Effect of Impeller Size on the Performance of a Single Blade Pump

Branislav Knížat[1,*], *Dušan* Šlachtič[1], and *Róbert* Olšiak[1]

[1]STU in Bratislava, Faculty of mechanical engineering, Nám. slobody 17, 81231 Bratislava, Slovakia

Abstract. Changing the impeller diameter is a frequently used method for adjusting pump performance parameters. In the case of conventional multi-blade impellers, this is done by reducing the diameter on the machine tool to the prescribed shape. The other dimensions of the pump (diffuser, inlet) remain unchanged. This method is called trimming. The article deals with the diameter reduction and subsequent modification of the performance parameters of a single blade pump. These pumps are characterized by certain specific features. First of all, it is an unsymmetrical impeller that must be both statically and dynamically balanced. This plays an essential role in the whole modification process. Research results obtained on a pump with an impeller diameter of 138 mm are presented. The change in diameter was monitored on a total of 3 impellers. Experiments were carried out in the laboratory of hydraulic machinery. The results were verified by CFD calculations.

Research background: The article concerns the modification of impellers of single blade pumps. The effects of diameter modifications on multi-blade pumps are currently known. However, these have symmetrical impellers and can therefore be changed without restriction. For asymmetric (single blade) impellers, the problem is more complex as additional mass must be added to provide static and dynamic balance.
Purpose of the article: The aim is to determine the nature of the change in performance parameters when the output diameter is changed. The results of the research can be applied in the prediction of the change in the operating point and the creation of tombstone charts.
Methods: Two kinds of methods were used in the research: experiment and CFD calculation. A total of 4 impeller sizes were investigated.
Findings & Value added: The results of the paper can be divided into two areas. In the experimental area, a device was designed to measure the characteristics of single blade pumps. Four impellers were manufactured and tested. In the area of CFD calculations, simulations of the hydraulic parameters around the best efficiency point (BEP) were performed. The calculation results were verified by experiment. The nature of the change of the BEP when the diameter of the impeller changes up to 87.9 % was found.

Keywords: *single blade pump; trimming; balancing; impeller.*

* Corresponding author: branislav.knizat@stuba.sk

1 Introduction

Parameter adjustment by changing the impeller diameter is a common procedure in the field of turbomachinery, i.e. compressors and pumps and also pumps operating as turbines (PAT). After the modification, a new turbomachine with altered hydraulic parameters is created. By reducing the diameter, the flow rate, head and efficiency are shifted to lower values. In the case of symmetrical impellers (with multiple blades), this is an after-treatment realised after production, carried out on machine tools and is therefore called trimming. In the case of multi-bladed symmetrical impellers, this modification can be carried out without restriction due to the fact that the trimming does not cause mechanical unbalancing. Trimming is an economical method of parameter adjustment and a diameter change of up to a maximum of 15 % is generally recommended. The characteristics of the untrimmed impeller and the maximum trimmed one are defined by the so-called tombstone chart of the particular machine type. The shape of the tombstone chart is determined by the type of turbomachine and the variation of the operating curves depends mainly on the magnitude of the blade angles in the outlet area.

Unsymmetrical impellers of single blade pumps are another case. Here the situation is more complicated, because trimming the impeller results in unbalancing. Performing this operation after manufacture is therefore out of the question. As a rule, manufacturers offer a number of impellers needed to form a tombstone chart of a given type, but these cannot be further modified. They are derived from the original impeller by trimming the outer diameter during design and adding or removing mass as appropriate to ensure overall static and dynamic balance. This paper discusses the effect of reducing the outer diameter of the impeller of a single blade pump on its performance characteristics. The physical nature of the flow as well as the principle of flow and head reduction are the same as for other turbomachines. The difference is that single blade impellers have a large wrap angle and small blade angles, so the sensitivity to diameter change is higher. When making diameter adjustments, it is often necessary to change the blade length to simplify subsequent balancing. However, the basic knowledge is applicable from general turbomachinery theory. Many methods of changing the geometrical parameters by trimming are known. In [10] a so-called pump impeller shroud trimming (PIST) is presented, where the front disk is turned in the range of up to 25 %. The hydraulic parameters as well as the effect of the impeller change on the radial force loading the shaft are monitored. Often the effect of different modifications of the trailing edge is investigated. In fact, the cut does not have to be linear but can be V or U shaped. Such cases are presented in the study [3]. An important observation is the determination and prediction of the change of the position of the best efficiency point (BEP) depending on the change of the output diameter of the impeller. This involves the development of preferably simple relations called trimming laws, derived on the basis of affinity laws. They are presented in papers [1] and [15]. Both experimental methods and CFD simulation methods are used to investigate the effects of trimming. When simulation methods are used, the authors usually verify the results obtained experimentally. In reference [2], the authors investigated the effect of trimming on a hydrodynamic pump with low specific speed. They used CFD methods to investigate the performance parameters (dimensionless head and dimensionless flow) and compared them with the test data. Pressure pulsations and hydraulic noise were monitored depending on the trimming. The results were explained by the shape of the velocity field, which was formed by the interaction of the impeller and the spiral. One of the results is that trimming reduces the pressure pulsations. The authors of the paper [5] investigated the possibility of modifying the impeller of a pump operating as a turbine (PAT). Both experimental research and CFD simulation are presented, and the results of both are verified against each other. The effect of impeller diameter, inlet blade angle, outlet blade angle and wrap angle is analysed. The phenomenon investigated in literature [6] is the axial trimming of the blade. This method is also sometimes used for pump impellers without a front disc. CFD methods have shown that it is possible to reduce the flow rate while maintaining the output pressure and the overall pump efficiency. In [9], the authors analyse the trimming method in terms of its effect on energy losses and the magnitude of radial force. Three impeller trimming schemes are investigated. The pump on which the research was carried out was a double suction pump. The performance parameters calculated by CFD methods were verified by experiment. In this paper, energy dissipation analysis has been carried out to find out the cause of losses and their evolution during the impeller trimming. Another aim of the research was to observe the radial force fluctuations. A significant decrease in forces was found as the impeller diameter decreased. The authors of the paper [13] present an overview of methods to predict the BEP position when the pump is used as a turbine (PAT). One of the methods to adjust the position of the working point is to trim the impeller.

So far, there is not much knowledge in the literature about trimming of single blade pumps. However, this procedure has to be carried out in a different way from the conventional turbomachines described above. Since single blade pumps are being marketed in specific areas such as pumping of solids-containing substances, it is important to deal with this issue. Manufacturers therefore offer tombstone charts for each type. However, the number of impellers in the chart is usually limited to 3-4, as the trimmed impeller has to be designed separately, as it has to be statically and dynamically balanced by adding or removing mass on the rotor body. Balancing is thus an essential part of the design.

There are many articles that deal with the design of single-blade impellers, the analysis of the flow and the resulting consequences such as the radial load on the impeller. These are, for example, the papers [7], [11], [12] and [14]. The effect of the number of blades on the pump operation is discussed in [4].

2 Single blade pump geometry

The machine under investigation is a hydrodynamic single blade pump with a volute diffuser. The impeller has an inclined trailing edge and a maximum output diameter of $D_2 = 138$ mm.

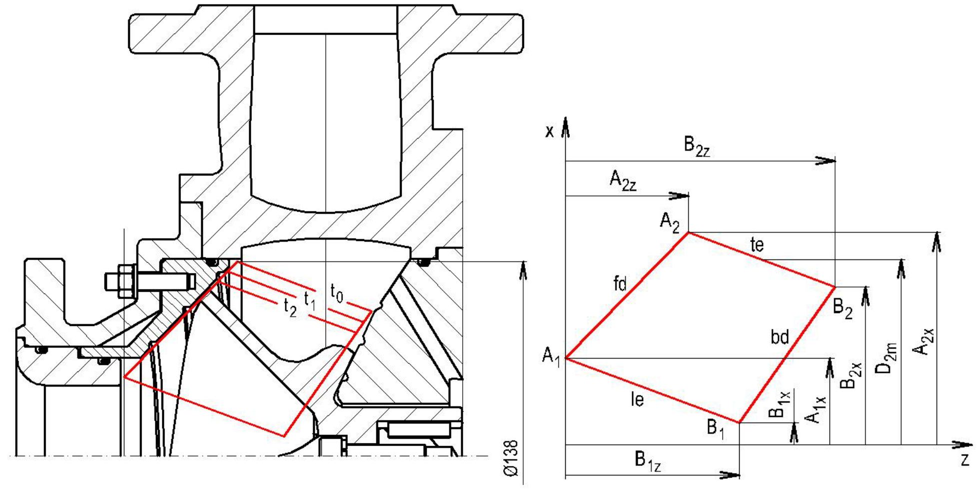

Fig. 1. Dimensions of the tested pump.

Figure 1 shows the basic dimensions of the meridional cut of the pump. The suction and discharge throat of the pump have a diameter of 50 mm. The line markings are as follows: trailing edge (te), leading edge (le), front disc (fd) and back disc (bd). The basic non-trimmed impeller has a blade with a trailing edge marked t_0. From it, the trimmed impellers are then derived with trailing edges marked t_1 and t_2, respectively. Figure 1 shows the shape of the meridional cut as a polygon between the determining points $A_1(A_{1x}, A_{1z})$, $A_2(A_{2x}, A_{2z})$, $B_1(B_{1x}, B_{1z})$ and $B_2(B_{2x}, B_{2z})$ in the coordinate system x, z. The coordinates of these points are given in Table 1.

Table 1. Parameters of meridional cut.

	t_0	t_1	t_2
A_{1x}	28	28	28
A_{1z}	0	0	0
A_{2x}	69	65.5	62
A_{2z}	40	36.6	33.2
B_{1x}	7	7	7
B_{1z}	56.6	56.6	56.6
B_{2x}	51.4	47.6	43.8
B_{2z}	87.6	85	82.3

A view of the impeller from the axial direction is shown in figure 2. It shows the streamlines on the front disc (fb) and back disc (bd) and the trailing edge (te).

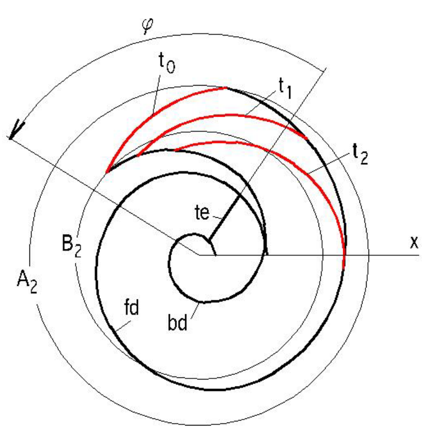

Fig. 2. Axial view on the impeller.

The polar coordinate (angle φ) defining the position of a point on the (fb) and (bd) lines is explained. The shapes of the trailing edges t_0, t_1 and t_2 of each impeller are also indicated in figure 2. Circles A_2 and B_2 correspond to the particular points from the meridional section in figure 1. The same is true for the labels (fd), (bd), (le) and (te). Figure 3 shows the course of the blade angles on the streamlines of the front (fd) and back (bd) discs as a function of the polar coordinate (angle φ from figure 2). The end points of the basic non-trimmed impeller t_0 and the particular trims t_1 and t_2 are marked on the streamlines.

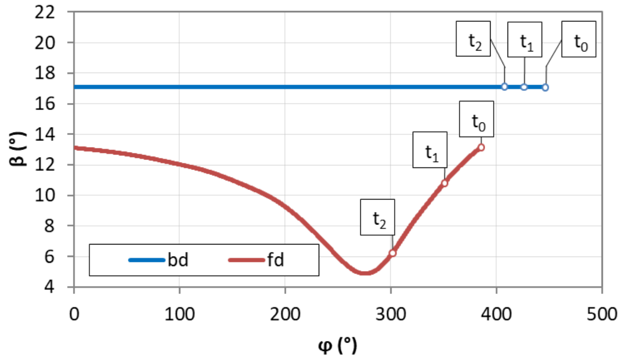

Fig. 3. Blade angle *β* along streamlines (bd) and (fd).

The surface of the blade is stretched between the front and back disc streamlines and geometrically it is a developable surface that can be unfolded in parts. The impeller had to be rebalanced after each trimming. For an overview of rotor balancing methods, see [8].

3 CFD analysis

The core of the theoretical research was CFD flow analysis. The main objective was the determination of the power curves $Y(Q)$ and $P(Q)$. Pump simulations were performed using the Fluid flow CFX module included in the ANSYS 2019R2 software. Before running the actual simulation, it was necessary to prepare a 3D model of the internal part of the pump which was subsequently inserted into ANSYS. For easier resolution, the model was divided into 3 domains: inlet pipe, impeller (OK) and volute. These domains are connected by so-called interface surfaces. The model thus prepared was meshed using the Meshing module and setting the boundary layer thickness to 10^{-5} m. The type of flow analysis was chosen as Steady State together with the turbulent BSL EARSM (Baseline Explicit Algebraic Reynolds Stress Model) applied to the domains and Stage (Mixing-Plane) applied to the interface surfaces. The OK domain was set as rotating with a rotation speed of 2900 min^{-1}. The pressure at both the pump inlet and outlet was set as atmospheric and the simulation was controlled by specifying the flow rate at the pump outlet. By repeating the simulation with the changed flow rate, the Q-Y dependence of the pump was obtained.

4 Experiments

Figure 4 shows a scheme of the measurement system. The designation of the particular parts of the diagram is as follows: MM – driving motor; AP – auxiliary pump; AM – driving motor for AP; MM – motor; MP – measured pump; TV – throttle valve; SV – shut off valve; FT – feeder tank; CT – collector tank.
The measured parameters speed (n), flowrate (Q), suction pressure (p_1), discharge pressure (p_2), pressure differential ($\Delta p=p_2-p_1$) and torque (M_k) are collected by the DAQ system and evaluated in a PC computer unit. From the measured data, the specific energy Y, the power P and the overall efficiency η are evaluated. The system is characterised by the fact that the

liquid (water) is pumped by a metering pump (MP) from the upper tank (FT) to the lower tank (CT). Subsequently, the liquid is pumped back to the upper tank (FT) by an auxiliary pump (AP).

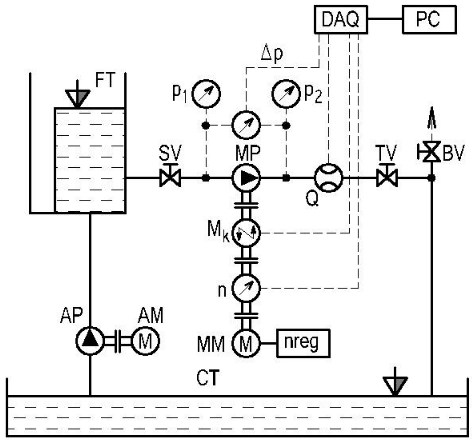

Fig. 4. Scheme of the measurement system.

The suction tank is therefore at a higher geodetic elevation than the discharge tank. This configuration was chosen because the pump has a low height and to compensate for losses in the piping when measuring higher flow rates. The impellers were manufactured from ABS plastic using Rapid Prototyping methods.

5 Results of investigation

All 3 impellers (the original t_0 and the trims t_1 and t_2) shown in figure 2 were tested on the experimental setup and simulated by CFD methods. Thus, by measurements and calculations, the characteristics of the pumps as a function of the flow rate Q were determined: $Y(Q)$, $P(Q)$ and $\eta(Q)$, where Y – specific energy, P – power input and η – efficiency. The simulations were performed as steady state, therefore their results are only relevant in the vicinity of the BEP. The best agreement between the measurement and simulation results was obtained for the non-trimmed impeller t_0. As the impeller was trimmed (i.e., as its outlet diameter decreased), the differences increased.

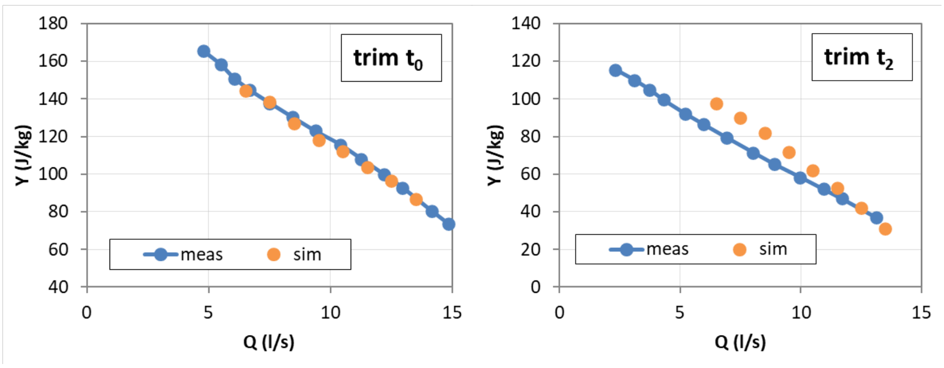

Fig. 5. Measured and simulated specific energy Y. Speed n = 2900 min^{-1}.

In order to compare the degree of agreement between experiment and CFD, we present the results of only the non-trimmed impeller t_0 and the maximum trim t_2, for which the differences were the largest.
All presented curves were measured or calculated at pump speed n = 2900 min^{-1}. Figure 5 shows the specific energy Y curves. For the variant t_0 (original impeller) we see a very good agreement of the results, indicating a good quality of the computational mesh.

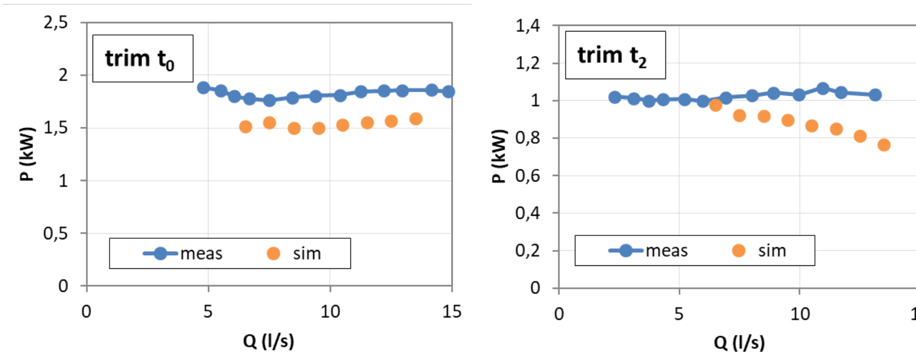

Fig. 6. Measured and simulated input power P. Speed n = 2900 min^{-1}.

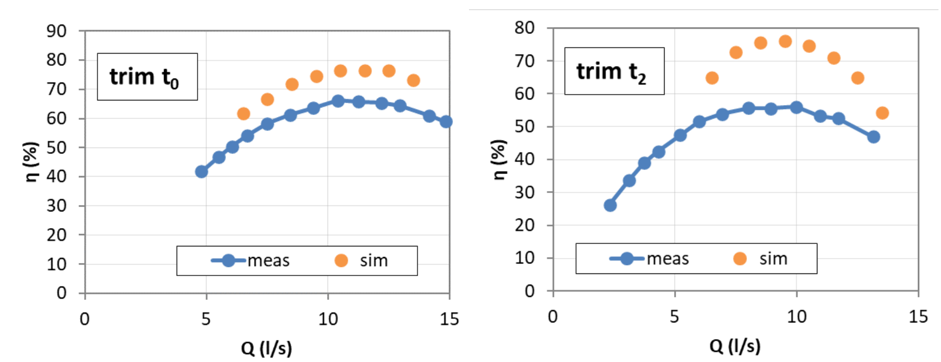

Fig. 7. Measured and simulated efficiency η. Speed n = 2900 min^{-1}.

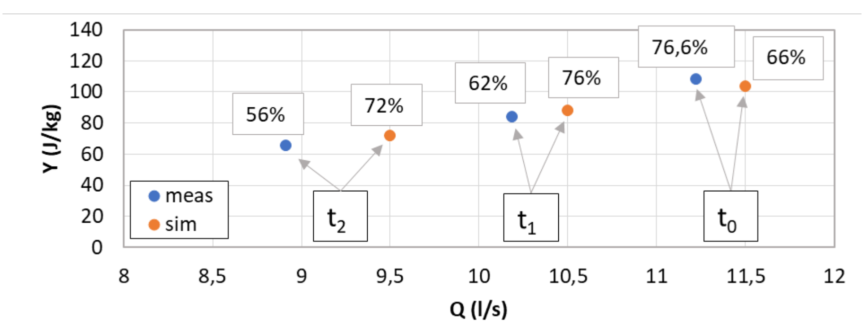

Fig. 8. Best efficiency points with maximum efficiencies. Speed n = 2900 min^{-1}.

In the case of the maximum trim t_2 the difference is larger and especially for smaller flows far from the BEP. The slope of the curves in the case of t_0 is in agreement, for the trim t_2 the calculated curve is steeper than the measured one.

Figure 6 shows the course of the input power P for the variants t_0 and t_2. In the case of the original impeller t_0, the measured power is higher than the calculated one, but the overall character of the course is identical (the values are only shifted by almost the same value with respect to each other). This difference can be explained by the fact that the experimentally determined input power includes the friction in the seal and bearings as well as the disc losses, so it must be higher than the calculated one. In the case of a trimmed impeller t_2 the calculated power input decreases with flow, although the measured power input is kept constant. The above difference is also due to the mechanical losses of the experimental device.
Figure 7 shows the efficiency η for the trims t_0 and t_2. The optimal flow rate determined by experiment and calculation is almost identical. The simulated efficiencies are hydraulic efficiencies and are therefore higher than the measured total efficiencies.
Figure 8 presents the measured and calculated positions of the BEPs of all three investigated impellers (t_0, t_1 and t_2). Numbers indicating the value of the maximum efficiency are attached to the points. The positions of the BEPs lie on a straight line. The largest contribution to the difference between calculation and measurement is the flow rate (for the t_2 trim it is about 7 %). It makes no sense to compare the efficiencies as we would be comparing hydraulic efficiencies with overall efficiencies.
Some studies provide rules that can be used to calculate the performance parameters for impeller trimming. For example, the authors in [15] give the following relations. They introduce the parameter λ as the ratio of the trimmed diameter D_{2i} to the original diameter D_{20}:

$$\lambda = \frac{D_{2i}}{D_{20}} \tag{1}$$

The calculation of the position of the BEP is done by applying the rules, which are modified affinity laws:

$$\frac{Q_i}{Q_0} = \lambda^{1.445} \tag{2}$$

$$\frac{Y_i}{Y_0} = \lambda^{2.090} \tag{3}$$

where Q a Y represent flow rate and specific energy in the BEP.

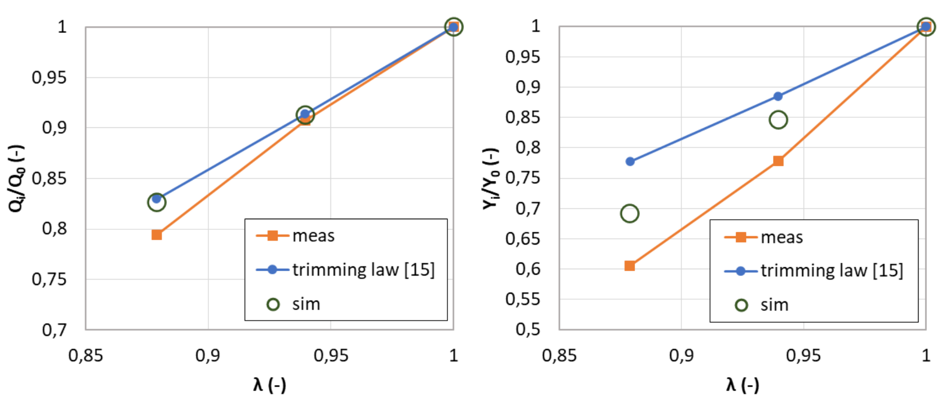

Fig. 9. Comparison with trimming law published in [15].

The index i takes the values 0, 1, 2 according to the impellers t_0, t_1 and t_2. The exponents in relations (2) and (3) are determined experimentally and are valid for multi-blade pumps.

Figure 9 shows a comparison of the measured and calculated ratios Q_i/Q_0 and Y_i/Y_0 versus the parameter λ with the results of relations (2) and (3). Decreasing the diameter of the single-blade impeller has almost the same effect on the change of the optimum flow rate as that of the multi-blade pump. However, the change in the optimum specific energy is more pronounced. Therefore, care must be taken when trimming the impeller of a single blade pump to ensure that the drop in height is not too great after the diameter has been reduced.

6 Conclusions

Analyses have been carried out regarding the influence of the single blade impeller trimming on the pump performance parameters. The research was carried out on a single blade pump with a maximum output of 2 kW. Experimental research methods were used as well as CFD methods. A total of three impellers with different trim sizes made of ABS plastic by method of Rapid Prototyping were tested.
The experimental and CFD simulation results are compared at the level of power curves. A good agreement between the specific energy curves for all impellers is achieved. The best agreement was in the case of the base size t_0 (non-trimmed impeller). The deviation increased with decreasing impeller diameter and was largest for the trim t_2. The measured power reached higher values (about 15 %) than the power simulated by CFD methods. This difference is due to the fact that neither mechanical losses nor hydraulic losses due to friction between the discs and the stator are included in the simulated power input. For the same reason, the efficiency found by simulation is then higher than the efficiency found by measurement.
The nature of the decrease of the parameters flow rate Q and specific energy Y as a function of the relative trim λ is presented in the paper. These results are compared with previously known results for multi-blade pumps. As far as the flow rate is concerned, the decrease in its proportional value is in agreement with the multi-blade pumps. However, the drop in the relative specific energy is much steeper than for conventional pumps. This result is due to the physical conditions of the flow in a single blade impeller, in particular the large slip and small blade angles. It is therefore not possible to automatically adopt the results and experience known from multi-blade pumps.

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References

1. Li, CX; Wang, SL and Jia, YK (2011). The performance of a centrifugal fan with enlarged impeller. *Energy Conversion and Management* 52 (2011) 2902–2910.
2. Gangipamula R, Ranjan P, Patil RS (2022). Study on fluid dynamic characteristics of a low specific speed centrifugal pump with emphasis on trimming operations. *International Journal of Heat and Fluid Flow* 95 (2022) 108952.
3. Khoeini D, Tavakoli MR (2018). Flow Characteristics of a Centrifugal Pump with Different Impeller Trimming Methods. *FME Transactions* (2018) 46, 463-468
4. Elyamin G.R.H.A., Bassily M.A., Khalil K.Y., Gomaa M.Sh (2019). Effect of impeller blades number on the performance of a centrifugal pump. *Alexandria Engineering Journal* (2019) 58, 39–48.
5. Sheng Y.S., Yu K.F., Ming J.W., Yun Q.X. (2013). Effects of impeller trimming influencing pump as turbine. *Computers & Fluids* 67 (2012) 72–78

6. Swain D, Engeda A. (2014). Performance impact of impeller blade trimming on centrifugal compressors. *Proc IMechE Part A: J Power and Energy* 2014, Vol. 228(8) 878–888.
7. Tan L, Yang Y., Shi W., Chen Ch., Xie Z. (2021). Influence of Blade Wrap Angle on the Hydrodynamic Radial Force of Single Blade Centrifugal Pump. *Appl. Sci.* 2021, 11, 9052.
8. Li L., Cao S., Li J., Nie R., Hou L. (2021). Review of Rotor Balancing Methods. *Machines* 2021, 9, 89.
9. Deng Q, Pei J, Wang W, Lin B, Zhang C, Zhao J. (2021). Energy Loss and Radial Force Variation Caused by Impeller Trimming in a Double-Suction Centrifugal Pump. *Entropy* 2021, 23, 1228.
10. Shadab M, Karimipour M, Najafi A F, Paydar R, Nourbakhsh S A (2022). Effect of impeller shroud trimming on the hydraulic performance of centrifugal pumps with low and medium specific speeds. *Engineering Applications of Computational Fluid Mechanics* 2022, VOL. 16, NO. 1, 514–535
11. Chen J, Shi W, Zhang D. (2021). Influence of blade inlet angle on the performance of a single blade centrifugal pump. *Engineering Applications of Computational Fluid Mechanics* 2021, VOL. 15, NO. 1, 462–475
12. Benra, F. K. (2006). Numerical and Experimental Investigation on the Flow Induced Oscillations of a Single-Blade Pump Impeller. Journal of Fluids Engineering 2006, 128, 783–793
13. Liu M, Tan L, Ca S. (2022). Performance Prediction and Geometry Optimization for Application of Pump as Turbine: A Review. Optimization for Application of Pump as Turbine: A Review. *Front. Energy Res.* 9:818118.
14. Quan H, Li R, Su Q, Han W, and Wang P (2013). Analysis on Energy Conversion of Screw Centrifugal Pump in Impeller Domain Based on Profile Lines. Hindawi Publishing Corporation *Advances in Mechanical Engineering Volume* 2013, Article ID 512523, 11 pages
15. Li W. G. (2011). Impeller Trimming of an Industrial Centrifugal Viscous Oil Pump. *Int J Advanced Design and Manufacturing Technology*, Vol. 5/ No. 1/ December – 2011