

Research Article

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Analysis of the influence of a stator type modification on the performance of a pump with a hole impeller

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Abstract: The article presents the results of numerical analyses and experimental research of the influence of various types of stators on a liquid flow through a centrifugal pump with a hole impeller. It is a continuation of authors research of cooperation pump stators with alternative types of impellers which work in ultra low specific speed. Hole impellers have become a significant alternative to classical ones in a range of extremely low specific speed $n_q < 10$. The aim of the research is to verify the quality as well as quantity of computer modeling results, and to estimate accuracy by examining the impact of a grid and a turbulence model with which the numerical simulations reflect the actual flow. Knowledge concerning construction of hydraulic elements of centrifugal pumps working in the range of parameters corresponding specific speed ($n_q < 10$) is insufficient. The outlet elements were tested in various configurations of constructional features. The complexity of the construction of the stator can significantly affect the manufacturing costs of pump unit.

Keywords: Low specific speed, centrifugal pump, hole impeller, stator type, CFD

1 Introduction

Pumps are the most commonly used among energy consuming devices in the industry. It is estimated that the

transport of liquid in the economy absorbs 20-30% of energy production. Energy savings in pumping systems can be significant, there are estimates of the amount of possible energy savings in all areas of pump use at the level of 40% [1]. In many installations improper working conditions are imposed (for example too high pressure). In some systems, the pump units are old and work with low efficiency. This forces a constant research for new construction of pumps parts – for example impellers or stators. It is necessary to increase the efficiency.

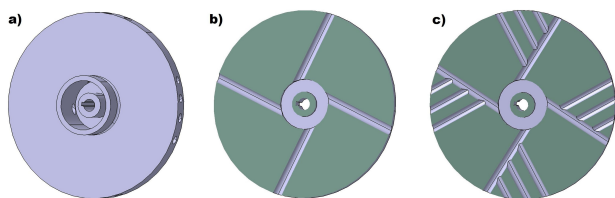
Wanting to improve the design of centrifugal pumps – especially working in ultra low specific speed – a better understanding of the flow phenomena of such machines is required. This paper discusses with the experimental and numerical study of the flow in the two types of pump stators – annular casing and spiral casing – which works with hole impeller. To a certain extent, it is a continuation of authors research [2] of cooperation pump stators with alternative types of impellers which work in ultra low specific speed. Similar research have been performed in literature [3–7]. But in this studies have been used blade impellers in different types and configurations. On the basis of these papers can be formulated fair conclusion, that the role of the geometry of outlet element becomes particularly important when considering innovative and unique constructions of the centrifugal rotor [8–10]. An good example of such a construction is a hole impeller which works in a range of ultra low specific speed $n_q < 10$ (def. of n_q according [1]), whose construction details were presented in Figure 1. It is a construction which uses a classical centrifugal liquid flow through the internal passages of the rotor to convert mechanical energy to hydraulic energy.

Due to a small outlet surface, located periodically along the circumference of the rotor, the construction of the stator cooperating with such a rotor is extremely demanding. To enhance conditions of that cooperation additional branch channels are sometimes applied in hole impellers (Figure 1b).

From a hydraulic point of view centrifugal pump consists of two, main elements: an impeller and the outlet

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element – stator. Rotor is responsible for converting mechanical energy of the electrical motor to hydraulic energy which is transferred to liquid. Pump stator collects liquid flowing out of the rotor and carries it to the outlet flange of the pump (single stage pump) or to the inlet of the next stage impeller (multistage pumps). Also, it converts kinetic energy of the liquid – after the impeller outlet – into potential energy. The efficiency of this alteration strongly determines performance of the whole pump. The last feature of pump stator: geometry of flow area affects the BEP (Best Efficiency Point) location on the pump flow characteristic. The outlet element is, therefore, a very crucial component, whose type and construction largely influence pump's performance, the shape of flow characteristic, BEP location as well as efficiency level, as show the literature [2–7].

The aim of the following publication is to investigate the influence of the type and construction of the stator, co-operating with a hole impeller with branch channels, on performance parameters and to recognize flow phenomena occurring in the pump using the earlier mentioned hydraulic elements. A volute type casing as well as an annular type casing at various configurations of construction details of the mentioned elements were taken into consideration in the paper. The main research methodology were the CFD numerical analyses. Their results were experimentally verified on an appropriately built test rig.

An interesting conclusion has been discussed in the literature [1]. The authors claims that for centrifugal pumps with low and ultra low specific speed it is better to apply a channel of a constant cross-section and simple geometry than an another stator – for example volute casing. As a main disadvantage of annular stator one can assume the fact that the streams of various speed are mixed, which in turn leads to higher hydraulic losses. The difference in efficiency between a annular channel and spiral casing diminishes, with the decrease of n_q . The authors suggest that at specific speed – $n_q < 12$ – the constant cross-section channel could represent higher efficiency than volute casing. This fact has been proved in the research presented in [2]. The results confirmed higher efficiency of centrifugal pump with multi-piped impeller in cooperation with an-

nular casing. Also, in this article has been verified this interesting claim.

2 Object of research

A basic object of the experimental and numerical analysis is a hole impeller with branch channels, whose geometry and dimensions were presented in Figure 1c. The impeller consists of additional branch channels which – as was assumed – increase uniformity of velocity and pressure distribution in the stator. Therefore, they improve operating parameters. Geometric measurements of the impeller were shown in Figure 2a. A basic stator was an annular type casing. Dimensions were presented in Figure 2b. The outlet element was designed in a way which would allow a seamlessly exchange of pump flow parts.

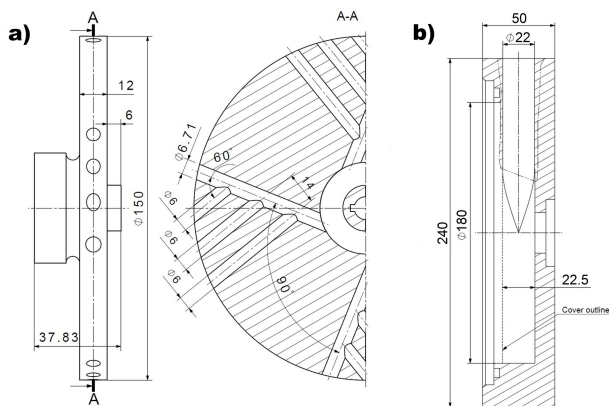


Figure 2: Dimensions of (a) basic hole impeller and (b) basic annular casing

3 Test rig

In order to verify the results of numerical calculations experimental tests on a test rig (Figure 3a) were planned and conducted. The main element of the test rig was a module, base pump (Figure 3b). The following hydraulic components were applied for the test: a hole impeller and an annular-type casing, shown in Figure 2, later referred to as the base configuration. The test rig is so versatile that it is possible to examine various types of centrifugal impellers on it. Similar experimental tests have been performed in [2].

The measurements at the test rig were totally automated. The control software was customized for test rig in compliance with the European Standard [11]. More details concerning measuring instruments of the test rig as well methodology of measurements can be found in [2, 12].

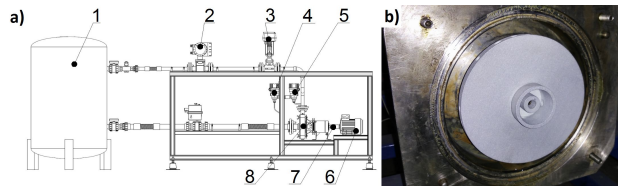


Figure 3: Presentation of test rig – (a) Test rig scheme and (b) the model pump without the front panel. Description of test rig scheme: 1 – tank, 2 – flowmeter, 3 – ball valve, 4, 5 – pressure detectors, 6 – electric motor, 7 – flexible coupling, 8 – pump

The results of experimental tests of energetic characteristics of the module pump in the base form were shown in Figure 4. The Best Efficiency Point (BEP) of researched base pump with hole impeller is characterized by the following operating parameters:

- capacity of pump $Q = 5.1 \text{ m}^3/\text{h}$ ($v_{st} = 5.7 \text{ m/s}$, Re was about $62 \cdot 10^3$),
- head pump $H_u = 21.8 \text{ m}$,
- total efficiency $\eta_c = 31.5\%$.

This values will be used to build and verification a CFD model of pump with both stator types.

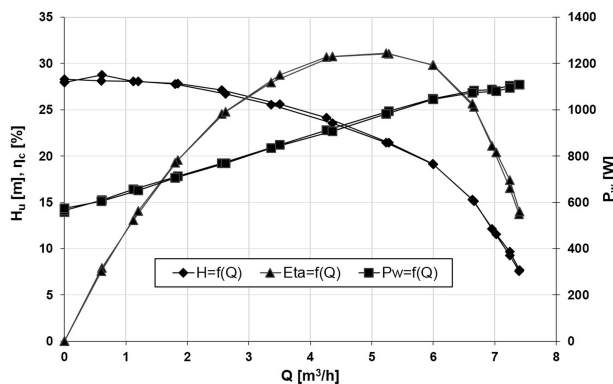


Figure 4: Characteristics of the basic pump with an annular casing and a hole impeller

4 Numerical modeling

A fundamental research method applied in the project was the CFD (Computational Fluid Dynamics) [13]. Geometry of a stator and a hole impeller applied in the test rig module pump was created as a 3D model. Later, it referred to as the base configuration of liquid model of pump – named VB. Algorithm of numerical research was based on change type and geometrical parameters of stators cooperating with the impeller, however geometry of the rotor remained unaltered. Geometry of base configuration of annular casing was designed according to one dimension theory of vortex pump, included in literature [1].

4.1 Numerical model

To determine and identify flow phenomena which take place during the flow in the pump with hole impeller with branch channels, for numerical CFD analyses Ansys Fluent from Ansys Workbench 16.0 package was used. A simplified geometrical model of a base pump used for discretization was presented in Figure 5, and it consists of flow volumes:

- The inlet element (the length of 4 diameters),
- The impeller,
- The annular-type casing,
- The outlet element.

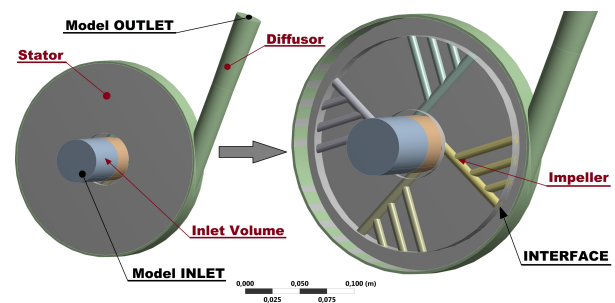


Figure 5: Geometrical model of the pump with an annular casing and a hole impeller for numerical calculations

The simulations were performed for the following settings:

- Liquid – clean water of 20°C,
- double precision Solver,
- calculation scheme: PISO algorithm [13–16],

- for all equations the second order scheme discretization was assumed;
- convergence criterion for each equation was $\epsilon = 10^{-6}$;

The CFD calculations were performed as transient, time step determined in accordance with [16] was $t_s = 5.2 \cdot 10^{-4}$ s. Rotational speed of the hole impeller was $n = 2870$ rpm (the speed of the electrical motor shaft). One rotation of an hole impeller has been divided at 120 time steps. Other boundary conditions were defined compliant with Figure 5, as:

- Inlet model – inlet velocity corresponding to given efficiency of the pump, the intensity of turbulences at the inlet was determined $T_i = 4\%$.
- Walls – zero pressure gradient $dp/dn = 0$, velocity $u_x = 0$.
- Outlet model – constant mass flow and liquid viscosity, static pressure corresponding roughly to the obtained head of the pump $p = 300$ kPa, intensity of reverse flow turbulence $T_{ibf} = 2\%$.

4.2 Computational grid

To determine an optimal size of a grid The Grid Independence Test was performed. Calculations for four variants of grids were made, whose parameters were presented in Table 1. In each case, the tetrahedral type elements were used to discretization the grid. Boundary layer model was applied and concentrated in the areas of walls of the impeller and stator, which was shown in Figure 6. To Grid Independence Test $k-\epsilon$ Standard model was used.

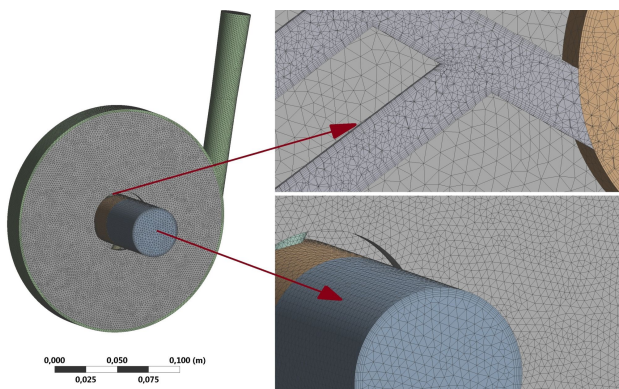


Figure 6: Computational grid with close-up area of high elements concentration

In order to expand the meaning of the headings used in Table 1 should be mentioned that:

- The aspect ratio of an finite element in 3-D describes the proportional relationship between radius of circumscribed and inscribed circle at this element.
- Skewness is one of the primary quality measures for a mesh. Skewness determines how close to ideal (i.e., equilateral or equiangular) a face or element is.
- The Element Quality option provides a composite quality metric that ranges between 0 and 1. This metric is based on the ratio of the volume to the sum of the square root of the cube of the sum of the square of the edge lengths for 3D elements. A value of 1 indicates a perfect cube or square while a value of 0 indicates that the element has a zero or negative volume [15, 16].

To assess the impact of the grid size on accuracy and time of calculations the following alterations of values were analyzed: moment on the blades (M_t) and the head (H), determined by the equation (1). The calculations results can be found in Figure 7.

While analyzing the results found in Figure 7 it can be noticed that the biggest difference exists between the variants 1 and 2 (more than 23% for H and 6% for M_t). Between 2 and 3 the differences in values do not exceed 13% for H and 2.9% for M_t . However, the distinction in results between the grids 3 and 4 amounts to 2.7% for H and 0.8% for M_t . For further calculations the computational grid corresponding variant 3 was assumed (element size was 0.125 mm for stator and 0.15 mm for impeller).

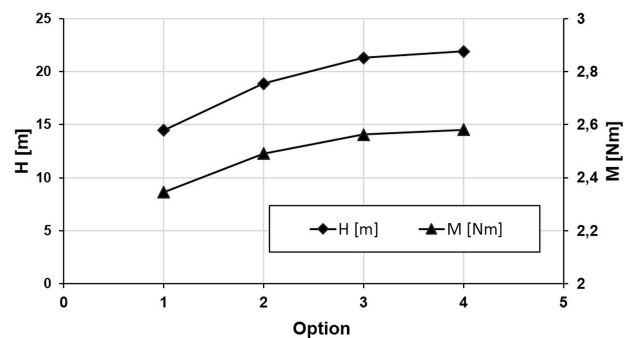


Figure 7: Influence of the grid size on the value of key computational parameters

4.3 Turbulence model

To choose an optimal turbulence model [15, 16] for the discussed flow calculations for all two-equation turbulence models available in Fluent software were done. The calcu-

Table 1: Properties of computational grids for each variant

Variant	Number of finite elements	Skewness (average)	Aspect ratio (average)	Element Quality (average)
1	6 754 554	0.23	2.6	0.65
2	9 597 516	0.22	2.18	0.67
3	14 192 358	0.22	2.12	0.68
4	19 453 674	0.22	2.06	0.74

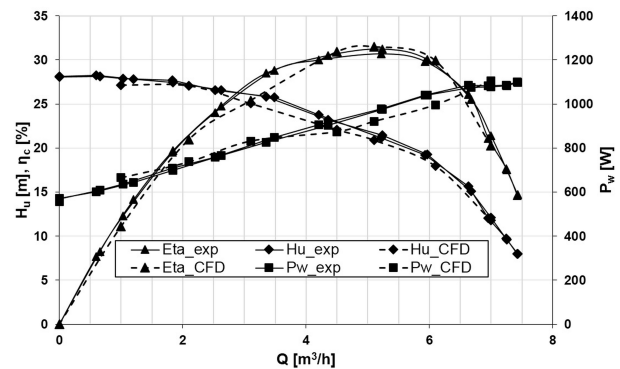
lations were performed for the computational grid variant 3 at a constant flow rate equal $Q = 5.1 \text{ m}^3/\text{h}$. The results are shown in Table 2.

Convergence of the solution was attained for all available versions of the two-equation models. Model $k-\omega$ SST reached convergence criterion the fastest (after 683 iterations). The value of H_u and total efficiency η_c obtained during the experimental tests amounts to $H_u = 21.8$ and $\eta_c = 31.5\%$. Obtained operating parameters for $k-\omega$ SST model were closest to experimental values. It can be inferred that the best accuracy was obtained for the turbulence model $k-\omega$ SST, which appears to be the best choice for the analyzed class of flows. In chosen model the $y+$ parameter on rotating surfaces does not exceed 1.

4.4 Validation of numerical model

To evaluate accurateness of the assumptions for the CFD model of analyzed pump, the validation of results in the whole range of a pump characteristic was performed. The experimental test results and numerical calculations results was compared. Figure 8 shows a comparison of characteristics obtained on the test rig with the ones obtained numerically. While analyzing the results the following conclusions can be drawn:

- In the whole area of a base pump – with a hole impeller and annular casing – the difference between the numerical results and experimental ones does not exceed 6%.
- For the flow rate corresponding BEP the difference between the experimental test results and numerical ones does not exceed 1.1%.
- For the flow rate higher than $6 \text{ m}^3/\text{h}$ a drop of the flow characteristic can be noticed. That is likely to be a result of additional transient phenomena, such as cavitation, which were not modeled numerically. Similar flow characteristics were obtained in the work [13].

**Figure 8:** Comparison of characteristics determined numerically and experimentally

4.5 Numerical study

The verified numerical model was applied in the tests on the influence of the type and construction of the stator cooperating with a hole impeller with branch channels on the pump's flow rate.

Main energetic parameters of the tested pump were computed with the following equations:

$$H_u = \frac{p_{cout} - p_{cin}}{\rho g} \quad (1)$$

$$P_w = M_t \omega \quad (2)$$

$$\eta_h = \frac{P_h}{P_w} = \frac{\rho g Q H}{M_t \omega} \quad (3)$$

$$\eta_c = \eta_h \eta_v \eta_m \quad (4)$$

Total efficiency of the tested pump (4) was determined assuming constant values of volumetric efficiency $\eta_v = 0.92$ and mechanical efficiency $\eta_m = 0.90$ – in accordance with previously conducted research in [2, 13] at the same test rig. The hydraulics efficiency is the key factor mainly for pumps working in $n_q < 10$. Mechanical efficiency it depends on the losses in bearings and sealing (losses can be treated as constant value). Volumetric efficiency especially depends on the pressure difference in an impeller gap sealing. It is unchangeable value in tests, so the volumetric efficiency can be treated as constant as well.

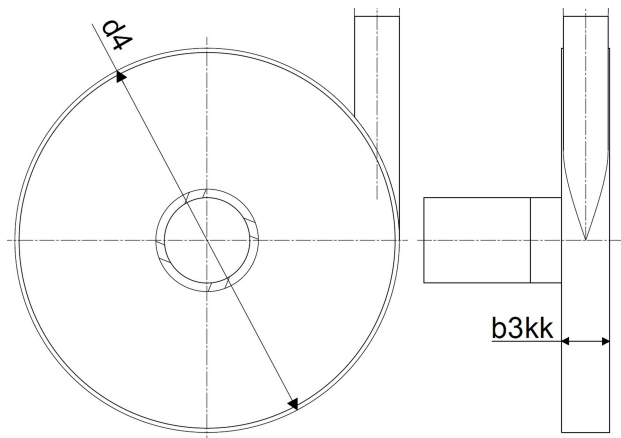
Table 2: Influence of the turbulence model

Lp.	Model	Convergence	Number of iterations	H_u	M_t	η_c
	-	-	-	m	Nm	%
1	k- ω Standard	YES	823	21.28	2.563	31.75
2	k- ω SST	YES	683	21.56	2.564	32.15
3	k- ϵ Standard	YES	688	20.98	2.614	30.69
4	k- ϵ RNG	YES	712	21.11	2.583	31.25
5	k- ϵ Realizable	YES	856	21.02	2.621	30.67

4.6 Cooperation of a hole impeller with an annular-type casing

To analyze the cooperation of a hole impeller with an annular-type casing at various configurations of construction features of the casing numerical calculations of the flow through the pump were performed. The following geometrical parameters of the annular type casing were then changed (Figure 9):

- The outer diameter d_4 of the channel - between 160 mm and 186 mm; base value 180 mm,
- The width b_{3kk} of the cross-section - between 16.5 mm and 26.5 mm; base value 22.5 mm,
- The cross-section shape - rectangular, semi-circular, trapezoidal.

**Figure 9:** Basic dimensions of the annular type casing

The results of the calculations can be found in Figure 10. Table 3, on the other hand, shows the results obtained after a change of the cross-section shape of a channel's profile. The tests were performed for a constant flow rate, corresponding BEP ($Q = 5.1 \text{ m}^3/\text{h}$). A variant of maxi-

mum efficiency (VM) was marked with a vertical solid line, a base variant (VB) was marked with a dashed line.

While analyzing the results of numerical calculations one can observe:

- A reduction of the diameter d_4 of 8% in relation to the base variant caused a relative increase of total efficiency of 9 percent point and the head of almost 17%, which resulted in a decline in power demand of more than 7%.
- A decrease of the width of b_{3kk} channel of nearly 18% in relation to the base variant caused a relative growth of total efficiency of about 4.6 percent point and a slight increase of the head – of 3.5%.
- A change of geometry of the flow area had a essential influence on the betterment of flow parameters. The size of the flow area affects at direction and value of velocity vectors and pressure distribution of the liquid throughout the all over width of the annular casing. In Figure 15 one can notice that the decrease of the flow area caused the increase of velocity, nonetheless, the distribution of velocity in the channel is definitely less uniform.
- The base shape the channel cross-section (rectangular) appeared to be definitively the best solution in comparison to the trapezoidal or semi-circular one – Table 3. That might be caused by the size of the cross-section area. Both the trapezoidal and semi-circular channels significantly increase the surface area, which, as previous tests indicated, deteriorates energetic parameters of the pump. Moreover, in Figure 11 one can observe strong recirculation and turbulence in the cross-section of the channel. This is the result of a violent change of the direction of the liquid flowing out of the rotor.

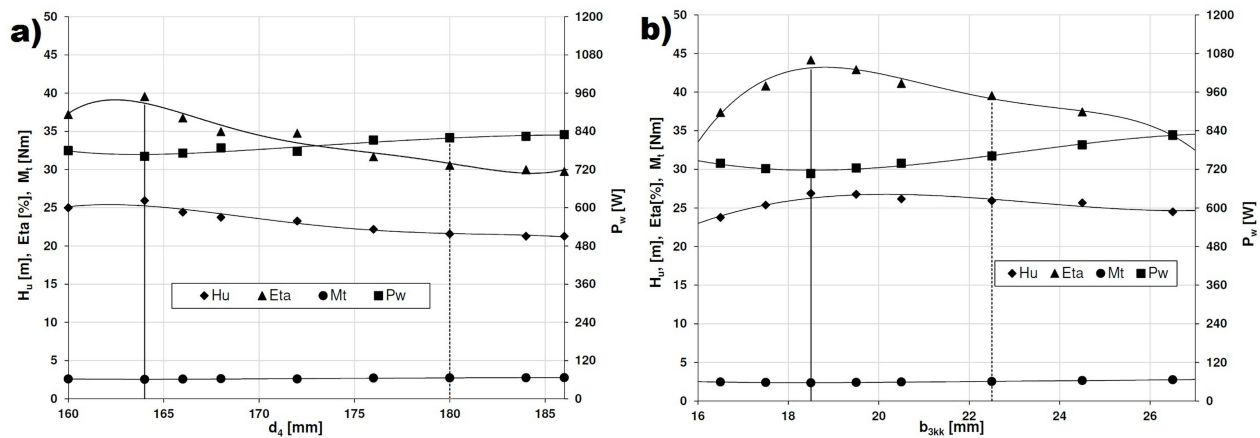


Figure 10: Influence of alteration of the diameter d_4 (a) and width b_{3kk} (b) of the annular casing

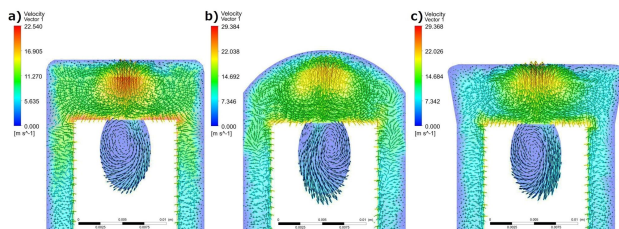


Figure 11: Distribution of liquid velocity on the model lateral plan: (a) rectangular profile, (b) semi-circular profile and (c) trapezoidal profile

Figure 12 presents a comparison of characteristics of a pump with a hole impeller cooperating with the base annular type casing VB and VM. Figure 13 – Figure 15, on the other hand, show the results in a graphic form obtained for flow rate $5.1 \text{ m}^3/\text{h}$ (BEP). The pictures depict the distribution of total (Figure 13) and static (Figure 14) pressure as well as the distribution of velocity (Figure 15). Reducing the outer diameter and the width (variant VM) of the channel resulted in the growth of velocity, nonetheless, velocity distribution in the channel is definitely more uniform in comparison to VB – Figure 15. It affects the decline hydraulic losses – lack of recirculation or turbulence areas. Furthermore, decreasing the outer diameter of the channel and its width (VM) affected at the uniformity of total and static pressure distribution. In Figure 14 it can be observed that the increase of the static pressure is more uniform than in case of VB. Unfortunately, consequently, far greater decline of the static pressure appeared on the inlet edges to the flow channels of the impeller. Working conditions of an outlet diffuser were improved. It is used in a much higher degree. However, still water does not flow into it with the whole perimeter of the cross-section.

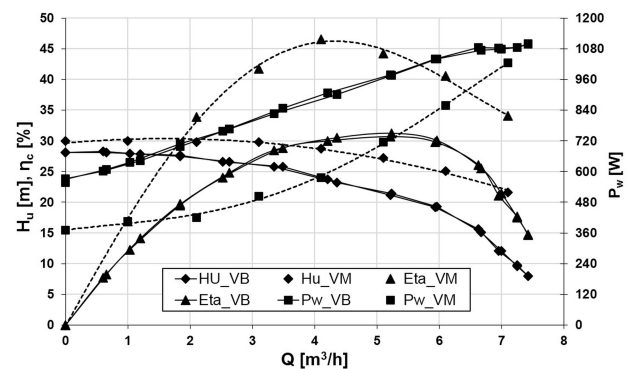


Figure 12: Characteristics of the pump with a hole impeller and two types of annular casing – VB and VM

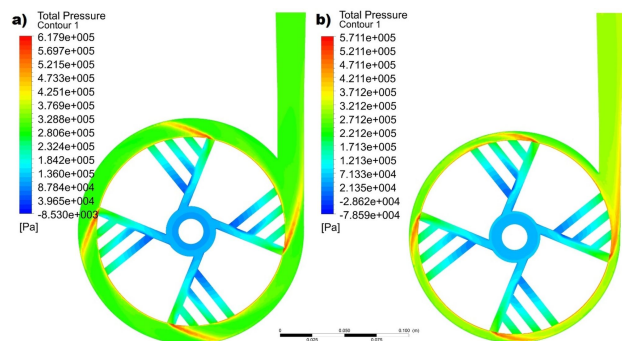


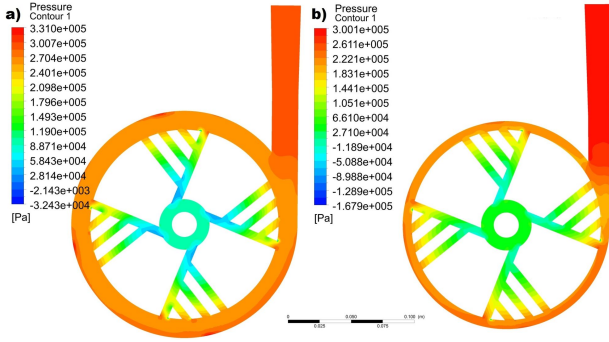
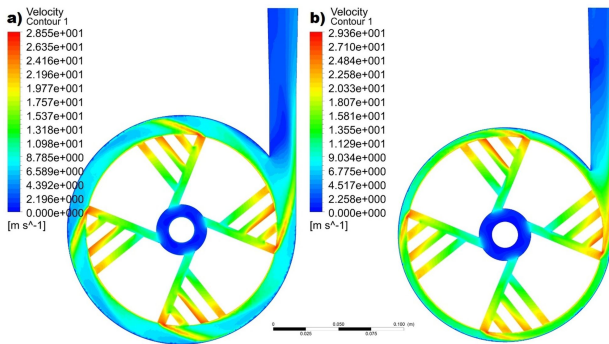
Figure 13: Distribution of total pressure on the model central plane (a) base annular casing VB and (b) final annular casing VM

4.7 Cooperation of a hole impeller with a volute type casing

In the next stage cooperation of a hole impeller with a spiral volute type casing at various configurations of its geo-

Table 3: Influence of changes in the shape of the cross-sectional profile in the annular casing.

Lp.	Profile	Q	M _t	H	P _h	P _w	η _h	η _c
	-	m ³ /h	Nm	m	W	W	%	%
1	rectangular profile	5.1	2.377	27.21	377.2	714.1	52.85	44.22
2	semi-circular profile	5.1	2.687	24.86	344.9	807.2	42.73	35.75
3	trapezoidal profile	5.1	2.401	26.54	368.1	720.9	51.06	42.72

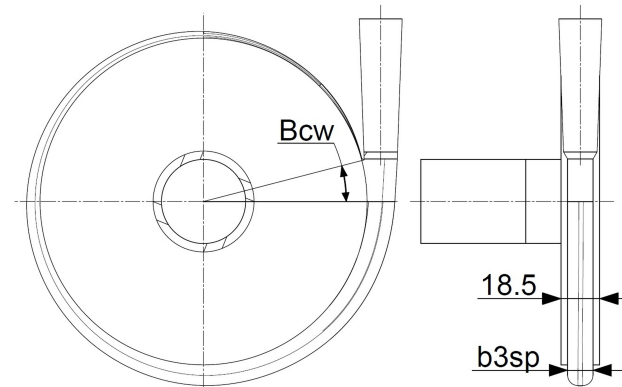
**Figure 14:** Distribution of static pressure on the model central plane (a) base annular casing VB and (b) final annular casing VM**Figure 15:** Distribution of liquid velocity on the model central plane (a) base annular casing VB and (b) final annular casing VM

metric characteristic features was analyzed. The following parameters were tested (Figure 16):

- The angle B_{cw} of the beginning of the volute tongue - between 0° and 26° ; basic value 0° .
- The width b_{3sp} of the channel's cross-section - between 10 mm and 20 mm; basic value 18.5 mm. For all CFD analyses the width of non-spiral part of volute casing was constant (value 18.5 in Figure 16).
- A change of a cross-section - rectangular, semi-circular, trapezoidal.

The results of numerical calculations for the change of d_4 and b_{3kk} were shown in Figure 17. In Table 4, however, one can find the results for the change of the channel's profile. Tests were performed for constant flow rate, cor-

responding BEP ($Q = 5.1 \text{ m}^3/\text{h}$). In Figure 17 a variant of maximum efficiency VM1 was marked with a vertical solid line, a dashed line, on the other hand, represents the base variant VB1. Geometry of primary volute casing was calculated and designed in accordance with theoretical models based on one-dimensional flow theory of vortex pump [1]. The constant and averaged velocity of liquid method was used (in the whole width of collector after impeller outlet).

**Figure 16:** Basic dimensions of a volute type casing

While analyzing the obtained results it can be noticed that:

- The location of the beginning of the spiral volute, so called volute tongue is crucial in case of pumps working at low specific speed $n_q < 10$. The best solution appeared to be the angle $\beta_{cw} = 15^\circ$. Total efficiency was higher of 5 percent point in relation to other obtained values. Additionally, the head was over 20% higher.
- Reducing the width of the volute casing b_{3sp} in a slight degree impacted on the pump's parameters. The increase of efficiency was insignificant and amounted to only 0.19 percent point. The head also grew not more than 0.2%. The decrease in power demand is practically negligible. An essential advantage of the change of the channel's width is a question of enhancing the radius making the outline of the volute casing, thereby, the channel becomes more regular.

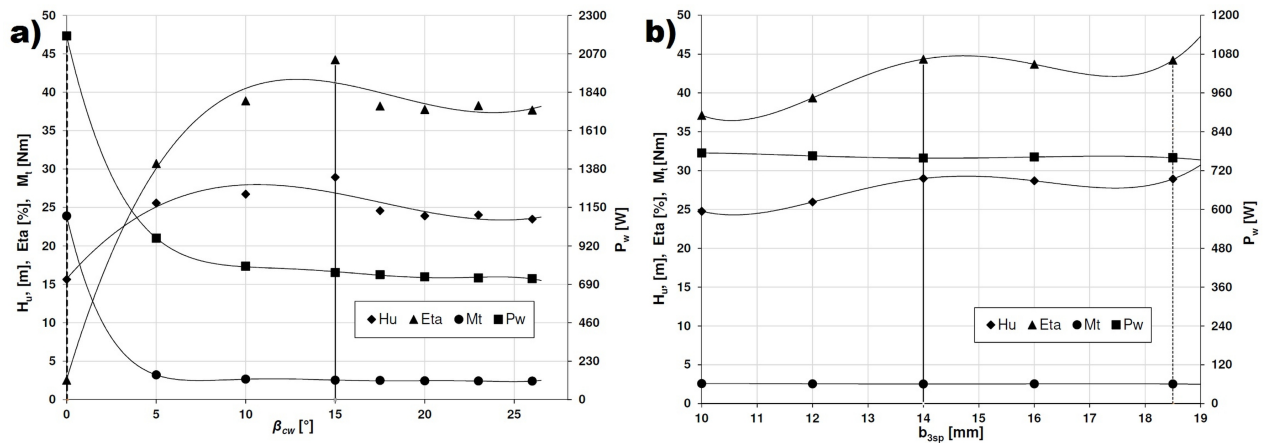


Figure 17: Impact of changes of the angle B_{cw} location of a volute tongue (a) and the width b_{3sp} (b) of the spiral casing

Table 4: Influence of changes of the shape of the cross-sectional profile in a volute casing.

Lp.	Profile	Q	M _t	H	P _h	P _w	η _h	η _c
		m ³ /h	Nm	m	W	W	%	%
1	rectangular profile	5.1	2.525	28.99	402.1	759.6	52.93	44.28
2	semi-circular profile	5.1	2.663	26.08	369.5	802.1	46.06	38.53
3	trapezoidal profile	5.1	2.543	28.28	392.3	768.2	51.06	42.72

- The alteration of the flow area relevantly influences the improvement of flow parameters. In Figure 11 it is visible that the reduction of the flow area caused velocity increase, nevertheless, velocity distribution in the passage is definitely more uniform.
- As in the case of the annular type casing the results of a change of the cross-section shape did not improve pump's characteristic. The application of a semi-circular profile in the volute casing also generates strong recirculation and turbulence in the channel's cross-section, narrowing the outlet from the impeller (Figure 11). A slight increase of the flow area takes place in this case as well, which results in the decrease of flow parameters. The rectangular profile, again, represents the best geometric features.

In Figure 18 characteristic of a pump with a hole impeller cooperating with a volute type casing VM1 can be found.

Figure 19 – Figure 21 are graphic representations of the calculations results for the pump with an annular casing VB and a spiral type casing VM1, obtained for flow rate 5.1 m³/h (BEP).

In the pictures the distribution of total (Figure 19) and static pressure (Figure 20), and velocity distribution (Figure 21) were presented. As in the case of an annular type casing VM, a reduction of cross-sections in a volute type

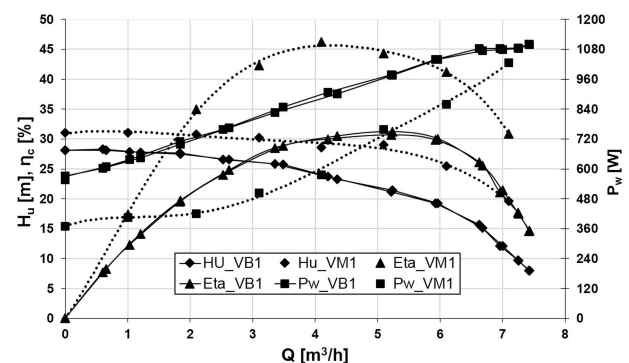


Figure 18: Characteristics of the pump with a hole impeller with an annular casing VB1 and a final volute casing VM1

casing VM1 caused the increase of the liquid velocity while uniforming its distribution. Consequently, the zones of liquid recirculation and turbulence became eliminated – except for the area in the neighborhood of the volute tongue. In a volute type casing VM1 a more uniform distribution of static pressure was obtained (Figure 20) than in case of a final annular type casing VM. This being due to the increase of the radius making the outline of the volute casing, thus, the channel becomes more regular and transforms flow energy much better.

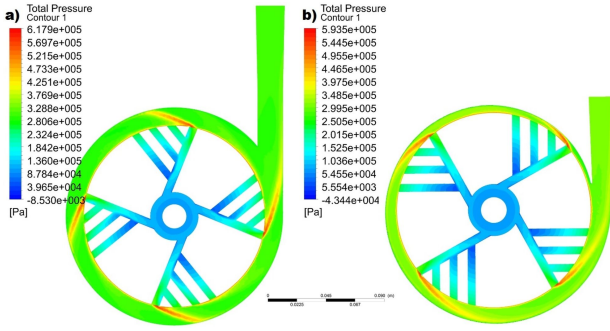


Figure 19: Distribution of total pressure on the model central plane (a) base annular casing VB and (b) final volute casing VM1

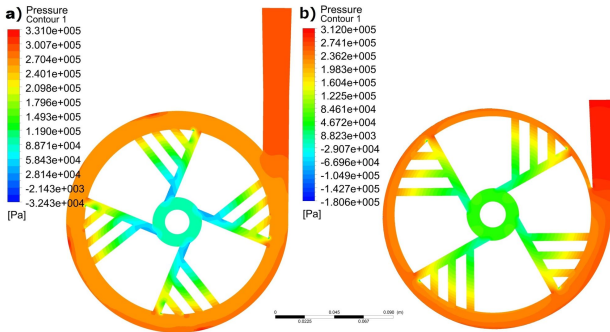


Figure 20: Distribution of static pressure on the model central plane (a) base annular casing VB and (b) final volute casing VM1

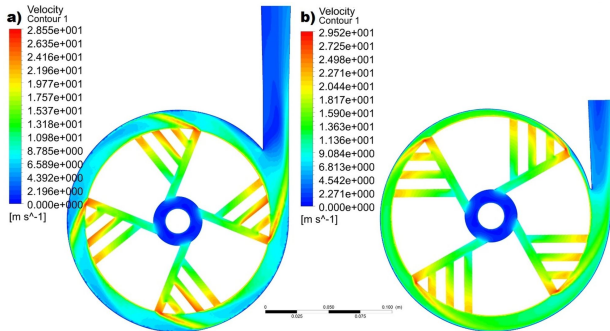


Figure 21: Distribution of liquid velocity on the model central plane (a) base annular casing VB and (b) final volute casing VM1

5 Summary

The results of stator type modification in cooperation with hole impeller which is dedicated to work in ultra low specific speed has been presented herein. Construction details of base and rationalized pump elements have been demonstrated as well as CFD and experimental test results. Hole impeller have become an important alternative to blade rotor in a range $n_q < 10$.

Summarizing the various stages of the above tests one can formulate the conclusions:

- In the considered class of flow problem, the best turbulence model from the accuracy and calculation duration point of view is the model $k-\omega$ SST.
- Calculation error of total efficiency and total head pump – obtained according to [17, 18] – at pump capacity corresponding best efficiency point (BEP), did not exceed 2.6%.
- It has been proven that a proper choice of the type and construction of the stator cooperating with a hole impeller can reduce turbulence and recirculation in the channel's cross-section. This affects the improvement of the velocity distribution and total pressure. By proper selection of construction parameters like: outer diameter of a stator, width of an annular casing, width of volute casing and angle of volute tongue it is possible to improve efficiency (of nearly 14 percent point) and total head pump (over 20%) and a decrease in power demand by the pump (of almost 15%).
- While discussing the cooperation results of a hole impeller with annular casing and spiral casing, it can be detected that the results obtained are almost comparable. This confirms the conclusion included in the literature [1, 2]. It is reasonable to indicate that the annular casing is the best choice for cooperation with a hole impeller in the range of extremely low specific speed $n_q < 10$. As a mainly advantages of annular stator are: simplicity of construction and low production costs.
- Knowledge concerning construction of hydraulic elements of vortex pumps working is insufficient, especially if we move in range of extremely low specific speed. As shown in the paper, the annular type casing VB cooperating with a hole impeller, designed in accordance with [1], reached far poorer operating parameters than the VM construction in a configuration with the same impeller.

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NOMENCLATURE

b_3 – width of a volute casing inlet [mm]

b_{3kk} – width of an annular casing [mm]

b_{3sp} – width of volute casing [mm]

d_4 – outer diameter of a stator [mm]

g – acceleration due to gravity [m/s^2]

H_u – total pump head [m]

M_t – total moment on the impeller rotating surfaces [Nm]

n – rotational speed [rpm]

n_q – kinematic specific speed ($n_q = n\sqrt{Q}/H_u^{3/4}$) [rpm]

p – ambient pressure [Pa]
 p_{cin} – total pressure in an impeller inlet section [Pa]
 p_{cout} – total pressure in a pump outlet section [Pa]
 P_h – hydraulic power [W]
 P_w – power on a pump shaft [W]
 Q – flow rate [m^3/h]
 Re – Reynolds number,
 t_s – time step [s]
 u_x – velocity liquid in a perpendicular direction to the wall [m/s]
 v_{st} – average fluid velocity in stator [m/s]
 β_{cw} – start angle of a spiral volute tongue [deg]
 ϵ – convergence criterion
 η_c – total efficiency
 η_h – hydraulic efficiency
 η_m – mechanical efficiency
 η_v – volumetric efficiency
 ρ – density [kg/m^3]
 ω – angular velocity [rad/s]

References

- [1] Gulich J., *Centrifugal Pumps*, Springer, Berlin, 2008
- [2] Chomiuk B., Skrzypacz J., *Comparison of energy parameters of a centrifugal pump with a multi-piped impeller in cooperation either with an annular channel and a spiral channel*, Open Engineering, 2018, p. 513-522, 8(1). DOI: <https://doi.org/10.1515/eng-2018-0063>
- [3] Kagawa S., Choi Y., Kurokawa J., Matsui J., *Performance of very low specific speed Centrifugal pumps with circular casing*, Journal of Fluid Science and Technology, 2007, p. 130–138, 2(1).
- [4] Matsui J., Kurokawa J., Choi Y., Nishino K., *Flow in the low specific speed Centrifugal pump with circular casing*, in XXIII rd. IAHR Symposium, Yokohama, 2006, p. 1–10.
- [5] Choi Y., Kurokawa J., Matsui J., Imamura H., *Internal flow Characteristics of a Centrifugal pump with very low specific speed*, in: XXI st IAHR Symposium on Hydraulic Machinery and Systems, Lausanne, 2002, p. 1–7.
- [6] Kelder J.D.H., Dijkers R.J.H., van Esch B.P.M., Kruyt N.P., *Experimental and theoretical study of the flow in the volute of a low specific-speed pump*, Fluid Dynamics Research, 2001, p. 267–280, 28(4).
- [7] Kurokawa J., Matsumoto K., Matsui J., Imamura H., *Development of high efficiency volute pump of very low specific speed*, in: Sixth Asian International Conference on Fluid Machinery, Johor Bahru, 2000, p. 1–6.
- [8] Macneille M. B., *Method of making centrifugal pumps*, US1986836 A, 1933.
- [9] Koji K., *Low Specific Impeller*, JP7208392A, 1995.
- [10] Koji K., *Low Specific Impeller*, JP9209983A, 1997.
- [11] The European Standard EN ISO 9906:2000, Rotodynamic pumps. Hydraulic performance, acceptance tests., Grades 1 and 2, BSI, 2003.
- [12] Skrzypacz J., *Investigating the impact of drilled impellers design of rotodynamic pumps on the efficiency of the energy transfer process*, Chemical Engineering and Processing, 2015, p. 60–67, 87.
- [13] Anderson Jr., John D., *Computational Fluid Dynamics: The Basics With Applications*, McGraw-Hill Science, New York, 2012.
- [14] Schwingea J., Wiley D.E., Fletcher D.F., *A CFD study of unsteady flow in narrow spacer-filled channels for spiral-wound membrane modules*, Desalination, 2002, p. 195–201, 146(1–3).
- [15] Jasak H., Gosman A.D., *Automatic resolution control for the Finite-Volume method, Part 3: Turbulent flow applications*, Numerical Heat Transfer Part B, 2000, p. 273–290.
- [16] Jasak H., *Turbulence Modelling for CFD*, (website status available on the date: 28.02.2019): <http://www.powerlab.fsb.hr/ped/kturbo/OpenFOAM/SummerSchool2009/lectures/Turbulence.pdf>
- [17] Richardson L.F. (2000), “The Approximate Arithmetical Solution by Finite Differences of Physical Problems Involving Differential Equations, with an Application to the Stresses In a Masonry Dam”, *Transactions of the Royal Society of London*, Ser. A, 210, pp. 307-357.
- [18] Celik I., Karatekin O., *Numerical Experiments on Application of Richardson Extrapolation With No uniform Grids*, ASME Journal of Fluid Engineering, 1997, p.584-590, 119.