

# Influence of recirculation on Y-Q characteristic curve of hydrodynamic pump

Roman Klas<sup>1,a</sup>, František Pochylý<sup>1</sup> and Pavel Rudolf<sup>1</sup>

<sup>1</sup>Brno University of Technology, Faculty of Mechanical Engineering, Victor Kaplan Department of Fluids Engineering, Technická 2, 61669 Brno, Czech Republic

**Abstract.** Contribution is focused on discussion of different design modifications of the volute, impeller and rotor-stator cavity in case of very low specific speed pump with recirculation channels. Amount of the liquid flowing through the recirculation channels has significant effect on delivery height, stability of the head curve and hydraulic efficiency. Analysis of these effects is based on the evaluation of the dissipated power in different internal parts of the pump and for different flow rates. It has already been proved in our previous research that volute has substantial impact on stability of the head curve. It is apparent that similar effect can also be attributed to distribution and shape of the recirculation channels. This fact is connected with the inflow into the channels and with magnitude of the flow rate through the recirculation channels. Influence of mentioned parameters on recirculation is discussed in present paper.

## 1 Introduction

If it is necessary to use the classic low specific speed hydrodynamic pumps for specific condition, we cannot avoid some issues during their design. These issues are mainly related to low efficiency and sometimes are connected with reaching of the required parameters. So called local eddy is formed within the blade channels, which is a special form of secondary flow [1]. Vortices fill the area of the blade channel, which causes the streamlines at the outlet of the impeller to bend. Angle, at which fluid leaves the outlet of the impeller (within the rotating frame of reference) to the spiral case, is smaller due to aforementioned phenomena. It is obvious, from Euler equations, that this fact will also bring the decrease of the pump head. There will also occur an increase in hydraulic losses. Both of those facts have a negative impact on the value of the hydraulic efficiency of the pump [2].

We can avoid the above mentioned drawbacks by using of specific designs of impellers. Careful analysis of the flow and the effort to suppress an initialization of the secondary vortex brought the idea of an impeller with thick trailing edges or alternatively an impeller with thick trailing edges with recirculation channels. This design will result in a different flux inside the impeller and primarily a different way of the spiral case filling.

The first designs of the impellers with thick trailing edges came out recently and their options are not yet sufficiently explored with respect to the design and optimization of the parameters. Especially the influence

of recirculation channels on hydraulic efficiency and specific energy of pumps is not very well assessed. Therefore, the influence of modifications of the spiral case and the area between rotor and stator discs on flow rate through the recirculation channels will be examined. Flow rate through those channels has its significant influence especially on the hydraulic efficiency and total specific energy of the pump. Achievable specific energy will be observed also in terms of the Y-Q curve stability. As a tool for stability assessment of this curve, the dissipation function and its analysis with respect to power dissipated inside the individual parts of the interior of the pump was chosen. CFD methods were the main tool of the analysis. Results of CFD simulations were compared with experiments to prove its relevance [3, 4].

## 2 CFD simulation parameters

The basic tool for analysis of the interior of the pump and its parts was simulation software Ansys Fluent 14.5.7. Preprocessor GAMBIT 2.4.6 was used for modeling of the computational domain. The main parameters are listed in the Table 1.

## 3 Main parameters

Design parameters of tested pump are listed in the Table 2. Basic views on impellers with and without recirculation channels are shown in the Figure 1. There were used two types of spiral cases for all CFD

<sup>a</sup>Corresponding author: klas@fme.vutbr.cz

simulations, see the Figure 2. The appropriate notation is also contained within the figures.

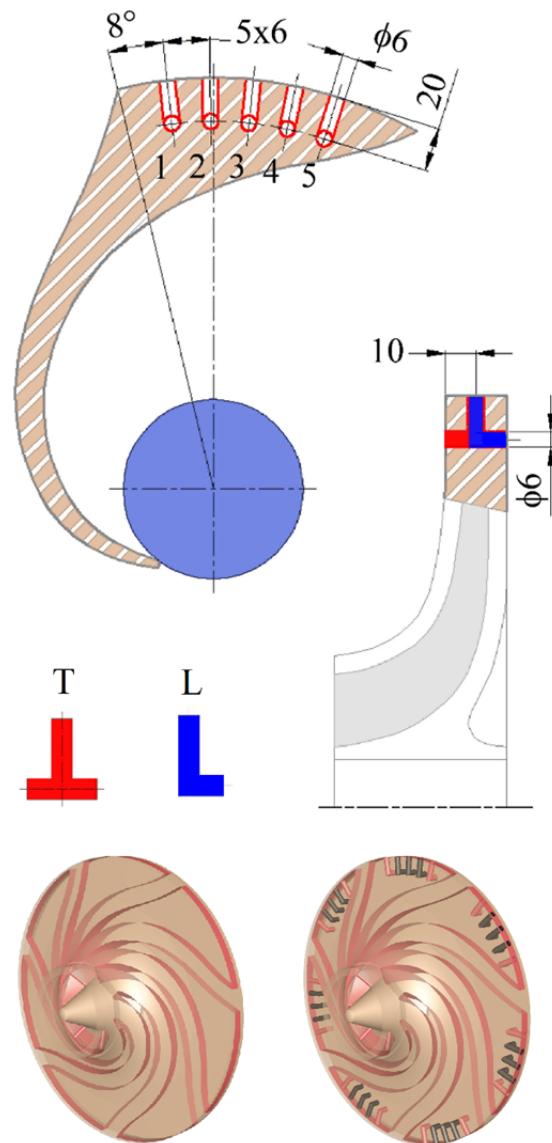
**Table 1.** Basic design parameters and description of numerical model.

	B	T	L
B	Impeller with thick trailing edges		
T, L	Impeller with thick trailing edges and recirculation channel T,L - shape of the recirculation channels		
z - number of blades	5	7	
$\beta_2$ (°) – average value of outlet blade angle	32	63	
$D_2$ (m) – impeller diameter	0.320		
Q (l s <sup>-1</sup> ) – flow rate	6.94		
Y (J kg <sup>-1</sup> ) – specific energy	314		
n (1 s <sup>-1</sup> ) – rotational speed	24.167		
$n_s$ (1 min <sup>-1</sup> ) – specific speed	33		
Number of computational cells	12 – 13 milions		
Turbulence model and near wall modeling	realizable k – $\epsilon$ non equilibrium wall function		
Boundary conditions	Inlet: velocity inlet Outlet: pressure outlet		
Calculation mode	unsteady, sliding mesh incompressible flow		

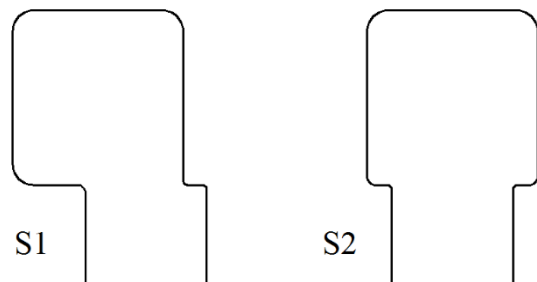
The flow areas in individual sections of the spiral case were preserved. The spiral case S1 was in fact designed for specific assembly conditions of the pump [5].

Both shown impellers differ from each other only in the built-in recirculation channels, but geometry of the main blade channels and also all main dimensions remain the same. Impeller without recirculation channels is marked as B. The recirculation channels could be T or L-shaped and are marked same as in the figures. All recirculation channels have the same diameter and in the direction perpendicular to the rotation axis they are drilled to the same depth. Number of the recirculation channels is the same for all designs to make the comparison easy. Correctness of the data, which were obtained from CFD simulation (e.g. dissipation) and

could not be verified by appropriate experiments, was authenticated primarily by using integral characteristics like hydraulic efficiency and specific energy curves. There is correspondence between experimental data and data obtained from CFD in case of spiral case S1 [5].



**Figure 1.** Impellers with thick trailing edges and with and without recirculation channels.



**Figure 2.** Cut through spiral cases.

In case of impeller with thick trailing edges with L-shaped recirculation channels there could possibly appear a moderate instability in the area of shut-off point. In case

of the impeller without recirculation channels (variant B) we are nearly on the stability border.

Although the impeller with T-shaped recirculation channels reached the highest values of specific energy, it also reached the lowest values of hydraulic efficiency. These results are clearly illustrated in [5].

#### 4 Dissipated power

With respect to the curves of specific energy or in other words the pump head, it is important to define criteria of stability appropriately, so that we can assess the impact of the recirculation on stability of those curves. As already mentioned, the main aim should be to assess the influence of changing the spiral case and the area between rotor and stator discs to curves of specific energy and their stability. A general criterion of stability (1) is defined by the following equation.

$$\frac{\partial Y}{\partial Q} < 0 \quad (1)$$

It turns out that it is beneficial to use the stability criterion based on the analysis of dissipated power  $2D_H$ . More information about this criterion is summarized in [6].

The most important advantage of so-defined criteria is the possibility to split the entire interior of the whole hydrodynamic pump into basic functional parts. Then a distribution of dissipated power in dependence on the flow through the pump can be determined within those parts.

This procedure enables to assess the contribution of individual components to the stability or instability of specific energy curves. Magnitude of the dissipated power is defined e.g. in [5].

Unfavorable fact is that the magnitude of the rate-of-strain tensor  $v_{ij}$  could not be determined precisely enough through CFD simulations, see [5]. Therefore, it is appropriate to use the difference of input and output powers instead to determine the magnitude of the dissipated power.

In addition to this, it is necessary to consider a significant fact. The required data, which are close to the pump shut-off point, cannot be obtained by CFD simulations.

The dependencies of specific energy and efficiency are plotted for different impellers with the spiral case S2. Figure 3 shows curves of specific energy and Figure 4 represents curves of hydraulic efficiency. However, comparison between CFD simulations and laboratory experiments for the spiral case S2 is not available.

Regarding to the maximum attainable specific energy or delivery height it is obvious, that a change of design parameters did not occur. There is one interesting fact connected with the curve of specific energy of the impeller with L-shaped recirculation channels. There can be seen some shift of this curve, which causes the curve of specific energy to appear more stable than it was shown in case of spiral case S1 [5]. Values of hydraulic efficiency remained preserved in the design point. The

shift to an area of higher hydraulic efficiency occurred in the area of higher flow rates.

Distribution of the power dissipated within the pump interior is shown in the Figure 5 and 6. This interior will be separated into the following regions for further assessment: spiral case; blade channels; area between shroud and the static parts of the pump and finally an area between hub and the cover of the pump. The contribution of aforementioned regions to stability of specific energy curves will be assessed afterwards, at first the curves of dissipated power will be analyzed.

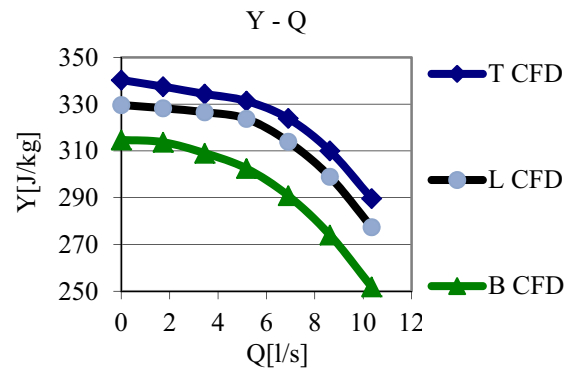


Figure 3. Curves of specific energy, spiral case S2.

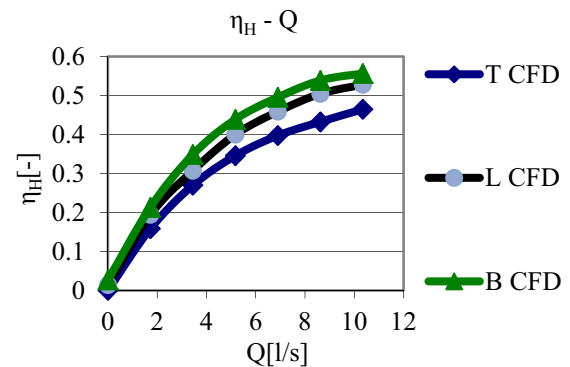
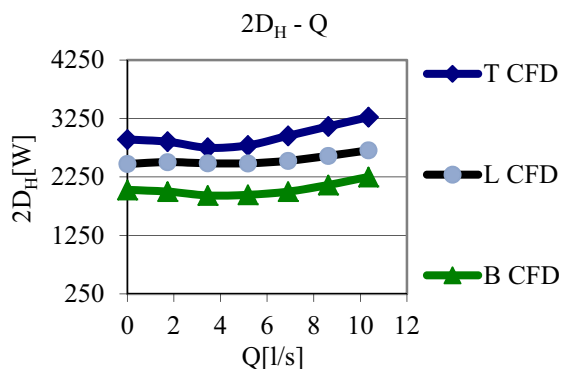


Figure 4. Curves of hydraulic efficiency, spiral case S2.

The reasons for modifications of the spiral case S1 to the spiral case S2 (see Figure 2) can be found in [5].

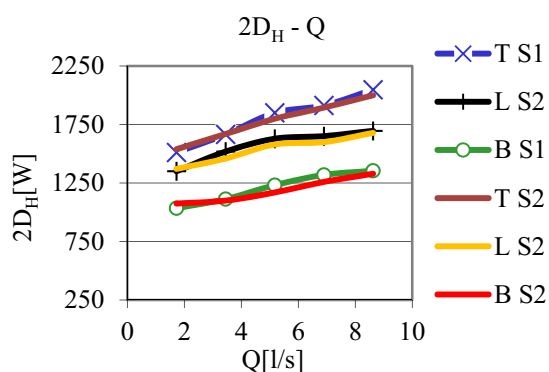
Regarding to the condition, there is a significant contribution to the instability of specific energy curves caused by the combination of spiral case S1 and impellers with thick trailing edges with or without recirculation channels (see Figure 6). Regardless of whether this curve is stable or not, recirculation channels are designed as L-shaped and T-shaped. In case of spiral case S2 and its combination with impeller with thick trailing edges without recirculation channels and impeller with thick trailing edges with L-shaped recirculation channels, there is slight contribution to the stability observable. This corresponds to conclusions in Figure 3. It appears that the correct design of the spiral case could have a decisive influence on the final quality of the whole design [8]. Significance of this is obvious from comparison of the power dissipated inside the spiral case S2. It is necessary to keep in mind, that the flow inside the spiral case is considerably affected by the flow in the impeller and

conversely that individual effects are to some extent coupled [9]. Subsequently, dissipated powers in remaining regions of the pump will be plotted.



**Figure 5.** Power dissipated within the pump with the spiral case S2.

The following section provides curves of power dissipated within spiral cases [8] S1 and S2 (see Figure 7).



**Figure 6.** Power dissipated within spiral cases S1 and S2.

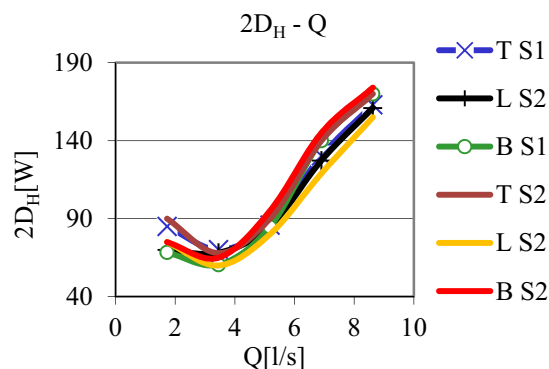
Contribution of blade channels to the stability of specific energy is very low for the impellers with thick trailing edges (see Figure 7). Differences in dissipated powers are negligible also for different spiral cases S1 and S2. It can be demonstrated that when discussing the classic impellers, the positive effect, which leads to stability of specific energy curves, is crucial. Negative impact of spiral case on stability is also obvious (see for example [7]).

It remains to assess the magnitude of dissipated power in the gap between impeller shroud and static parts of the pump (see Figure 8) and in the gap between impeller hub and the cover of the pump (see Figure 9).

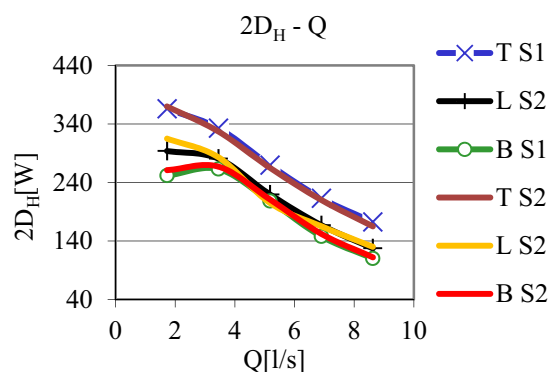
The influence of the spiral case S2 on the impeller with thick trailing edges with L-shaped recirculation channels and the flow through this gap could be obtained from the analysis of the dissipated power in the gap, see Figure 8. Changes were caused by an increase of recirculation inside of the spiral case. Similar effect can be observed also in T-shaped recirculation channels.

By contrast, the gap between the hub and static parts of the pump has a positive impact on stability for all

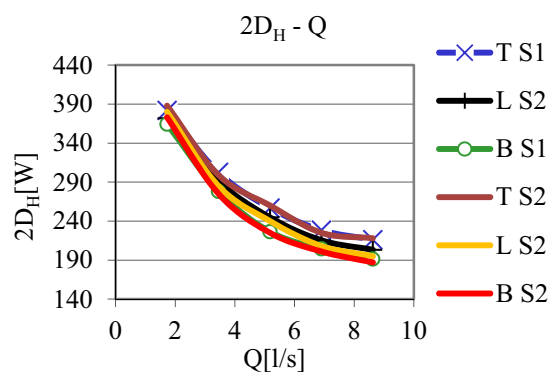
types of impellers regardless of the shape of the spiral case.



**Figure 7.** Powers dissipated inside blade channels of pump with spiral cases S1 and S2.



**Figure 8.** Power dissipated within the gap between impeller shroud and static parts of the pump with spiral cases S1 and S2.



**Figure 9.** Power dissipated within the gap between hub and static parts of the pump with spiral cases S1 and S2.

### 5 Modifications of the spiral case on the side of hub

Another spiral case, S3, was designed to assess the effect of recirculation on curves of hydraulic efficiency and specific energy. This spiral case is slightly wider than the spiral case S2 (see Figure 10). Cross-sectional areas remain preserved in the majority of sections. Except these changes, the gap on the hub side was extended up to

depth of 30 mm, see Figure 11. Original size of the gap was 2.5 mm and it corresponds to position no. 0. Subsequently, the gap was gradually enlarged always by 0.5 mm. The aim was to assist the recirculation of fluid in the impeller with T-shaped recirculation channels and then to verify possibilities of the influence on the size of the hydraulic efficiency. This impeller was also the only one which was tested in this part of work. Width of the gap could not be changed completely arbitrary due to its connection to the magnitude of the hydraulic losses [1].

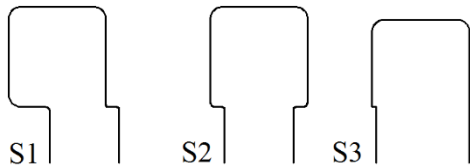


Figure 10. Sections of the spiral cases S1, S2 and S3.

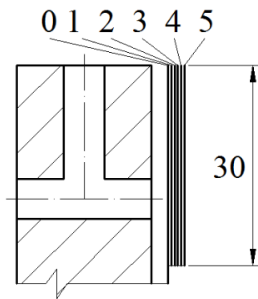


Figure 11. Gap between the hub and the cover of the pump.

Data were evaluated only on the basis of CFD simulations. First of all, there are the curves of specific energy for the impeller with T-shaped recirculation channels and the spiral case S3 presented. Labelling of the width of the gap corresponds to the Figure 11.

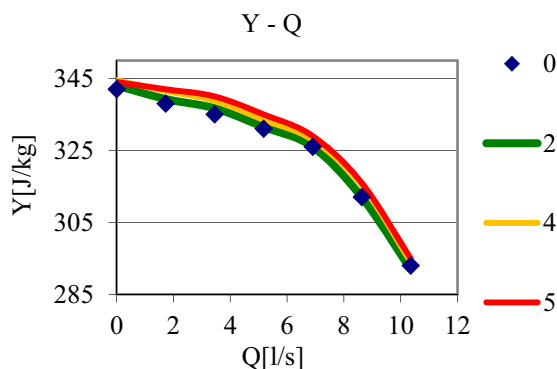


Figure 12. Specific energy inside the pump with the impeller with T-shaped recirculation channels and the spiral case S3.

According to curves shown in Figure 12 it appears that there is minimal impact of spiral case width on the specific energy. The only noticeable influence is connected to positions 4 and 5. There were observed some changes close to the shut-off point of the pump. These were connected with a decrease of hydraulic resistance on the inlet to the recirculation channels on the side of the hub. This caused the characteristics of pump

with impeller with T-shaped recirculation channels to resemble more to the characteristics of the pump with impeller with L-shaped recirculation channels.

The similarity with described phenomena is also observable in curves of hydraulic efficiency, see Figure 14.

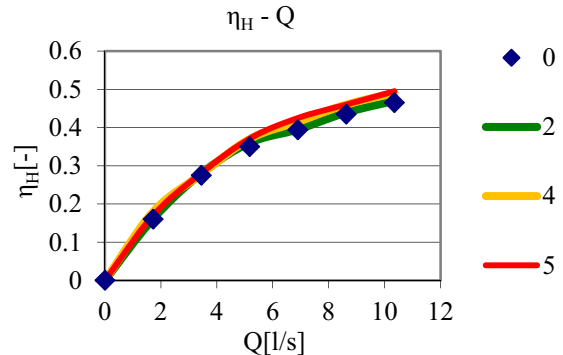


Figure 13. Hydraulic efficiency for the pump with impeller with T-shaped recirculation channels and with spiral case S3.

Increase of mechanical energy of the pump with gaps on the side of the hub 4 and 5 produces only slight increase in hydraulic efficiency compared to pumps with spiral cases S1 and S2 and original width of the spiral case.

## 6 Pressure pulsations within the pump

CFD simulations were used to obtain information about the size of pressure pulsations inside the pump. Pressure differences between suction and discharge were determined near the cut-off point and design point of the pump for spiral cases S1 [5] and S2.

Pumps with spiral case S2 also show similar results, see Figure 14 and Figure 15. The worst results are connected with impellers with T-shaped recirculation channels. While impellers with thick trailing edges without recirculation channels appears to be the best.

There is observable increase in the pressure pulsations nearby the design point of pumps with impellers with thick trailing edges without recirculation channels. On the contrary, impellers with thick trailing edges with recirculation channels shows significant decrease in static pressure pulsations.

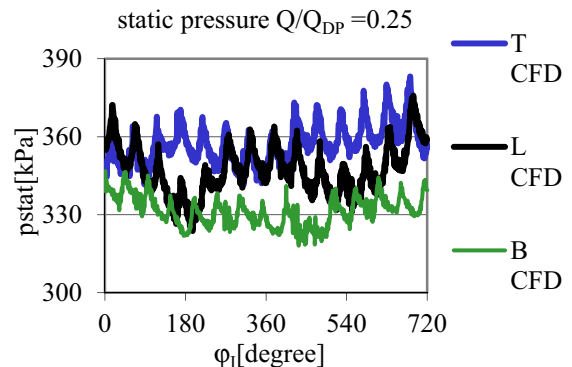
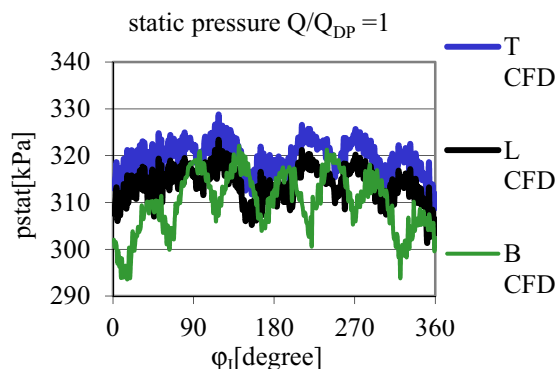


Figure 14. Pulsations of static pressure inside the pump with spiral case S2 in close proximity to shut-off point.



**Figure 15.** Pulsations of static pressure inside the pump with spiral case S2 in close proximity to design point.

Although, amplitudes of the static pressure are lower in these cases, conclusions about the properties of individual impellers near the cut-off point and around of the design point apply.

## 7 Conclusions

It can be summarized that regarding to stability of specific energy curves of the impellers with thick trailing edges, the best results were obtained for impellers with T-shaped recirculation channels. However, the total and hydraulic efficiency is smaller compared with other types of impellers. This fact was not significantly influenced even after some modifications were made to spiral cases S2 and S3, where the width of the gap between the impeller hub and the cover of the pump was increased. There was marginal increase of hydraulic efficiency for particular gap width and for enhanced inflow to recirculation channels, but the change was very small. Enlarging of the gap width on the side of the impeller hub improved the inflow into the recirculation channels, thus the impeller with T-shaped recirculation channels started behaving like the impeller with L-shaped recirculation channels in terms of specific energy curve stability.

Modification of spiral case S1 to the spiral case S2 brought us the improvement of the slight instability of specific energy curve for the impeller with L-shaped recirculation channels. The general significant contribution of the spiral case to possible instabilities of specific energy curves of hydrodynamic pumps was confirmed.

It was found that the gap between the impeller shroud and static parts of pump has a substantial impact on the stability. On the other hand, the impact of the gap between the impeller hub and the cover of the pump on the rate of recirculation (the shape of recirculation channels) corresponds to the course of specific energy (Euler turbomachinery equation). Recirculation enhances the stabilizing effect. The effect of stabilization is further reinforced by the optimization of the spiral case (change from S1 to S2), but the change is not significant from a quantitative point of view.

Effect of blade channels is only slight with respect to stability of the curves of specific energy.

Pulsations of static pressure in the interior of the pump close to the shut-off point are increasing while using the impellers with thick trailing edges with the highest rate of recirculation. This fact mainly concerns the T-shaped recirculation channels and it is related to the secondary flow rate within the spiral case.

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