The influence of the impeller construction on the performance of one channel pump

P Szulc¹, G Moliński² and O Moloshnyi³

¹ Department of Mechanics, Machines, Devices and Energy Processes, Wroclaw University of Science and Technology, Wroclaw, 50–370, Poland
² Pompax Sp. z o.o., Rydzyna, 64–130, Poland
³ Applied Hydro- and Aeromechanics Department, Sumy State University, Sumy 40007, Ukraine

E-mail: o.moloshnij@pgm.sumdu.edu.ua

Abstract. The paper presents the comparison of the impeller’s construction on the operating parameters of the sewage pump. Three types of impellers were taken into consideration. The main geometrical parameters were identical, the differences were applied in the meridional cross–section and in the shape of the passages. These three objects of research represent three approaches of the construction of single channel pumps. The analysed impellers were cooperated with the same spiral casing where the flow was identified by means of numerical simulation. The obtained results allow to choose appropriate parameters of the pump in terms of the operating point and shape of the characteristic curves.

1. Introduction

Sewage pumps pump fluids with a large amount of solid particles. In this group, one channel pumps are often one of the most popular types dedicated for contaminated fluid transportation. The advantage of these pumps could be defined as a large width of the passage which is the result of one blade application. It leads to high clog resistance. It is commonly known that meridional cross–section and blade trajectory could alter, thus many types of the construction could be defined. This results in vast variety of such constructions. Moreover, there is no currently accurate method dedicated to designing a flow element in considered pumps. The most effective way to calculate a new machine is based on calculations of the existing unit using hydrodynamic similarity laws with some additional adjustments. That is why the analysis of the one channel pumps is interesting from a scientific point of view.

In literature, the design and examination problem of one channel pumps has not been sufficiently discussed. Kim et al. [1] designed one channel pump buy means of Stepanoff theory but the result was not proper and after modernization of the impeller geometry the authors reached stable velocity distribution in the impeller cross–section and more uniform pressure and velocity distribution. Nishi et al. [2] obtained satisfying results of new analytical approach compared to the numerical method based on the determination of the geometrical parameters of one channel pump impeller. Kim et al. [3] optimized impeller and volute of a single blade pump to improve the hydraulic efficiency and reduce the unsteady radial force by changing the cross–section area. Tan et al. [4] studied the three impellers with different wrap angle. Increase of the wrap angle improves the working parameters and decreases radial forces. Tan et al. [5] compared also three impellers with different blade outlet angle and noted
that increase of the blade outlet angle causes the rise of the head and radial forces. Tan et al. [6] stated that the form of the outlet element is responsible for pressure fluctuations. Souza et al. [7] checked the use of Stepanoff method to calculate three volutes at different reference speeds and drew the conclusion that this method meets all the requirements.

Gulich [8] based on the radial force measurements in the one channel pump had presented the recommendation for their parameter prediction. Nan et al. [9] studied the radial force and noted that they are higher in the case with volute than with vortex casing. The increase of flow rate results in the rise of the radial force amplitude and reduction of the rotational speed has very small effect on its value. Pei et al. [10] analysed the fluctuation of the hydrodynamic forces and pressure distributions in correspondence to the impeller position.

A review of the literature indicates that the numerical simulation is an accurate method for predicting the flow in this type of pumps. But there is no information about the influence of the impeller shape on the working parameters and operating conditions of these units. The main purpose of this work is to analyse flow structures in the three types of single blade pumps equipped with three different impellers. In such analysed cases the volutes were identical.

2. Object of research

The object of the research is a sewage pump where three different types of impellers were investigated. These three types of impellers are presented in Figure 1. The main operating and geometrical parameters of the analysed one channel pumps are listed in Table 1.

Table 1. Operating and geometrical parameters of the analysed one channel pumps.

| Name               | Symbol | Impeller No. 1 | Impeller No. 2 | Impeller No. 3 |
|--------------------|--------|----------------|----------------|----------------|
| **Performance parameters of the pump** |        |                |                |                |
| Design flow rate   | $Q, \text{m}^3/\text{h}$ | 123            | 129            | 122            |
| Design head        | $H, \text{m}$ | 10.8           | 11.1           | 11.5           |
| Rotating speed     | $n, \text{1/min}$ | 1450           | 1450           | 1450           |
| Specific speed     | $n_q$  | 45             | 45             | 43             |
| **Geometrical parameters of the impeller** |        |                |                |                |
| Outlet diameter    | $d_2, \text{mm}$ | 243            | 240            | 246            |
| Inlet diameter     | $d_1, \text{mm}$ | 105            | 100            | 100            |
| Inlet width        | $b_1, \text{mm}$ | 106            | 82             | 70             |
| Outlet width       | $b_2, \text{mm}$ | 85             | 82             | 82             |
| Blade wrap angle   | $a, ^\circ$ | 330            | 320            | 425            |
| Outlet blade angle | $\beta_2, ^\circ$ | 14             | 18             | 26             |

As we can see in Figure 1 the main differences in the impellers shape was realized by the utilisation of alteration of meridional cross-section and blade trajectory but also in wrap angle. Impeller No. 2 had a precisely defined passage (circular shape cross-section), instead of a clearly determined blade (models No. 1 and No. 3). The flow channel was made by ball extrusion along a defined path where the angle of convergence was assumed.
3. Methods of research
A solid model of a fluid computational domain of the sewage centrifugal pump includes: inlet pipe, impeller, volute and outlet pipe (Fig. 2). Due to pump calculation performance, the geometry of the flowing part did not include the chambers with the suction wear ring. This reduces complexity of the model and increases the speed of calculation.

Tetra mesh for impellers, volute and hexa mesh for inlet and outlet pipes were generated using ICEM–CFD software (Fig. 3). The size of the elements was selected according to mesh independence test and layers of prismatic elements were created near solid walls. The mesh contained approximately 2.8 million of nodes. In details, the volute contains 1.2 million nodes and the impellers contain: 0.7 million, 0.8 million and 0.9 to 0.45 million respectively to the model type (No. 1-3).

The numerical simulation of the pumping process in one channel-pump passages was conducted by means of ANSYS CFX software. The Reynolds–Averaged Navier–Stokes method with the standard k–ε turbulence model was used [11]. Boundary conditions were set as the mass flow rate at the pump inlet and the opening pressure and direction at the pump outlet. Numerical simulations were carried out at the following flow rate range (0.3–1.7) $Q_{nom}$ for steady and (0.7; 1.0; 1.3) $Q_{nom}$ transient states. Working fluid was assumed as water at 25˚C.

4. Results and discussion
The value of the head and power was determined as base variables to compare the analysed types of the impellers. The obtained results are presented in Figure 3. The head curves for all models are stable. In general, obtained $H$–$Q$ curves have similar steepness. It could be concluded that increase of flow rate leads to decrease in the head, but the same exceptions can be observed. It is the result of irregular flow structure inside the impellers. The differences in the head between each impeller are not regular in function of discharge. In the design point, the difference between the achieved outcomes equals approximately 3–5 % and the maximum value – 10% – was noticed at lower flow rates.
Impeller No. 2 is characterized by specific shape of the meridional cross-section and blade (it is very wide), which has influence on the slope of the $H-Q$ curves. At the flow rate below $0.7Q_{nom}$ it is becoming flatter. The reason could be found in hydraulic loss reduction in the volute and impeller.

![Figure 2. Model dedicated for numerical simulations.](image)

Efficiency curves of impellers No.1 and No. 2 are similar, but the curve obtained for impeller No. 2 has a separate form. It is located higher for flow rates in the range $(0.3-1.0)Q_{nom}$ reference to other impellers and maximum values of efficiency were noted for flow rates in the range $(0.7-1.5)Q_{nom}$. The reduction of hydraulic losses is the main reason of the achieved results. The difference at the $Q_{nom}$ is

![Figure 3. The comparison of numerical simulation results for different impellers.](image)
approximately 6%. It could be concluded that the impellers where the passage has circular shape cross–section have flat efficiency curve.

The power consumption curves of the analysed pumps have similar steepness. A slight difference could be observed on the curves form caused by alteration in the head and losses characteristics. Those differences close to the best point are less than 4%.

Due to the transient character of the flow and lack of symmetry in hydraulic components unsteady simulations were conducted. The graph – Figure 4– shows the pulsation of head and power at $Q_{nom}$. The peak–to–peak height values are: 39 m (32%), 2 m (18%) and 4 m (30%), for pump with impellers: No. 1, No. 2 and No. 3, and the average values are: 12.2 m, 11.5 m and 13 m respectively. The highest values were obtained to the position in which the trailing edge of the blade is located 120° to the tongue of the casing (Fig. 5 (a)). The smallest value was obtained to the position when the trailing of the blade edge is passing the tongue of the volute (Fig. 5 (c)). A similar shape of received curve for pump with impellers No. 1 and No. 3 can be observed. It could be explained by similar structure of the meridian cross–section and also form of the blade. Impeller No. 3 has higher blade wrap and outlet blade angle. The pump with impeller No. 2 revealed fluctuation of the head twice smaller than with impellers No. 1 and No.2.

When power consumption trend is taken into consideration, the peak–to–peak height values are about: 1.1 kW (23%), 0.6 kW (15%) and 1.3 kW (27%) for pump with impellers: No. 1, No. 2 and No. 3. The average values are: 4.6 kW, 4.2 kW and 4.8 kW respectively. Power curves have different shape in comparison with head curves. Rapid increase of power consumption is observed when the trailing edge is passing the tongue. This does not correspond to the high peak of the head. After that, slow decrease of power for impellers: No.1 and No. 3 and sudden drop for No.2 is observed. In general power consumption curve analysed in function of rotation angle is more uniform for impeller No. 2. The reason can be explained by smaller hydraulic losses in the impeller and volute casing.

![Figure 4. Fluctuation of the head and power refers to blade position.](attachment:image)

In the next step, the quality analysis of the flow structure was made. The absolute pressure distributions at the mid–span of the impeller (in different position) and volute in the $Q_{nom}$ were presented in Figure 5. The pressure distribution depends on the position of the impeller. The comparison of the absolute pressure fields indicates that they are similar for both cases: No. 1 and No.
2. There are no sudden changes in the pressure gradient. The small number of blades (one blade) cause not uniform pressure distribution. The differences in pressure for pump with impeller No. 2 and the other two units are quite large. The highest divergences were achieved in suction side of the blades and the smallest in the volute casing. The outlet diffusers were characterized by the identical pressure distribution.

![Figure 5. Pressure distribution in considered pumps: (a) impeller No. 1 (max. head), (b) impeller No. 2 (max. head), (c) impeller No. 3 (max. head); (d) impeller No. 1 (min. head), (f) impeller No. 2 (min. head), (e) impeller No. 3 (min. head).](image)

The comparison of the velocity distributions in the analysed impellers and volute cross-sections was presented in Figure 6. The conducted analysis indicates that there is not a great difference in volute casing operation for all impellers. In the outlet diameter of the impellers velocity distribution is similar. Nevertheless, near the leading edge the same difference was noticed. Impeller No. 2 has a small zone of separation (low velocity near pressure side of the blade). It can be explained by application of a very thick blade to eliminate this area.

Uneven pressure distribution around the blade and in the volute casing encourages to make the analysis of the radial forces in the pump. Calculated radial hydrodynamic loads for all impellers, by means of equation 1:

\[ F = (f_x^2 + f_y^2)^{1/2} \]  

in the function of the position of the impeller are presented in Figure 7. In addition, the center of the force is indicated. It is possible to say that radial forces are imbalanced, because the divergences from the centre are approximately 100 N on axis X and 20 N on axis Y for pump with impeller No. 1 and No. 3. When analysing impeller No. 2 it is 70 N and 40 N respectively. It means that the centres of hydrodynamic forces are shifted mainly along X and Y axis and may cause additional vibrations. The amplitudes of the values are about: 1800 N, 2200 N and 1500 N for the Y axis for impellers: No. 1, No. 2 and No. 3 respectively. The smallest hydraulic radial force was obtained in the course of model No. 3 examination. The possible reason of this situation is the biggest wrap angle of the blade. Considering point of operation with maximum head, the hydraulic forces are directed in the same way...
for all impellers. It confirms that the position of the trailing edge determines the maximum head. The points of minimum head operation correspond to the vectors of the hydraulic force characterized by different direction for each impeller.

Figure 6. Velocity distribution in the considered pumps: (a) impeller No. 1 (max. head), (b) impeller No. 2 (max head), (c) impeller No. 3 (max. head); (d) impeller No. 1 (min. head), (f) impeller No. 2 (min. head), (e) impeller No. 3 (min. head).

Figure 7. Calculated radial hydraulic force (N).
5. Conclusions
The paper presents the results of the steady and transient flow simulations in one channel pumps with three different impellers and one volute casing. The analysed impellers were designed by means of two different methods. Impellers No. 1 and No. 3 were equipped with a blade, similar to a classical centrifugal pump. The main differences were assumed in meridional cross-section and in the trajectory of the blade and also in wrap angle. The specific speed equals 45. Impeller No. 2 had clear passage (circular shape cross-section), instead of a blade. The flow channel was made by ball extrusion along a defined path. In accordance with the conducted analysis the following conclusions could be drawn:

- The head curves for all variants of the analysed impellers are stable, some fluctuation of the achieved values was observed. In general, obtained \( H-Q \) curves have similar steepness. The differences between the achieved outcomes obtained for each impeller are not regular in function of discharge. In the design point it is approximately 3–5 %.
- Efficiency curves of impellers No.1 and No. 2 are similar, but the curve obtained for impeller No. 2 has a separate, more advantageous form. We can see that for lower discharge the efficiency is higher. The operation range where the maximum of efficiency is secured covers almost 100 m\(^3\)/h. It is more than a half of discharge.
- The differences between the power consumption curves close to the best point equal less than 4% and this value is rising away from the \( Q_{nom} \).
- Unsteady simulation shows the pulsation of head and power at \( Q_{nom} \). The peak–to–peak height values are: 39 m (32%), 2 m (18%) and 4 m (30%), for the pump with impellers: No. 1, No. 2 and No. 3. The highest values were obtained to the position in which the trailing edge of the blade is located 120° to the tongue of the volute and the smallest for the position of the impeller when the trailing edge of the blade is passing the tongue of the spiral casing. The pump with impeller No. 2 has fluctuation of the head twice smaller than with impellers No. 1 and No.2.
- The peak–to–peak height values of the power are about: 1.1 kW (23%), 0.6 kW (15%) and 1.3 kW (27%) for the pump with impellers: No. 1, No. 2 and No. 3. Power curves have different shape in comparison with head curves. In general, power consumption curve analysed in function of rotation angle is more uniform for impeller No. 2.
- The absolute pressure distributions at the mid–span of the impellers and volute cross-sections indicates that inner pressure distributions are similar for cases with impellers: No. 1 and No. 2. The application of one blade causes irregular pressure distribution. The differences in the achieved outcomes between pump with impeller No. 2 and the other two are quite large.
- The comparison of the velocity distributions in the analysed impellers and volute cross–section indicates that there is not a significant difference for variants: No. 1 and No. 3. Impeller No. 2 has a very small zone of low velocity located near the leading edge at the pressure side of the blade.
- Calculated radial hydrodynamic force in a function of the position of the impeller confirms its unbalance, the centres of hydrodynamic forces are shifted mainly along X axis and could cause additional vibrations. The amplitudes of the values are about: 1800 N, 2200 N and 1500 N for Y axis for impellers: No. 1, No. 2 and No. 3 respectively.

References
[1] Kim J-H, Cho B-M, Kim Y-S, Choi Y-S, Kim K-Y, J-H and Cho Y Optimization of a Single-Channel Pump Impeller for Wastewater Treatment International Journal of Fluid Machinery and Systems 9(4) 370
[2] Nish Y, Fujiwara R and Fukutomi J 2009 Design Method for Single-Blade Centrifugal Pump Impeller Journal of Fluid Science and Technology 4(3) 786
[3] Kim J-H, Ma S-B, Choi Y-S and Kim K-Y 2019 Simultaneous Optimization of Impeller and Volute of a Single channel Pump for Wastewater Treatment International Journal of Fluid Machinery and Systems 12(2) 99
[4] Tan M, Ji Y, Liu H, Wu X and Zhu Z 2018 Effect of Blade Wrap Angle on Performance of a Single-Channel Pump *The Society for Experimental Mechanics* 1

[5] Tan M, Zhu Z, Liu H, Wu X and Feng J 2017 Effect of the blade outlet angle on unsteady characteristics of a single channel pump *JVE International Ltd. Journal of Vibroengineering* **19**(5) 3768

[6] Tan L, Shi W, Zhang D, Zhou L and Wang C Numerical and experimental investigations of pressure fluctuations in single-channel pumps *Proc IMechE Part A: J Power and Energy* 1

[7] De Souza B, Niven A and McEvoy R 2010 A Numerical Investigation of the Constant-Velocity Volute Design Approach as Applied to the Single Blade Impeller Pump *Journal of Fluids Engineering* **132** 1

[8] Gülich J F 2014 *Centrifugal Pumps* 3rd Edition (Springer Berlin, Heidelberg, New York)

[9] Tan L, Shi W, Zhang D, Zhou L, Wang C Zhou L and Mahmoud E 2018 Numerical and experimental investigations on the hydrodynamic radial force of single-channel pumps *Journal of Mechanical Science and Technology* **32** *(10)* 4571

[10] Pei J, Yuan S and Yuan J 2013 Fluid-structure coupling effects on periodically transient flow of a single-blade sewage centrifugal pump *Journal of Mechanical Science and Technology* **27**(7) 2015

[11] ANSYS CFX Reference Guide, Release 15.0 2013 http://www.ansys.com

**Acknowledgments**

Calculations have been made using resources provided by Wroclaw Centre for Networking and Supercomputing (http://wcss.pl), grant No. 444/2017.