. This calculation was performed for the pump feed rates of 5, 20, 45, 70, 90, 110, 130, 150, 170, 190, and 210 m^3/h. The computational mesh consisted of 4.2 million cells. The boundary conditions for this calculation setting were the outlet pressure, which was set at the pump inlet, and the mass flow rate at the inlet, which was set at the pump outlet nozzle.

At the same time, due to the large number of calculation points and a rather fine mesh, the calculation took a long time (3.5 days) on a powerful computer (16 cores or 32 threads). In this regard, a simplified model was compiled, including only one slot seal, which, due to its symmetrical geometry, can be represented as a sector (Fig. 4). The essence of this method is that the values between the interfaces (indicated by the red and green dotted lines) are cyclically transmitted, resulting in an axisymmetric solution.

The sector was chosen equal to 45°. This reduces the number of calculation cells but also constructs a mesh in the slot itself with higher quality (Fig. 5).

The comparative results of the leakage value for this approach are presented in Fig. 6. The calculation dependencies are presented in the form $\mu=f\left(\bar{Q}\right)$, where μ is the slot flow coefficient, $\bar{Q}=\frac{Q}{Q_{nom}}$ is the dimensionless flow rate, Q is the pump flow rate, and Q_{nom} is the nominal pump flow rate.

As can be seen, it follows from the given error values that it does not exceed 5%, with the exception of two

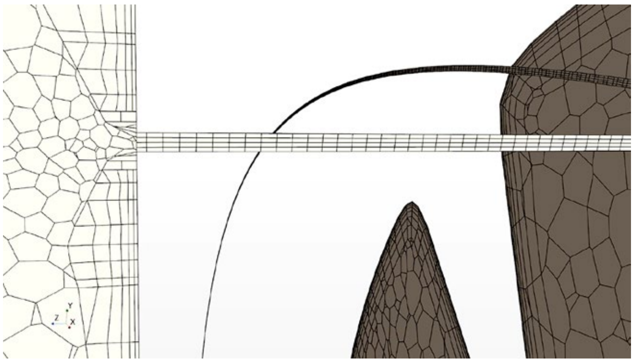

Fig. 3. The mesh for the calculation in full setup.
Рис. 3. Сетка при расчёте в полной постановке.

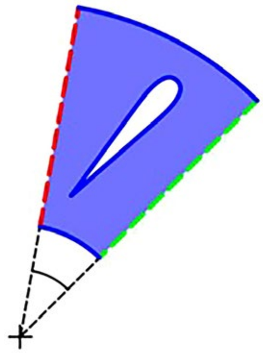

Fig. 4. The calculation method using the sector.
Рис. 4. Способ расчёта с использованием сектора.

points located to the left of the point corresponding to a relative flow rate of 0.2. This discrepancy in the extreme right points of the energy characteristic is caused by the complex vortex-like nature of the flow, which complicates the calculation while using simplified methods. However, given that the pump is rarely operated at this point of the characteristic, the proposed simplified method can be considered for calculating the flow in the slot of the slot seal, since in the rest of the feed range, the error remains within acceptable values for hydrodynamic calculations.

DESCRIPTION OF THE STUDIED GEOMETRY

To study the hydrodiode geometry, its 3D model was constructed, while the liquid model is presented in Fig. 7. The geometry is similar to that described in [10]. The grooves were annular (Fig. 8) and formed using two straight lines and a circle to which they are tangent (Fig. 9). For evaluating such geometry, a typical geometry of a slot seal with annular grooves (Fig. 10) was chosen for comparison. In addition, taking into account the influence of the hydrodiode groove size, an enlarged version was created, and the dimensions of its geometry are presented in Fig. 11.

This geometry, in addition to the changed dimensions, also differs in the presence of a rounding at the entrance to the annular groove, which was designed to prevent the separation of the boundary layer. The combination

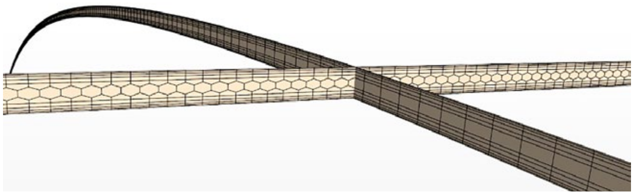

Fig. 5. The mesh for the calculation of the sector.
Рис. 5. Сетка при расчёте сектора.

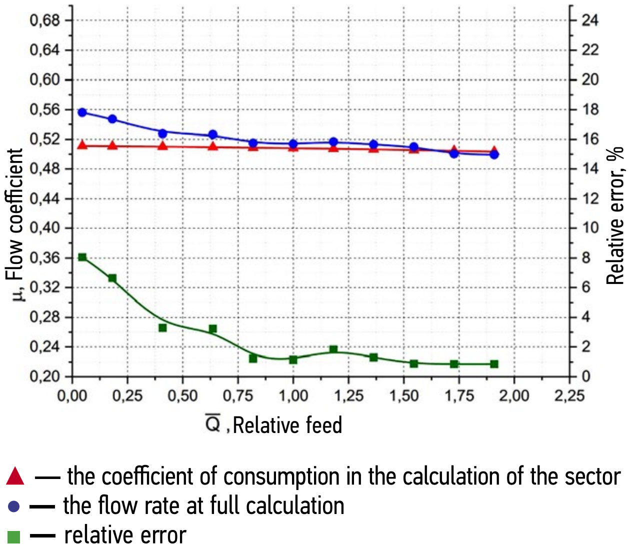

Fig. 6. Comparison of flow rate coefficients for the full calculation and the sector calculation.
Рис. 6. Сравнение коэффициентов расхода для полного расчёта и расчёта сектора.

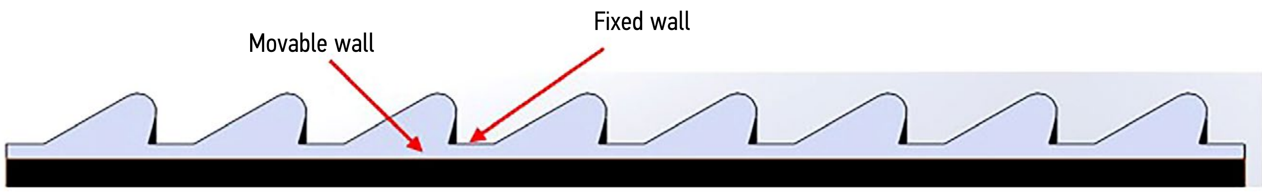

Fig. 7. A liquid model of a slot seal in the shape of a hydrodiode.
Рис. 7. Модель жидкости щелевого уплотнения в форме гидродиода.

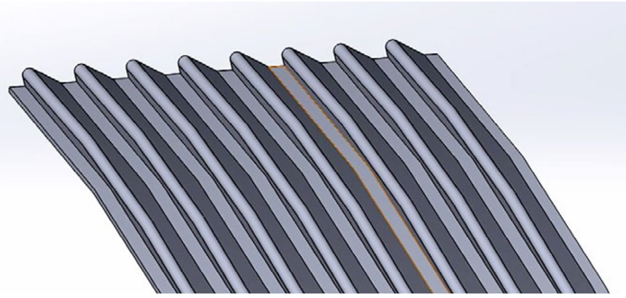

Fig. 8. The geometry of the grooves.
Рис. 8. Геометрия канавок.

of the moving and fixed walls is the same as in the geometry in Fig. 7. The influence of the location of the grooves on the stator or rotor was studied using the geometry presented in Fig. 12. The groove dimensions in the study are the same as that in the case in Fig. 11.

Using a sector as calculation geometry significantly improved the mesh quality. For the geometries presented in Figs. 7 and 10–12, the corresponding calculation mesh is presented in Figs. 13–16, respectively. The RANS turbulence model SST k–ω [7], [8], [9] was used for the calculations.

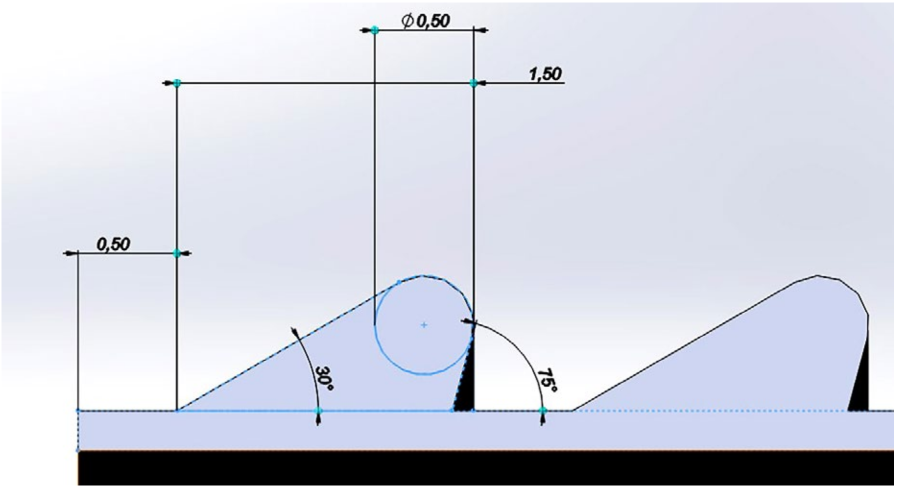

Fig. 9. The sizes of the grooves of hydrodiode.
Рис. 9. Размеры канавок гидродиода.

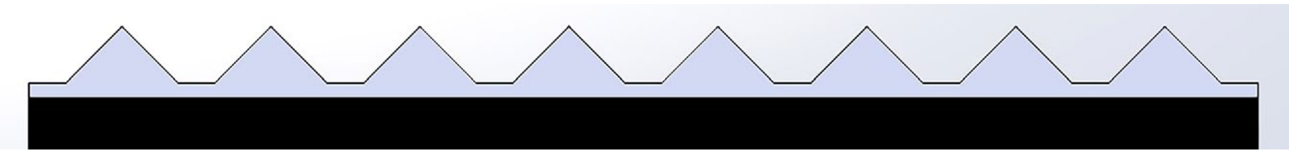

Fig. 10. The liquid model of the slot seal with annular grooves.
Рис. 10. Модель жидкости щелевого уплотнения с кольцевыми канавками.

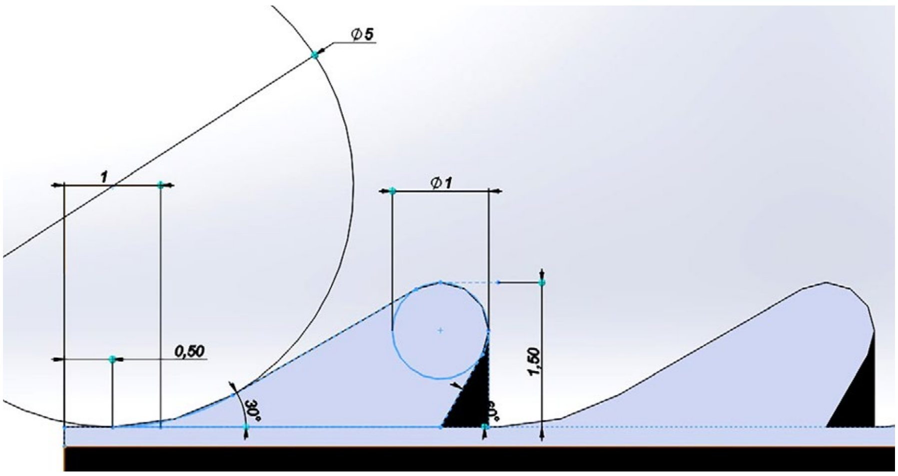

Fig. 11. The hydrodiode with increased groove sizes.
Рис. 11. Гидродиод с увеличенными размерами канавок.

RESULTS

The simulation results (i.e., vector fields of velocity distribution) are presented in Figs. 17–20. The velocity fields are constructed for the axial velocity component v_z. The calculation results are presented in Table 1, where Q is the pump flow; $Q_{hydrodiode}$ is the leakage at the seal in the form of a hydrodiode (Fig. 7); $Q_{standart}$ is the leakage at the seal in the form of a typical groove (Fig. 10); $Q_{hydrodiode\_boost}$ is the leakage at the seal in the form of an enlarged hydrodiode (Fig. 11). $Q_{hydrodiode\_boost\_reverse}$ is the leakage at the seal located on the rotor (Fig. 12). All flow values are given in m^3/h. The nominal flow rate for the pump is 110 m^3/h, for which the velocity fields are presented in Figs. 17–20. The results obtained from Table 1 are summarized in one graph presented in Fig. 21. The gap in the slit was $\delta = 0,2$ mm, and the diameter of the slit was $D_s = 184,6$ mm.

The analytically calculated Reynolds numbers Re at the slot inlet varied in the range from 14,800 to 14,100 with a flow rate of 5 m^3/h to 210 m^3/h, respectively.

CONCLUSION

1. According to the simulation, grooves on slot seals made in a shape similar to hydrodiodes do not significantly increase the volumetric efficiency of the pump, which, given the increased complexity of their execution and lower resistance to abrasive erosion, makes the idea insufficiently effective. This is due to the absence of a central body in the hydrodiode, which is impossible to make in a concentric groove, as well as the small dimensions of the groove, making it comparable in height to the thickness of the boundary layer in the slot, which, in turn, limits the possibilities

Fig. 12. The hydrodiode located on the rotor.
Рис. 12. Гидродиод, расположенный на роторе.

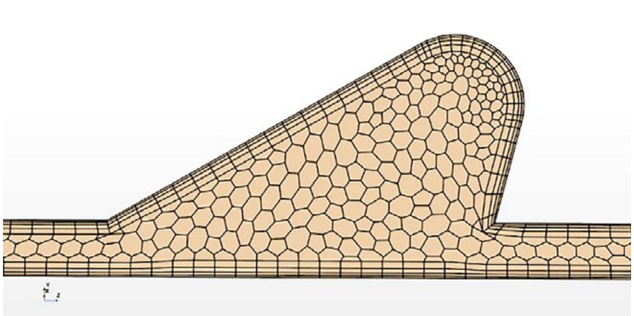

Fig. 13. The mesh of the hydrodiode.
Рис. 13. Расчётная сетка в гидродиоде.

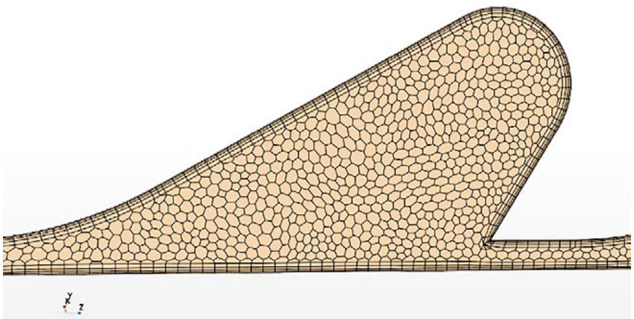

Fig. 15. The mesh of the enlarged hydrodiode.
Рис. 15. Расчётная сетка в увеличенном гидродиоде.

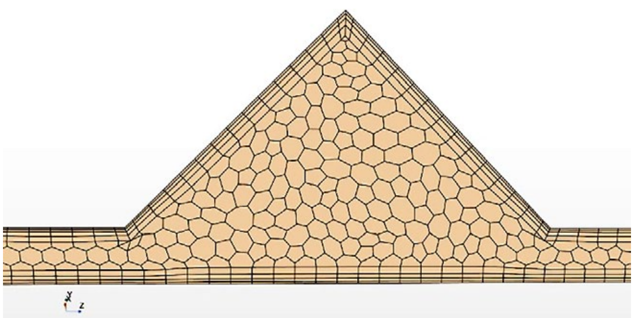

Fig. 14. The mesh of the standard groove.
Рис. 14. Расчётная сетка в стандартной канавке.

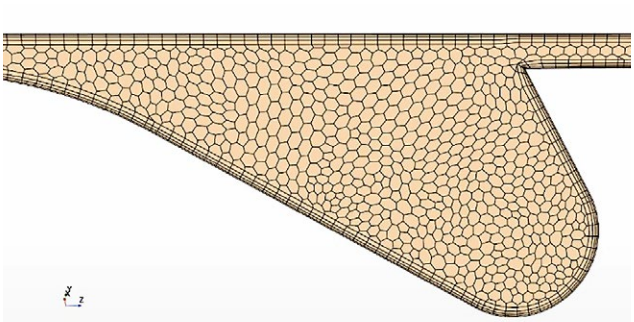

Fig. 16. The mesh of the hydrodiode on the rotor.
Рис. 16. Расчётная сетка в гидродиоде на роторе.

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Table 1. The results of calculations

Таблица 1. Результаты расчётов

Q	$Q_{hydrodiode}$	$Q_{standart}$	$Q_{hydrodiode\_boost}$	$Q_{hydrodiode\_boost\_reverse}$
5	6,295	6,488	7,215	7,337
20	6,258	6,45	7,175	7,293
45	6,19	6,381	7,101	7,214
70	6,115	6,305	7,017	7,127
90	6,05	6,238	6,947	7,052
110	5,981	6,168	6,872	6,971
130	5,907	6,092	6,789	6,884
150	5,828	6,012	6,702	6,792
170	5,743	5,925	6,614	6,693
190	5,655	5,834	6,514	6,589
210	5,56	5,737	6,41	6,477

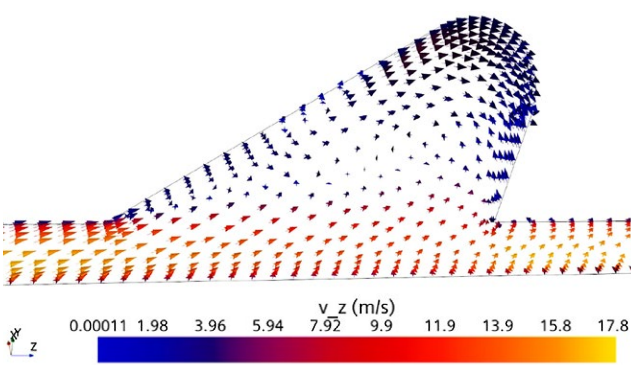


Fig. 17. Vector velocity field of the hydrodiode (change of velocity from 0 m/s to 18,8 m/s).

Рис. 17. Векторное поле скоростей в гидродиоде (изменение скорости от 0 м/с до 17,8 м/с).

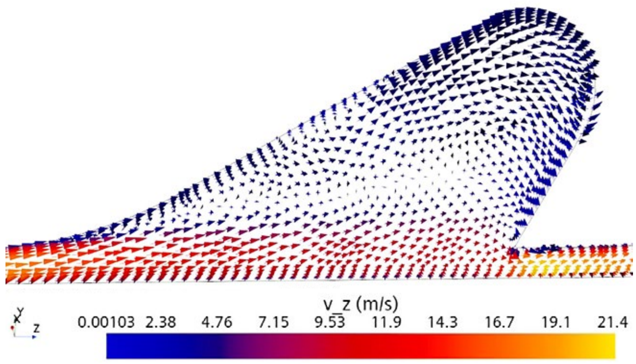


Fig. 19. Vector velocity field of the enlarged hydrodiode (change of velocity from 0 m/s to 21,4 m/s).

Рис. 19. Векторное поле скоростей в увеличенном гидродиоде (изменение скорости от 0 м/с до 21,4 м/с).

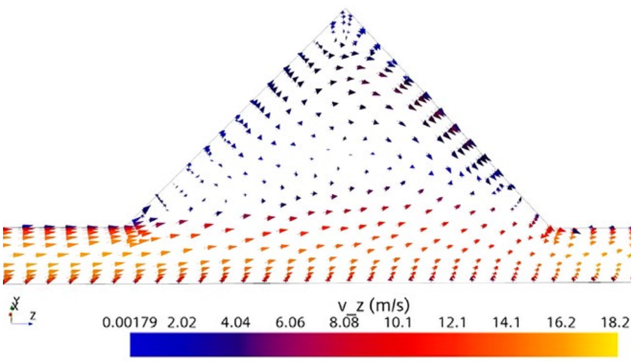


Fig. 18. Vector velocity field in a standard groove (change of velocity from 0 m/s to 18,2 m/s).

Рис. 18. Векторное поле скоростей в стандартной канавке (изменение скорости от 0 м/с до 18,2 м/с).

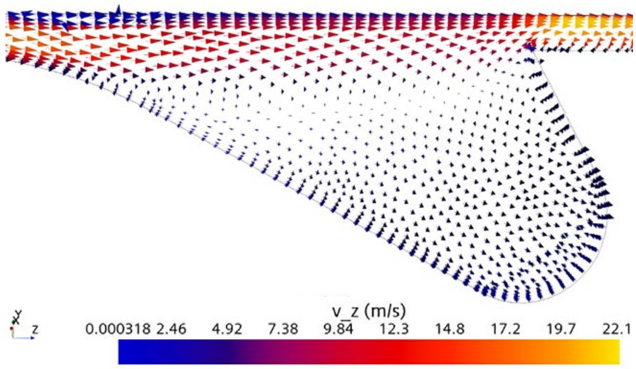
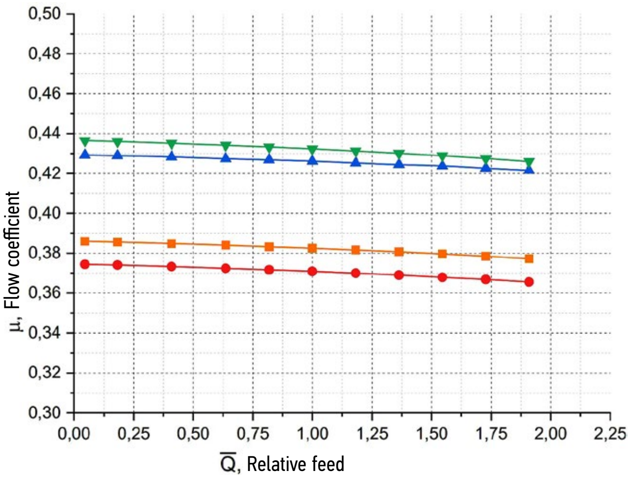


Fig. 20. Vector velocity field of the hydrodiode on a rotor (change of velocity from 0 m/s to 22,1 m/s).

Рис. 20. Векторное поле скоростей в гидродиоде на роторе (изменение скорости от 0 м/с до 22,1 м/с).



■ — the flow coefficient in the annular slot
● — the flow coefficient in the hydrodiode
▲ — the flow coefficient in the enlarged hydrodiode
▼ — the flow coefficient in the enlarged hydrodiode on the rotor

Fig. 21. Comparison of different slot seal options.
Рис. 21. Сравнение различных вариантов щелевого уплотнения.

for creating the necessary vortex structure opposite to the main flow in the groove.

2. Hydrodiode-shaped grooves can be used in slot seals of larger pumps, where the large dimensions of the grooves relative to the thickness of the boundary layer, to increase their efficiency.

ADDITIONAL INFORMATION

Authors' contribution. V. D. Fomenko — numerical modeling and research, preparation and writing of the manuscript; A.I. Petrov — problem statement, collection and analysis of literary sources, preparation and writing of the manusript; E.V. Efremov — author of the idea, preparation and writing of the manuscript. The authors confirm that their authorship meets the international ICMJE criteria (all authors have made a significant contribution to the development of the concept, research and preparation of the article, and read and approved the final version before publication).
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REFERENCES

1. U.S. patent 1,329,559; priority date Feb 21, 1916.
2. Rudnev SS, Matveev IV. *Methodical manual for course design of vane pumps.* Moscow: MVTU im NE Baumana; 1975. (In Russ.)
3. Lomakin AA. *Centrifugal and axial pumps.* Moscow, Leningrad: Mashinostroenie, 1966. (In Russ.)
4. Mikhailov AK, Malyushenko VV. *Vane pumps. Theory, calculation and design.* Moscow: Mashinostroenie; 1977. (In Russ.)
5. Patankar S. *Numerical methods for solving problems of heat transfer and fluid dynamics.* Moscow: Energoatomizdat; 1984. (In Russ.)
6. Loytsyansky LG. *Mechanics of liquid and gas.* Moscow: Nauka; 1987. (In Russ.)
7. Volkov KN, Emelyanov VN. *Modeling of large eddies in calculations of turbulent flows.* Moscow: FIZMATLIT; 2008. (In Russ.)
8. Wilcox DC. *Turbulence Modeling for CFD.* La Canada: DCW Industries, 1998.
9. Menter F. Two Equation Eddy-Viscosity Turbulence Modeling for Engineering Applications. *AIAA Journal.* 1994. Vol. 32. P. 1598–1605. doi: 10.2514/3.12149
10. Kondyurin AYu, Shcherba VE, Shalai VV, et al. Experimental Research Results of the Slot Seal Constructed as Hydrodiode for the Hybrid Power Piston Volumetric Machine. *Procedia Engineering.* 2016;152:197–204. doi: 10.1016/j.proeng.2016.07.691

СПИСОК ЛИТЕРАТУРЫ

1. Патент США 1,329,559; дата приоритета февраль 21, 1916.
2. Руднев С.С., Матвеев И.В. Методическое пособие по курсовому проектированию лопастных насосов. М.: МВТУ им. Н.Э. Баумана, 1975.
3. Lomakin AA. Центробежные и осевые насосы. М., Л.: Машиностроение, 1966.
4. Михайлов А.К., Малюшенко В.В. Лопастные насосы. Теория, расчёт и конструирование. М.: Машиностроение, 1977.

5. Патанкар С. Численные методы решения задач теплообмена и динамики жидкости. М.: Энергоатомиздат, 1984.
6. Лойцянский Л. Г. Механика жидкости и газа. М.: Наука, 1987.
7. Волков К.Н., Емельянов В.Н. Моделирование крупных вихрей в расчётах турбулентных течений. М.: ФИЗМАТЛИТ, 2008.
8. Wilcox D.C. Turbulence Modeling for CFD. La Canada: DCW Industries, 1998.
9. Menter F. Two Equation Eddy-Viscosity Turbulence Modeling for Engineering Applications // AIAA Journal. 1994. Vol. 32. P. 1598–1605. doi: 10.2514/3.12149
10. Kondyurin A.Yu., Shcherba V.E., Shalai V.V., et al. Experimental Research Results of the Slot Seal Constructed as Hydrodiode for the Hybrid Power Piston Volumetric Machine // Procedia Engineering. 2016. Vol. 152. P. 197–204. doi: 10.1016/j.proeng.2016.07.691

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