

Analysis of novel low specific speed pump designs

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Abstract. Centrifugal pumps with very low specific speed present significant design challenges. Narrow blade channels, large surface area of hub and shroud discs relative to the blade area, and the presence of significant blade channel vortices are typical features linked with the difficulty to achieve head and efficiency requirements for such designs. This paper presents an investigation of two novel designs of very low specific speed impellers: impeller having blades with very thick trailing edges and impeller with thick trailing edges and recirculating channels, which are bored along the impeller circumference. Numerical simulations and experimental measurements were used to study the flow dynamics of those new designs. It was shown that thick trailing edges suppress local eddies in the blade channels and decrease energy dissipation due to excessive swirling. Furthermore the recirculating channels will increase the circumferential velocity component on impeller outlet thus increasing the specific energy, albeit adversely affecting the hydraulic efficiency. Analysis of the energy dissipation in the volute showed that the number of the recirculating channels, their geometry and location, all have significant impact on the magnitude of dissipated energy and its distribution which in turn influences the shape of the head curve and the stability of the pump operation. Energy dissipation within whole pump interior (blade channels, volute, rotor-stator gaps) was also studied.

1. Introduction

Low specific speed centrifugal pumps are characterized, besides other problems, by low hydraulic efficiency. This fact is usually attributed to secondary flow in blade channels of impellers, where so called local eddy [1, 2] develops. This phenomenon causes increased losses due to recirculation and also higher streamline curvature at the impeller outlet, resulting in a significant decrease of the angle at which the liquid leaves the impeller (viewed in relative rotating space). As known from the Euler pump equation this deviation from the blade outlet angle poses reduction of attainable delivery head or specific energy.

Suppression of local eddy development is realized by various impeller design modifications presented in this paper. The new solutions include impellers with thick trailing edges and particularly impellers with thick trailing edges and recirculation channels. Mechanism of the hydraulic loss origin is different in these novel impellers, which puts emphasis on application of different approaches for their design and optimization.

Impellers with thick trailing edges of various shapes and with different position of the recirculation channels within the blade as well as the basic modifications of spiral case will be investigated in this contribution. The reason is that the impact of recirculation channels on the magnitude of specific



energy and hydraulic efficiency is substantial. The spiral case, as will be shown in the following analysis, has more significant contribution to the total hydraulic losses in comparison with the pumps with conventional impellers. Therefore its optimization can lead to increase of hydraulic efficiency [3]. Another important issue is influence of energy dissipation within the pump on stability of head curve. Contribution of individual working parts of pump to stability/instability will be illustrated for the case of this novel impeller design.

Complete description of the pump cannot be based only on integral characteristics (specific energy, efficiency) [4], but also on dynamic behaviour of the pump, which is based on evaluation of the pressure pulsations. Conclusions drawn from CFD simulations are finally verified by experimental testing.

2. Software tools for CFD analysis

To analyse the parameters of tested pumps software ANSYS Fluent 14.5.7 was used. The models of computational meshes were built in preprocessor GAMBIT 2.4.6. The parameters of unsteady CFD simulations are summarised in table 1. All results are valid for constant value of viscosity [5]. The pump with plastic impeller in real size was used for experimental verification of the results.

Table 1. Numerical simulation and calculation conditions.

Number of computational cells (impeller, spiral case, rotor-stator gaps)	Turbulence model and near wall modeling	Calculation mode	Boundary conditions
12 – 13 milions	realizable $k - \epsilon$	unsteady	Inlet: velocity inlet
-	non equilibrium wall function	sliding mesh	Outlet: pressure outlet
-	-	incompressible flow	-

3. Geometry of impellers and main operating parameters

Investigated pump has been designed for the parameters mentioned in table 2. Design of impellers is displayed in figure 1. In the initial simulations the same spiral case for all test cases was considered.

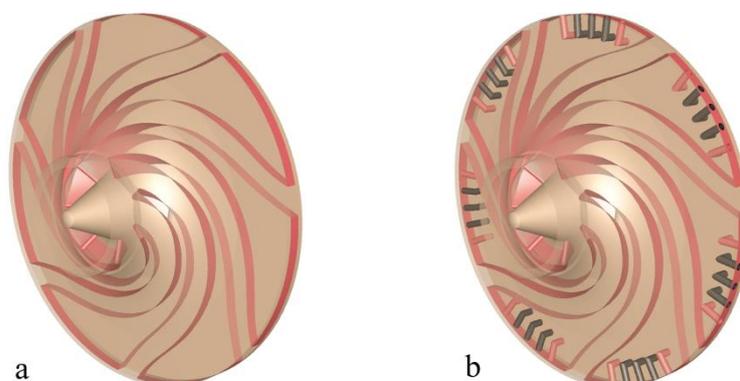


Figure 1. Impellers with thick trailing edges (a) and with / without recirculation channels (b).

Impellers were manufactured from polyamid using rapid prototyping. Wear rings are from bronze. Testing was done on a closed loop circuit with pressure taps in front and behind the pump and with induction flowmeter. Torque and rotational speed were measured using dynamometer.

Table 2. Basic parameters of pump and impellers.

Number of blades z	Average value of blade angle β_2 (°)	Impeller diameter D_2 (m)	Flow rate Q (l s ⁻¹)	Specific energy Y (J kg ⁻¹)	Rotational speed n (s ⁻¹)	Specific speed $n_b = nQ^{0.5}/Y^{0.75}$ (-)
7	63°	0.320	6.94	314	24.167	0.027

The basic geometry of the impeller with thick trailing edges (marked as TTE) remains unchanged as regards the shape of the blades, the diameter and width of the impeller. The only difference is in the design of the recirculation channels, which can be in the shape of letters I, L and T (see figure. 4). They are identically labelled in the figures. All recirculation channels have the same diameter and are drilled to the same depth in a radial direction. Same number of the channels in circumferential direction was selected for the initial comparison and evaluation of their effect on pump performance. Specific energy and efficiency curves will serve as the main criterion for verification CFD simulation against experimental results [6], see figure 2. (Note: flow rates are given in l/s, i.e. 1 l/s = 0.001 m³/s.) Reliability of the quantities, which cannot be measured (e.g. dissipation power in spiral case), will be judged according to agreement of integral characteristic curves.

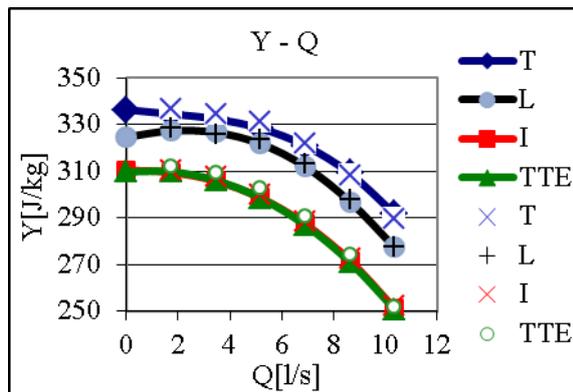


Figure 2. The specific energy curves obtained from the experiments (solid lines) and from CFD simulations.

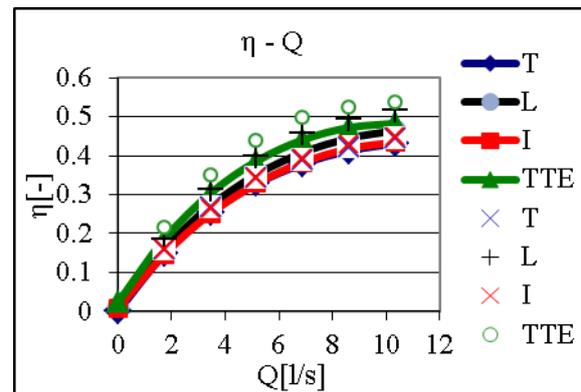


Figure 3. The curves of hydraulic efficiency (CFD) and total efficiency (experiment, solid lines).

It is obvious that in the case of L-shaped recirculation channels moderate instability around the shut-off point occurs. In the case of I-shaped channels and for TTE design without recirculation channels the head curve is just on the border of stability. On the other hand it is somewhat tricky to make a direct comparison of simulations and experiment for efficiency curves, see figure 3. Deformation of the plastic impeller due to axial forces occurred during experimental tests. Deflection of the pump discs resulted in friction of the impeller discs against the spiral case walls and deterioration of mechanical efficiency.

While T-shaped recirculation channels provide maximum values of specific energy it is compensated by the lowest values of total and hydraulic efficiencies. It also turns out that position of recirculation channels along the trailing edge is essential for their proper function, which is reflected on head and efficiency curves. Figure 4 therefore shows the cross-section of blade and illustrates how the recirculation channels are labelled.

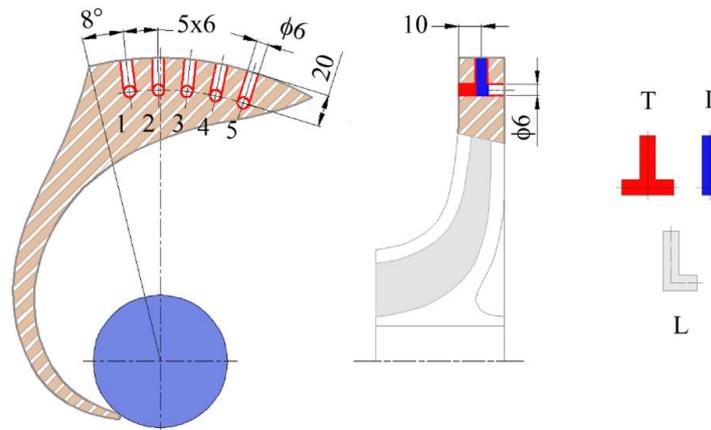


Figure 4. Cross-section of the blade.

4. Influence of shape and location of the recirculation channels

The influence of the recirculation channels distribution was tested for T-shaped channels. Numbers next to the head and efficiency curves indicate the active channels. The comparison of CFD simulation and experiment for the specific energy curves and efficiency curves is depicted in figure 5 and figure 6. Label T in the figures indicates that all T-shaped recirculation channels are active.

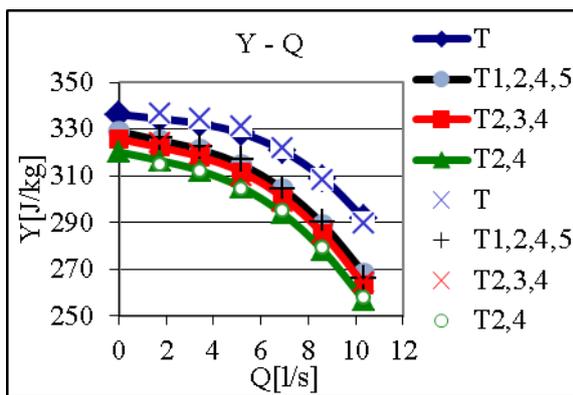


Figure 5. The specific energy curves for impellers with thick trailing edges and with T-shaped recirculation channels.

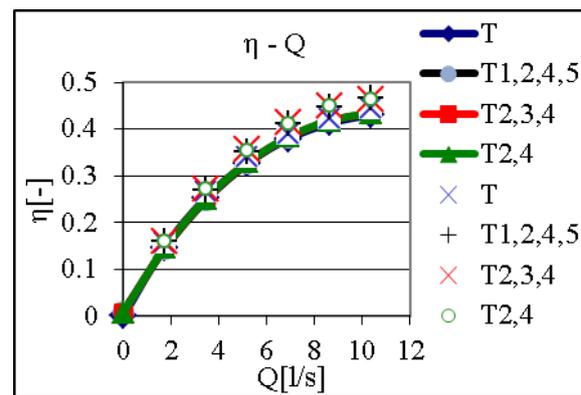


Figure 6. The curves of hydraulic efficiency and total efficiency for impellers with thick trailing edges and with T-shaped recirculation channels.

It is obvious that the specific energy increases with the number of recirculation channels, but the increase is not linear. We will study this fact in more detail. The differences in efficiency for different numbers of recirculation channels are very small. Noteworthy is also the fact that all specific energy curves are stable. More channel configurations and also different diameters of the channels, namely diameters of 3, 4, 5 and 6 mm were subsequently tested in the CFD simulations.

Because magnitude of the static pressure above the trailing edges of the impeller TTE is decreasing it is important to note that the recirculation channels have positive effect by reducing this pressure drop. The analysis and simulations show that the impact of the recirculation channels on the magnitude of the specific energy increases, the closer they are spaced. The maximum diameter of the channels is especially defined by width of the impeller, which is generally very small for low specific speed pumps.

If we find the center of the blade trailing edge and put in this place the channel to which we can symmetrically add another, increase of the achievable specific energy (in comparison with TTE) in the

pump for different operating points will follow approximately the equation (1), where S (mm^2) means the total size of the channels flow area.

$$\Delta Y = k_1 S^3 + k_2 S^2 + k_3 S \quad (1)$$

Wherein the corresponding coefficients can be determined from equations (2), (3) and (4).

$$k_1 = a_2 q^2 + a_1 q + a_0 \quad (2)$$

$$k_2 = b_3 q^3 + b_2 q^2 + b_1 q + b_0 \quad (3)$$

$$k_3 = c_4 q^4 + c_3 q^3 + c_2 q^2 + c_1 q + c_0 \quad (4)$$

Parameter q is defined by flow rate Q and by flow rate in the design point Q_{DP} (5). Coefficients are mentioned in table 3.

$$q = \frac{Q}{Q_{DP}} \quad (5)$$

Table 3. Values of coefficients.

	0	1	2	3	4
a	-2.18471E-06	-1.43690E-05	1.53705E-05	-	-
b	4.78645E-04	5.54435E-03	-5.52904E-03	1.09511E-03	-
c	1.66569E-01	-6.97903E-01	1.05437	-6.01932E-01	1.19654E-01

These equations correspond to the specific speed of tested pumps with the considered channels depth of approximately 20 mm (figure 4). The distances between the channels are given by the impeller design possibilities. In terms of the size of achievable specific energy it is reasonable to locate the recirculation channels uniformly from the center of the trailing edge symmetrically towards the main blade channels. This corresponds to the distribution of the static pressure above the trailing edge of impellers TTE and it is obvious that the lowest pressures can be seen just above the center of the trailing edge.

However, it is also important which shape of the recirculation channels should be chosen, whether shape I, L or T. Flow with characteristics of local eddy is formed in each type of these channels. Flow is not only influenced by diameter of the impeller and location of the channel inlet, but also by pressure in the spiral case and pressure between the impeller and the stator discs of the pump. Consequently the flow through the channels varies slightly depending on the impeller position in the circumferential direction respective to spiral nose (figure 7).

The inflow and outflow of liquid into the recirculation channels is relatively complicated and backflow occurs at each inlet and outlet from the respective channel. For this reason the theorem about mean value of integral calculus cannot be successfully applied for analytic solutions, because it is not clear how large is the cross-sectional area of the channel through which the liquid enters or leaves out. This fact may also influence the data obtained from CFD simulations.

From the specific energy curves (figure 2) it is apparent that there is slightly negative effect of the L-shaped recirculation channels on their stability. Conversely, the T-shaped recirculation channels have positive contribution to the head curve stability. The stability itself will be discussed hereinafter. Since to the worst results are achieved by impellers with I-shaped recirculation channels the rest of the investigation will be confined to L- and T-shaped recirculation channels. Their effect on the stability / instability of the specific energy curves relates to the distribution of the static pressure between the hub and the pump cover (stator disc) and between the shroud and the pump body. While filling of the L-shaped channel by liquid is dependent on the gap on hub side only, T channel is filled with liquid from both hub and shroud sides. This fact is most notably reflected in the vicinity of the shut-off point, where its effect can be observed on the average value of the specific energy (figure 8).

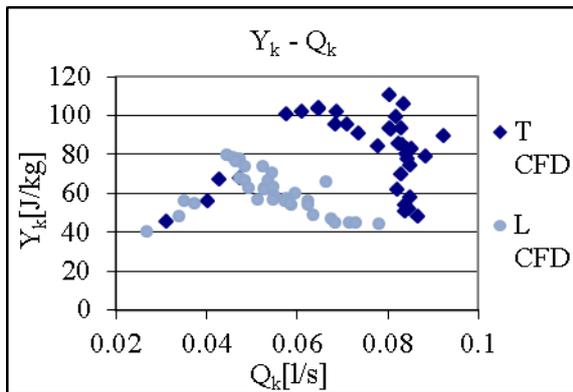


Figure 7. Specific energy close to the best efficiency point depending on the flow rate through the L-shaped and T-shaped recirculation channels.

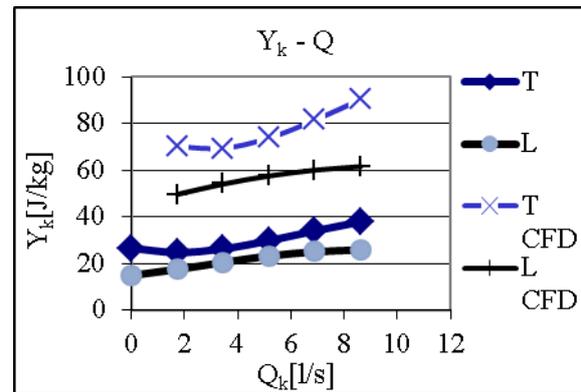


Figure 8. Average value of the specific energy of the L-shaped and T-shaped recirculation channels. (CFD simulations do not include influence of the spiral case in this case.)

Unfortunately, due to very complex flow, results of CFD simulation in the shut-off point are not very reliable. But it is just the shut-off point where T-shaped channel has the most pronounced effect. Therefore only a plot, which illustrates increase of the specific energy with flow area across the pump is presented, see figure 8.

Filling of recirculation channels by liquid and their impact on the specific energy of the pump is very closely related to axial forces acting on the impeller [7, 8]. However, compared to determining the magnitude of the axial forces it is necessary to achieve higher accuracies to predict the recirculation channels function. Unfortunately, in this case the usual equations determining [9] the distribution of static pressure in the gaps between the impeller and the stator of the pump fail. It is also problematic to determine the static pressure effect on the impeller in radial direction. In particular, the influence of impeller circumferential position cannot be analytically specified with sufficient accuracy and even small changes in pressure at the inlet or outlet of the recirculation channels cause a substantial change of the flow through the channel.

The size of flow rate through the recirculation channels also depends on the width of the gap between the impeller and stator (figure 9).

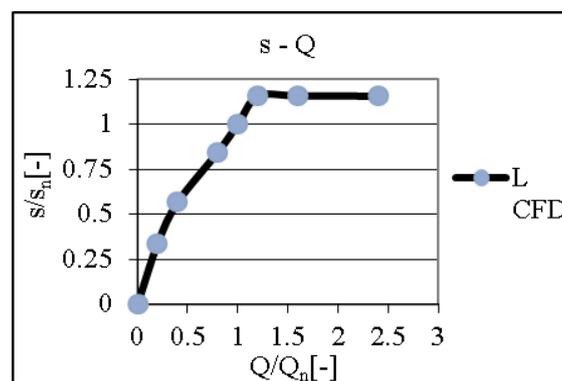


Figure 9. Width of a gap depending on flow rate through the recirculation channels.

Data included in this figure are valid for width of a gap on the side of hub and for sizes of L-shaped recirculation channels from figure 4. The flow in impeller corresponds to the design flow, nominal flow rate through the recirculation channels is 0.05425 l/s and nominal width of the gap is 2.5 mm.

The width of the gaps cannot be selected arbitrarily because it largely influences magnitude of disc friction losses.

The following two figures (figure 10 and figure 11) depict the influence of the inlet channel position that is expressed by the distance from rotation axis.

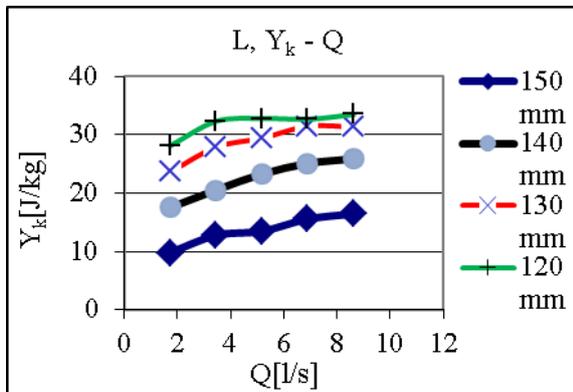


Figure 10. Influence of the position of the L-shaped recirculation channel inlet on increase of the specific energy in comparison with TTE.

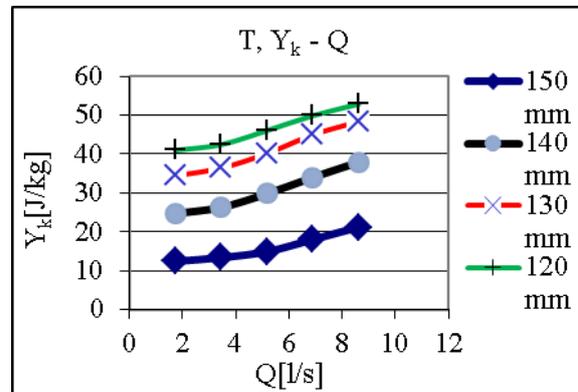


Figure 11. Influence of the position of the T-shaped recirculation channel inlet on increase of the specific energy in comparison with TTE.

5. Losses in the pump and the influence on stability of specific energy curves

The analysis of losses in a hydrodynamic pump enables to assess contribution of various parts of the pump on stability or instability of specific energy curves. This information then allows to make design modifications leading to elimination of the instability. The general stability criterion [9] says that the specific energy curve is stable when the following relation is valid over its whole range (6).

$$\frac{\partial Y}{\partial Q} < 0 \tag{6}$$

In order to have a better idea about the stability, it is necessary to introduce another criterion [10] based on dissipation power in shut-off point (7). However, this power has to be free of mechanical and volumetric losses. CFD simulations is a suitable tool for this purpose (figure 12).

$$\frac{\partial^2 2D_H}{\partial Q^2} > 0 \wedge \frac{\partial 2D_H}{\partial Q} < 0 \tag{7}$$

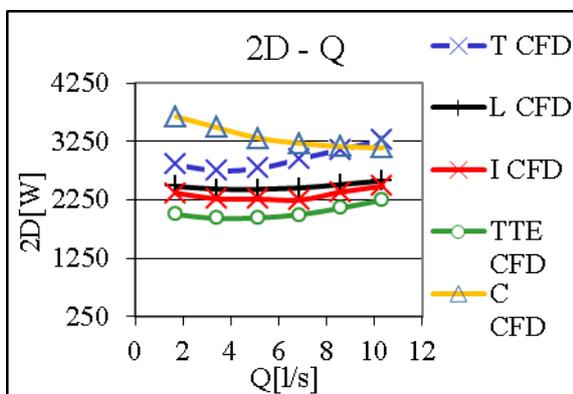


Figure 12. Dissipation power in the pump.

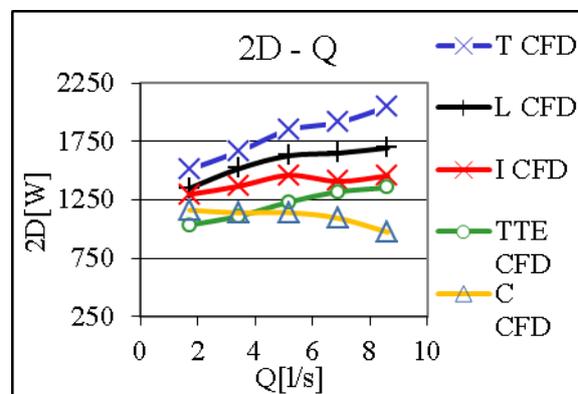


Figure 13. Dissipation power in spiral case.

For better description of the differences between the pumps with impellers with thick trailing edges and conventional impellers (i.e. impellers with blades of constant thickness), pump characteristics with conventional impeller labelled C are enclosed for comparison. The conventional pump has been designed for the same parameters as the impellers with thick trailing edges and has stable specific energy curve. Actual values of the specific energy or efficiency is not important, since the conclusions will be made upon the shapes of the curves and in sense of equation (7).

In the first step, the pump was divided into elementary parts, such as the blade channels, the spiral case and the gap between the impeller and the stator of the pump. Standard definition of dissipation function was used first to determine the size of the dissipation power (8).

$$2D_H = 2\eta \iiint_V v_{ij} v_{ij} dV \quad (8)$$

where v_{ij} is strain rate tensor. Errors in the evaluation of the dissipation function defined by (8) are affected by uncertainties in the formulation of fluctuating velocity components and by the subsequent derivatives. This approach is therefore inaccurate.

Due to problems with the exact expression of the strain rate tensor in CFD simulations, the balance of the total input and output powers was used as better definition (9).

$$2D_H = \sum_{i=1}^m P_{in\ i} - \sum_{j=1}^n P_{out\ j} \quad (9)$$

Dissipation power characteristics for spiral case are presented first (figure13). The most important feature, which can be drawn according to definition (7) is that the spiral case of pumps with impellers with thick trailing edges (either with or without recirculation channels) can contribute to instability of specific energy characteristics much more than in case of the conventional impeller C. The worst with respect to stability are the impellers with thick trailing edges and with L-shaped recirculation channels. However impact of the recirculation channels on dissipation power in spiral case is generally adverse for any type of the channels. The difference is reflected in other functional parts of the pump, into which the pump was divided for the evaluation. A slight improvement is observed only in variant I but this variant does not satisfy the required design parameters. A general problem that cannot be removed is the fact that close to the shut-off point [11] where the criterion (7) is valid, CFD simulations cannot provide the necessary data.

Because of differences between the impellers with thick trailing edges and conventional impellers it is foreseen that the suppression of so called local eddy will be very clearly reflected in the size of the dissipated power in blade channels and shape of $2D_H$ -Q curve (figure 14). Generally, only two curves can be discerned, which correspond to two basic impeller types. The stabilizing effect of classic blade channels is significant.

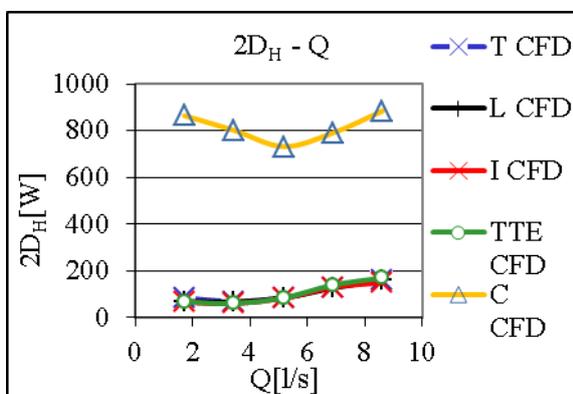


Figure 14. Dissipation power in blade channels.

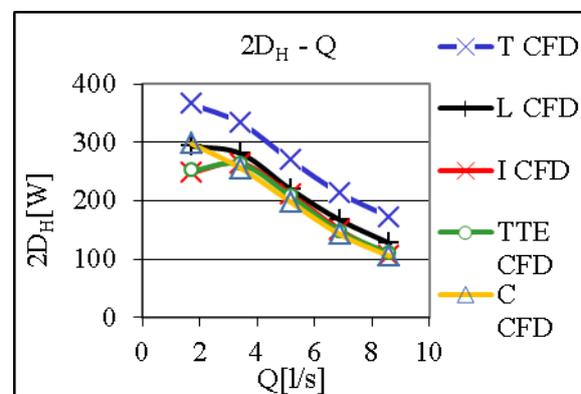


Figure 15. Dissipation power in the gap between the shroud and stator of the pump.

Another part of the analysis is focused on hydraulic losses between the shroud and the stator of the pump (figure 15) and losses between the hub and the cover of the pump (figure 16). It is observed that T-shaped recirculation channels can affect the size of losses in the gap in such way that the resulting characteristics of the specific energy will be stable, see figure 15 and figure 16. The gap between the hub and the pump cover is clearly a stabilizing element for all types of impellers. In the case of the conventional impeller the dissipation power was divided into two parts, because the impeller was partially opened on hub side.

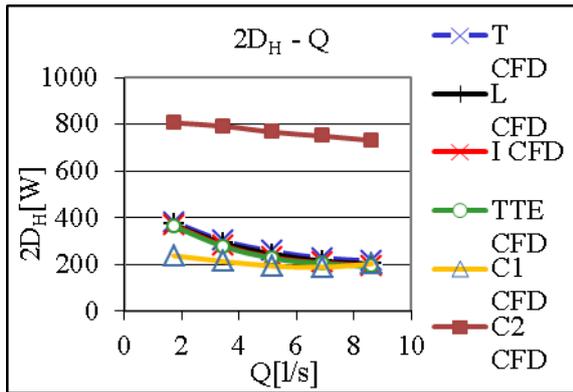


Figure 16. Dissipation power in the gap between the hub and cover of the pump.

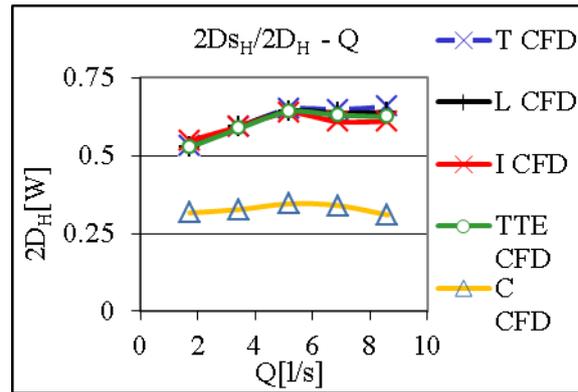


Figure 17. Proportion of dissipation power in spiral case in total dissipation power.

In view of differences between the pumps with conventional impellers and with the impellers with thick trailing edges it is interesting to see how the spiral case contributes to the total losses in the pump (figure 17). This is important because the spiral case contributes significantly to stability or instability, see figure 13. This figure introduces the adjustments on the spiral case which will be studied in the future, for possibilities see e.g. [12]. However, first the attention will be paid to question how the choice of impeller type affects the pressure pulsations [13] in the pump and subsequently its operation.

6. Static pressure pulsation

Static pressure is monitored in terms of pressure pulsations in a spiral case and in the whole pump interior. The amplitudes of pressure pulsations in the spiral case are monitored to make clear how the impellers with thick trailing edges influence the pressure field (figure 18 and figure 19). This view is also important in terms of hydraulic loss magnitude and impact on the stability of specific energy curves. The angle ϕ is the circumferential angle of the spiral case.

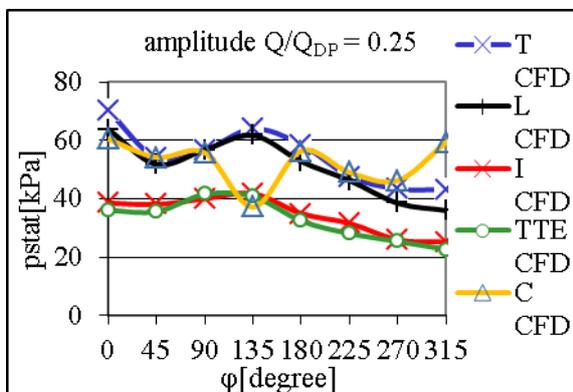


Figure 18. Amplitudes of pressure pulsations in spiral case close to the shut-off point.

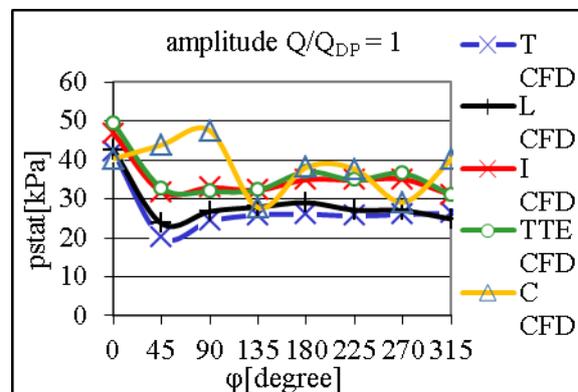


Figure 19. Amplitudes of pressure pulsations in spiral case close to the best efficiency point.

The amplitudes of the static pressure are lowest close to the shut-off point for the impellers with thick trailing edges and without recirculation channels and for the impellers with thick trailing edges and I-shaped recirculation channels. Near the design point the recirculation channels, specifically L-shaped and T-shaped, cause a reduction in the static pressure amplitude. From this point of view the conventional impeller seems to be the worst.

More important than the pressure pulsations in the spiral case are pressure pulsations in the whole pump interior. They will be monitored again close to the shut-off point and best efficiency point (figure 20 for two impeller revolutions and figure 21 for one impeller revolution). The angle φ_i represents angle in circumferential direction.

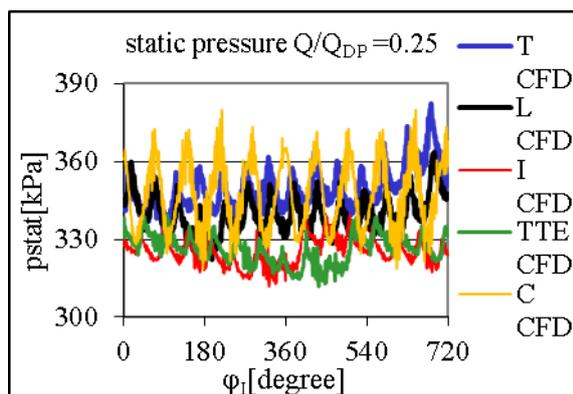


Figure 20. Static pressure pulsation in the whole pump interior close to the shut-off point.

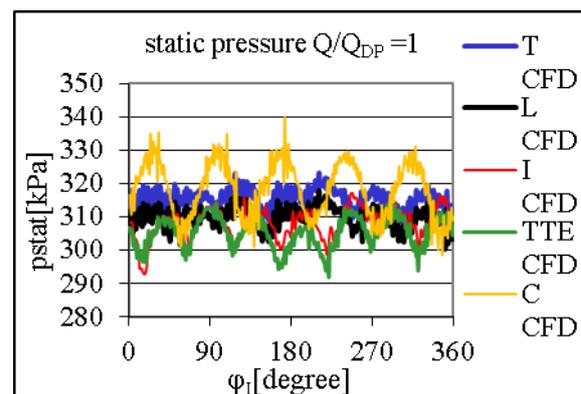


Figure 21. Static pressure pulsation in the whole pump interior close to the best efficiency point.

From figure 20 it is evident that the worst case is for the conventional impeller followed by the impeller with thick trailing edges with T-shaped recirculation channels. In contrast, in the best efficiency point (figure 21) the magnitude of the pulsations is favourably influenced by recirculation channels and even the impeller with thick trailing edges without recirculation channels has lower pulsations than the conventional impeller.

7. Conclusions

Extensive investigation of the impellers with recirculation channels leads to following conclusions:

- L-shaped and T-shaped recirculation channels are the only suitable and meaningful types. However, while L-shaped channels were sufficient to meet the required design parameters, they caused mild instability of the head curve. On the hand L-shaped channels provided the highest hydraulic efficiency, which was also confirmed by experiment.
- T-shaped recirculation channels contribute to stability of the head curve. T-shaped channels also ensure highest specific energy (i.e. delivery height). Unfortunately these positive properties are paid by lower efficiency.
- Resulting hydraulic parameters do not only depend on the shape of the recirculation channels, but of course also on their number, location and dimensions. CFD simulations showed that for the diameters of 3, 4, 5 and 6 mm it is possible to consider only the total flow area of the channels without significant error. Total flow area then enables to calculate total number of the channels. The recirculation channels should be distributed around the center of blade trailing edges, because the minimum of static pressure is situated just above the center. Dependence of the specific energy on pump discharge and on the flow area of the recirculation channels is presented in the paper.

It has been already stated that the L-shaped recirculation channels may be confronted with mild instability of specific energy curves. This instability can be evaluated on the base of the dissipation

power analysis. From the comparison of different impeller configurations it is evident that the spiral case is the origin of instability together with the gap between the shroud and the stator of the pump. It is also obvious that the dissipation power curve for flow within the gap between the shroud and the impeller can be influenced by the shape of recirculation channels. From this point of view, the worst configuration is the simple impeller with thick trailing edges (TTE). However, this drawback is balanced by more suitable dissipation power curve in a spiral case for TTE.

In terms of static pressure pulsations it was proved that the recirculation channels can reduce the pressure pulsation amplitudes in the design point compared to TTE impeller. The same conclusions apply to the pressure pulsations either in the spiral case or in the whole pump interior. The highest suppression of pulsations can be achieved by T-shaped recirculation channels. Conversely, close to the shut-off point the lowest pulsations are observed in the TTE impellers or impellers with thick trailing edges and with I-shaped recirculation channels. This fact is associated with the total area through which liquid can enter into the spiral case and this area is the smallest in TTE impellers.

To improve the properties of impellers with thick trailing edges and with L-shaped recirculation channels the modifications of spiral case should be considered, because the spiral case significantly contributes to instability of the specific energy curves as can be seen on dissipation power curves.

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