Research Article

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The investigation of the cavitation processes in the radial labyrinth pump

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Abstract: The paper concerns the analysis of the cavitation processes in the flow passages of the radial labyrinth pump. The object of the analysis contains the active (moving) and the passive (stationary) discs with straight channels trajectory and semi-circular cross-section. The conversion of the mechanical energy into hydraulic based on the exchange of the momentum between the liquid remaining in the moving and the stationary areas of the discs as well as on the centrifugal increase of the moment of momentum. The analysis of the cavitation processes was realized by the experimental research and the numerical simulation. In the article, the comparison of the cavitation characteristics was carried out. The numerical simulation had given similar results to the experimental one, the process of the cavitation was visualized. Furthermore, numerical investigations helped to describe the cavitation development. The results of the numerical research such as the distributions of the velocity, pressure and vapor volume fraction in the passages were presented. At first, cavitation starts on the back side and on the top of the wall between channels of the active disc. Further, the cavitation areas are growing along the axis of the channels. Eventually, they separation was observed and vortices of the vapour-gas mixture in the middle of the channels were formed. This phenomenon is so-called super cavitation vortices.

Keywords: radial labyrinth pump, cavitation performance, numerical simulations

1 Introduction

Labyrinth pumps are special flow machines characterized by the low value of specific-speed. The working process in such machines relies on the intensification of the turbulent fluid friction due to the presence of a large number of the blades or channels located on the working elements. The pumping performance is achieved by the exchange of the momentum, as the main operating process, between the liquid remaining in the stationary area and the moving pump impeller. For the first time, the description of the operation, structure and experimental studies of the labyrinth pump was presented by Golubev in 1961 [4]. The author distinguished two types of such pumps: axial and radial (Fig. 1). The improved calculation and design model, as well as new experimental results, were presented in [5]. In mentioned publications, the author focused mainly on axial labyrinth pumps for which he developed the theory of the working process. The approach to the design of the pumps with radial flow was not sufficiently recognized.

The basic theory of the flow of axial pumps may also be applied to radial labyrinth pumps assuming that change in the shape of working surfaces does not affect the process of energy conversion [4, 5]. Nonetheless, this is a very general assumption, because it does not take into consideration the centrifugal action of the liquid, which makes it necessary to complete the existing theory.

The theoretical model describes physical phenomena in the grooves of the axial labyrinth pump, and the investigations of the complex fluid phenomena are still developing. Golubev et al. [3] presented the improved method for calculating the head coefficient for different cross-sections of the channels. Andrenko et al. [2] presented the analysis of the existing design methods, improved the model of the operation process and developed an integral method for calculating pump flow characteristics, taking into account the shape of the screw channels. The mathematical model of the analyzed pump has also become the subject of works in [1, 8, 9].

Experimental results of the cavitation performance of the axial labyrinth pumps for different fluids were pre-

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sented in [5]. The results of the investigations confirm fallow correlation: for the smaller diameters of working elements, the NPSHR performance is more accurate (smaller). Lebedev [8] presented the method of the calculation of the cavitation criterion for the axial labyrinth pumps, determined by the influence of the structural dimensions and properties of the fluid.

Numerical modeling of the liquid flow in the hydraulic part of the axial labyrinth pump was identified by Lebedev [6]. As the result of this work, the construction guidelines for working elements were described. In spite of many laboratory tests and advanced knowledge of the operating process, the most effective method of the calculating a new labyrinth pump is a recalculation of the existing model pump by means of the laws of the hydrodynamic similarity. The labyrinth pumps with the radial flow have not been well recognized so far. Szulc [11] carried out the analysis based on the obtained results of the numerical simulation of the radial labyrinth pump and he concluded that despite the centrifugal action in such pump also the process of creating vortices in the space between the blades occurs. These tests were carried out for the pump with bladed discs, while the issues concerning the radial labyrinth pump with discs equipped with grooves were omitted. The conducted literature review showed the lack of the information concerns the numerical simulation of the cavitation performance in labyrinth pumps. Moreover, the cavitation process in the radial labyrinth pumps was not fully recognized so far, so appropriate measurements seems to be reasonable.

The main aim of the presented research is to develop the numerical model of the radial labyrinth pump with channel discs and to study the structure of fluid flow in these channels as well as the identification of the cavitation performance of analyzed unit.

2 The object of research

A radial labyrinth pump equipped with the straight trajectory of grooves was assumed as the object to be considered. The pump parameters were: nominal flow $Q_{nom} = 8.5 \text{ m}^3/\text{h}$, nominal head $H_{nom} = 20.5 \text{ m}$, rotational speed of the rotor $n = 2650 \text{ rpm}$. The hydraulic components of the pump were: active and passive discs on which the semi-circular shaped channels were made. The view of the discs is presented in Fig. 2.

The geometrical dimensions of both discs were the same: diameter of the entry to grooves $D_0 = 119 \text{ mm}$, outside diameter $D_2 = 180 \text{ mm}$, number of the grooves 50,

Figure 1: The example of the labyrinth pumps: a) with axial flow, b) with radial flow. Elements: 1 – sleeve, 2 – screw, 3 – passive discs, 4 – active disc

width and depth of the groove, respectively, $s_k = 4 \text{ mm}$ and $h_k = 2 \text{ mm}$. The inlet angle of the grooves equals $\beta_{1k} = 30^\circ$. The outlet cross-section of the passive disc has been closed. Prepared in this way discs were installed in the model pump and the relevant energy measurements were made.

Figure 2: The geometry of the tested set (3D model): a) active disc, b) passive disc. Description of the geometrical parameters: D_2 – outer diameter, D_0 – diameter of the entry to grooves, s_k – width of the grooves, h_k – depth of the grooves, β_{1k} – inlet angle of the grooves

3 The results of the experimental investigations

The consideration of the performance of the radial labyrinth pump began with the physical tests of the model unit with the radial flow installed on the specially prepared test rig. The three-dimensional model of the test rig is shown in Fig. 3.

The stand was built as closed. The pumping medium was clean water at temperature $t_w = 18^\circ\text{C}$. Water flows through the suction pipe 4 to the pump, where the hydraulic energy grew and finally the liquid was guided by the discharge pipe 3 to the tank 1. Behind the pump was installed a throttling valve 5, which allowed for smooth regulation of the operating parameters of the pump. The

Figure 3: Scheme of the test rig (3D model) used in course of the experimental tests: 1 – tank, 2 – radial labyrinth pump, 3 – discharge pipeline, 4 – suction pipeline, 5 – discharge throttling valve, 6 – cut-off valves, 7 – suction throttling valve, 8 – frequency converter, 9 – squirrel-cage motor, 10 – drainage, 11 – supplying pipeline, 12 – coupling cover, 13 – fan, 14 – manometer/vacuum gauge, 15 – electromagnetic flowmeter, 16 – absolute pressure meter, 17 – differential pressure meter, 18 – power meter, 19 – electronic tachometer, 20 – scale, 21 – power supply

stand was equipped with high-quality measuring equipment for measuring the differential pressure p_r , flow Q and rotational speed n . The power on the pump shaft was obtained indirectly by measuring the rotational speed and the torque. The stand was equipped with the option of the fluent regulation of the speed using a frequency converter. During the cavitation test, the flow was throttled by the valve 7.

Figure 4: The flow and the power consumption curves obtained in course of the experimental tests (test rig presented in Fig. 3) of the analyzed set of discs (Fig. 2): index “kan” – grooved passive disc, “gl” – smooth passive disc

The energy tests of the pump were carried out using the throttle method for constant revolution speed. As the first step of the research, the parameters obtained for the mutual cooperation of smooth passive and grooved active disc were investigated. As a result of the conducted research, the characteristics of the labyrinth pump were presented, which is shown in Fig. 4. The application of the passive disc with the lack of grooves leads to the drop in the value of head for a whole range of discharge alteration from $\Delta H = 9$ m for $Q = 0$ m³/h to $\Delta H = 6$ m for $Q = 11.2$ m³/h.

The characteristic of the power consumption is raising, the head curve is stable. Comparing the obtained results should be noticed the increase of the head H for the set with the passive disc equipped with grooves, which indicates the occurrence of the circulation phenomenon.

4 The result of the numerical simulations

4.1 Numerical model and its validation

The numerical simulations of the flow in the hydraulic part of the pump were realized using commercial ANSYS CFX software. The program allows, using the finite volume method, to iteratively solve equations of momentum, energy and mass conservation. The numerical analysis of the flow was made using the MRF (Multiple Reference Frames) model. Geometric parameters were put together in the form of three-dimensional solid models made in CAE software. The discretization process of the model was carried out using the ICEM CFD, where structural mesh was made by means of blocking method. The tests were carried out as steady and transient for specific boundary conditions, which were: mass flow at the inlet and static pressure at the outlet. The pumping medium was water at temperature 18 °C the calculation model is shown in Fig. 5.

The numerical model contained the following areas: Inlet pipe, space between disks, passive and active discs and vortex casing. The calculations were conducted with the assumption of periodic flow symmetry (circle section) and also the whole model was considered. The angular width of the periodicity sector was equaled 14.4° and contained 2 grooves of the active disc. Frozen rotor interface has been applied. The standard models of turbulence $k-\epsilon$ and $k-\epsilon$ RNG were used for the calculations. Simulations were turned off when the stabilization of energy parameters of the pump was achieved. The mesh was characterized by the parameters given in Table 1.

4.2 Quantity analysis of the flow

The results of the flow simulations of the radial labyrinth pump with grooved operating discs are shown in Fig. 6. All obtained flow curves are characterized by smaller steepness than the results of the real investigations, with the smallest differences near the best point.

When comparing parameters in the best point, the largest deviation from reality was observed for the model

Table 1: Mesh parameters of the calculated model (periodic)

Element	Number of nodes	Number of elements	Quality (ICEM)	Skew, °
Inlet pipe	152822	146880	0.7	> 58
Active disc	2073823	1977864	0.4	> 30
Passive disc	1516104	1419363	0.45	> 30
Vortex casing	1974656	1883558	0.5	> 30

Figure 5: The calculation models of the radial labyrinth pump dedicated to CFD investigations: a) the entire model, b) the periodic part of the model

with periodic symmetry using viscosity model $k-\epsilon$ for turbulence modeling and equals about 22% of the head obtained from the real experiment. For the $k-\epsilon$ RNG turbulence model, the difference was only 7%. When the whole model of the pump was considered the application of the $k-\epsilon$ turbulence model results in 14% difference in head and for the $k-\epsilon$ RNG turbulence model the difference was equaled 3%. The best results in terms of convergence with real experiments were obtained for unsteady simulations – 1%. In this case, and also for the other models, the steepness of the characteristic was different from the experimental data. This could be explained by the real experimental reasons. The discs dedicated for testing were made of the organic glass, which is characterized by high flexibility. This may lead to the situation in which, for the small flows, the large pressure is acting on the back wall of the active disc, which may cause a reduction of the clearance between cooperating discs. This situation could cause the increase of the head. Moreover, during the operation of the pump, due to its low efficiency, the flow elements could deform by means of the rise of the temperature. It may lead also to the reducing of mentioned clearance. The efficiency values are higher for the numerical simulations than real values, due to the simulation did not take into account the mechanical losses and the disc friction. Nevertheless prepared numerical models could be assumed as satisfactory.

Figure 6: The comparison of the flow and efficiency characteristic obtained during the real and numerical investigations (different viscosity models, steady and unsteady simulations)

4.3 Quality analysis of the flow

Based on the results of the numerical simulations, the flow structure in the grooves of the analyzed discs was identified. The results are presented in Fig. 7.

Figure 7: The views of the absolute and relative streamlines coloured by velocity: a) $k-\epsilon$ turbulence model, b) $k-\epsilon$ RNG turbulence model

In the grooves of the active disc, there is a centrifugal character of the flow along the trajectory correlated with the vortex movement. In the channels of the passive disc, the flow structure is a vortex, directed centripetally. When the water approaches the exit from the discs, the closed outlet cross-section of the passive disc and the raise in the distance between adjacent grooves increases the intensity of the vortices. The main aim of the application of the passive disc is not transport of the liquid, but the intensifica-

tion of the momentum exchange between the fluid parts enclosed in the movable and stationary part of the working area of the pump. This is confirmed by the distribution of the velocity vectors along the channels of the active disc, which is presented in Fig. 8. They demonstrate the formation of the vortex structures in the channels of the active disc. After the mutual interaction of the two grooves, the speed increases, which confirms the existence of momentum exchange between the liquid located in the active and passive discs. As a consequence, the centrifugal action, which causes the increase of the moment of momentum is superimposing on the exchange of momentum between the channels of the active and passive disc, which results in a higher value of the head than in the case of the cooperation of the grooved active disc with the smooth passive one [10].

Figure 8: The distribution of the velocity vectors coloured by the velocity in the disc channels (cross-section through the active disc channel axis) for $k-\epsilon$ RNG turbulence model

Figure 9: The distribution of static pressure in the discs channels for $k-\epsilon$ RNG turbulence model

In course of the increase of the pump discharge, the flow in the active disc increases, as well as speed. In the passive discs, the turbulence also strongly increases, and the flow structure alters at the entrance to the grooves. In

the clearance between the discs, a stream of fluid flows from neighboring grooves, but it does not exchange energy between the discs in a useful way, it also leads to pressure equalization which reduces the total efficiency of the pump. This stream is a volumetric loss. Another loss is the reverse flow in the passive disc. Conducted experiments and numerical simulations proved the positive effect of the passive disc, leading to the increase of the head, unfortunately with lower efficiency.

In Fig. 9 the distribution of static pressure in the active and passive disc for the $k-\epsilon$ RNG turbulence model was presented. Low-pressure zones at the entrance to the grooves of the active disc are potential places for the cavitation. The creation of this zones is caused by local speed increase. It could be noticed that the structure of this areas is similar to the vortex along the main axis of the channel. Moreover, the zones with local pressure drop in the grooves of the passive disc could be observed (in the points of contact with the active disc channels), which is another confirmation of the existence of the momentum change between the analyzed elements.

5 The results of the cavitation investigation

As part of the experimental tests, when the pressure on the suction side is insufficient, the occurrence of the phenomenon of cavitation, in the strongly developed phase, was noticed. In addition, due to the application of the transparent pump casing, the vortices of the steam/gas mixture were visible in the channels of the passive disc. For some condition, the supercavitation was observed (fig. 10). Because of this, the additional NPSH tests were carried out. The main aim of the investigation was to identify the cavitation performance of pump and compare the real and numerical outcomes.

The beginning of the cavitation effect was identified as a drop in the head of 3%. For the numerical calculation, the Rayleigh-Plesset equation and the standard $k-\epsilon$ turbulence model was used. Boundary conditions were set as the total pressure at the inlet and the mass flow rate at the outlet. The pumping medium was water at temperature 18 °C. The saturated steam pressure was set as 3167 Pa.

The investigation of the cavitation performance of the analysed pump was realized by the regulation of the pressure at the suction sided of the unit. The comparison of the cavitation, obtained by means of the experimental test and the numerical simulation for the flow rates (0.7; 1.0; 1.3) Q_{nom} are presented in Fig. 11.

Figure 10: The cavitation in the grooved of the passive discs: a) no cavitation pump operation, b) supercavitation – enlarged view represents the vortex of super cavitation

The analysis of received outcomes shown, that the cavitation performance curves are similar for both methods. The slight differences in the magnitude of the head drop are caused by the difference of the steepness of the pump characteristics. The values of the head for NPSH3% are 4.3 m, 4.8 m and 6.9 m for numerical simulation and 5.2 m, 6.6 m and 5.4 m for experimental test respectively for the flow rate (0.7; 1.0; 1.3) Q_{nom} . In the Fig. 11d the difference in the value and shape of the NPSHR curve could be observed. The main reason of such convergence could be the application of k-ε turbulence model. It is important to mention that the experimental investigations were made for quasi constant fluid temperature, and this could have an impact on achieved results. It is possible to see, that the head is not constant at the beginning of the experiment (Fig. 11a, and Fig. 11c). Nevertheless, the shape and value of achieved outcomes are similar, and the maximum deviation is acceptable, so the analysis of model could be used for the visualization and understanding of the creation of the vapour zones, which is desirable.

Figure 11: The comparison of the cavitation performance curves obtained by means of the experimental and numerical tests for different flow rates: a) 0.7 Q_{nom} ; b) 1.0 Q_{nom} ; c) 1.3 Q_{nom} ; and external characteristic: d) NPSH3% curves

In Fig. 12. was presented the process of the cavitation vortex development in the grooves of cooperating discs for Q_{nom} . The presentation of archived results was made for different values of suction pressure.

When decreasing of the suction pressure, at the beginning of the cavitation process the vapour zones starts to occur on the back side of the grooves and on the top of the wall between channels of the active disc (Fig. 12c). Further, the cavitation areas are growing along the axis of the channels (Fig. 12f). Eventually, they separation was observed and vortices of the vapour-gas mixture in the middle of the channels were formed. This phenomenon is so-called super cavitation vortices. Comparing the result of the numerical and real experiments the view of the flow is similar. In the axis of the groove is visible the region of lower pressure, which fills with the mixture of the gas and water vapour.

Figure 12: The process of the cavitation vortex development in the passages of the radial labyrinth pump for NPSH equals: a) 20.1 m; b) 16 m; c) 9.9 m; d) 6.8 m; e) 5 m; f) 3.8 m; g) the comparison of the cavitation performance curves obtained by means of the experimental and numerical tests – the letters described points correspond to the view of the cavitation structures presented in figures a–f

6 Conclusions

Based on the results of experimental studies and numerical simulations, the following conclusions could be drawn:

- For the flow simulation in the radial labyrinth pump, it is necessary to use transient calculation methods,

characterized by the smallest deviation from the experimental data.

- The $k-\epsilon$ RNG turbulence model shows better convergence results with real data than $k-\epsilon$ and is more suitable for such machines.
- The results of the simulations could be treated qualitatively, while they do not fully reflect the quantitative aspect of the examined parameters. The reasons for this fact should be found in test rig reasons.
- Carried out simulations confirm the existence, besides increasing the moment of momentum, the momentum exchange as two ways of exchanging energy in the analyzed pump. This process can be called hybrid as well as the impeller.
- The obtained results of the flow structure expanded the understanding of the operating process in the radial labyrinth pump. The realized numerical model allows to calculate the labyrinth pump operating parameters to further improvement of its design.
- The flow separation and vortex structure at the inlet to the active disc indicate the occurrence, under appropriate conditions, of a cavitation rope vortex.
- The application of the numerical model with the Rayleigh-Plesset equation for the modeling of the cavitation is acceptable due to the appearance of the supercavitation vortices that was discovered during the experimental research.
- The ways to reduce the probability of occurrence of the cavitation are scheduled for the further analysis.

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