CFD ANALYSIS OF MIDDLE STAGE OF MULTISTAGE PUMP OPERATING IN TURBINE REGIME

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This work deals with the CFD analysis of flow in the middle stage of the radial-flow multistage pump operating in turbine regime. The pump specific speed nq is equal to 23. Two complete stages are analysed with fully unsteady simulation to avoid the influence of boundary conditions and the position of rotor and stator parts on the results. The ANSYS CFX commercial CFD package is used to solve the three-dimensional Reynolds-averaged Navier-Stokes equations using Menter's SST turbulence model.

Keywords: *CFD analysis, multistage radial-flow pump, turbine regime*

1. Introduction

Small hydropower plants are very attractive for producing energy from water in these days. The main problem for developing such units is the high price of installation as well as the time needed for their construction and maintenance. One solution of this problem is the extensive use of pumps that can operate in the reverse mode as turbines. They are easily found on the market at much lower price compared to a standard turbine. The maintenance of the pump is very low and the pumps can be operated by non-highly trained people. Pumps operated in the reverse mode are usually of the single stage type with low or medium delivery head and the description of flow inside such pump-turbines can be found in literature, e.g. [1]–[4]. It is possible of course to use also the multistage pump in the turbine regime in the case when sufficiently high head is available, nevertheless there is lack of information about applications of the multistage pumps in the reverse mode, especially about the flow phenomena occurring inside these pumps when operating as turbines.

This study deals with the flow phenomena occurring inside the middle stage of the radial-flow multistage pump operating in the turbine regime and the corresponding flow characteristics. These phenomena are compared to the ones obtained from the CFD analysis of the same stage operated in the pump regime. Because the real pump has been tested in both the pump and turbine regimes in the test facility of the SIGMA Research and Development Institute in Lutin, the calculated integral characteristics can be compared to the experimental ones. The aim, within our projects, is to get enough information about flow patterns inside the multistage pumps so that we could modify the pump parts in order to be able to operate at high efficiencies [5]–[8].

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2. Pump model

The modelled multistage pump is of the radial-flow type, with the specific speed nq equal to 23. As it is supposed that the pump will operate with four or more stages, the influence of the inlet and outlet casings on the overall pump integral characteristics is relatively small and the numerical analysis can concentrate on the flow in the middle stage of the pump. The stage consists of the impeller with six blades and the stator of the channel type with eight channels (Figure 1). Two versions of the impeller are taken into account: the first one with the full design diameter of 0.25 m and the second one with the diameter reduced (including the hub and shroud discs) to 0.22 m.

The ANSYS CFX commercial CFD system has been applied to solve the fully unsteady three-dimensional Reynolds-averaged Navier-Stokes equations together with the Menter's SST model of turbulence. Two complete stages are analysed to avoid the influence of boundary conditions on the results. Moreover, a straight piping has been added in front of the first impeller and an additional impeller has been placed behind the second stage, followed with a vaneless diffuser with counter-rotating walls (Figure 2). The computational grid is block-structured and represents approximately 1.4 mil. nodes. In the pump regime the flow rate and the flow direction are prescribed at the front of the straight piping. At the end of the vaneless diffuser the average static pressure is set. Between the rotor and stator parts there are six interfaces: three rotor-stator interfaces and three stator-rotor ones (Figure 3). In the turbine regime the boundary conditions are similar, but the flow rate and the flow direction are prescribed at the end of the vaneless diffuser and the average static pressure is set at the front of the straight piping.

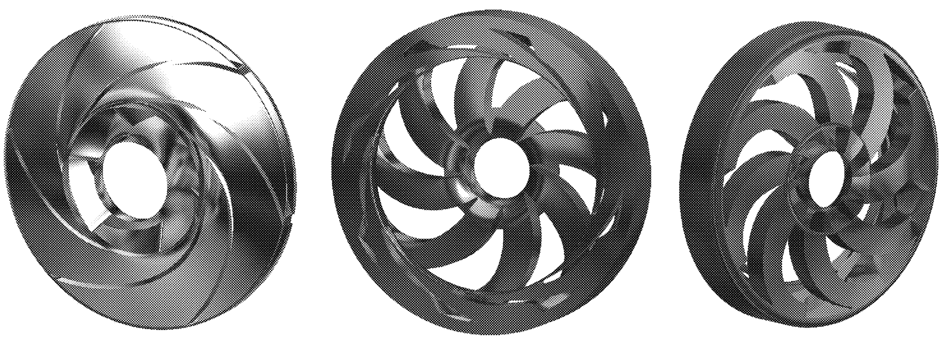

Fig.1: 3D geometry of the stage rotor and stator

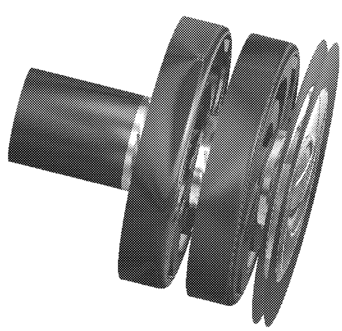

Fig.2: Computational domain

3. Results

The flow analysis has been carried out for several rotational speeds; nevertheless this paper shows only the results for the rotational speed of 1000 rpm. Seven flow rates are used to obtain the flow characteristics in the supposed operating range both for the pump and turbine regimes. In the case of the pump regime the flow rate changes from 6 to 17 l/s and the optimal rate of flow is 10.8 l/s. In the case of the turbine regime the flow rate changes from 8.3 to 23.6 l/s. Here the optimal rate of flow is 14.9 l/s for the full impeller diameter and 13.8 l/s for the reduced impeller (Figure 4). The peak hydraulic efficiency of the stage in the pump regime is 83.5 %.

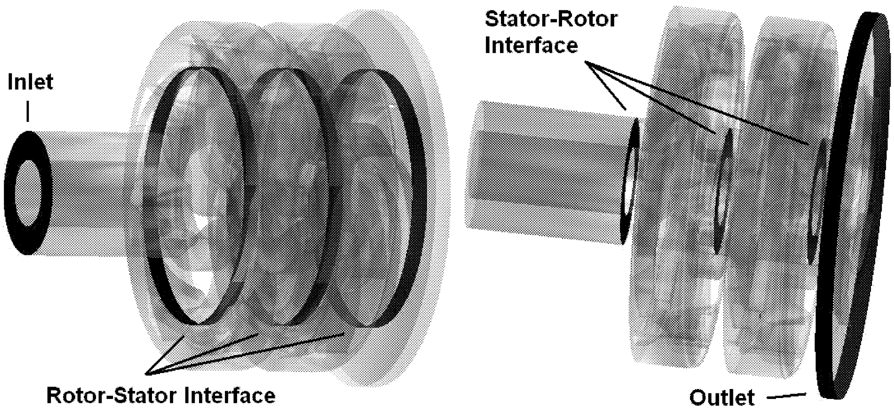

Fig.3: Boundary conditions in the pump regime

For the full scale impeller the peak hydraulic efficiency of the stage in the reverse mode is 90.5 %; the stage with the reduced impeller has slightly lower hydraulic efficiency of 90.1 %. The calculated performance can be (at least approximately) compared to the one obtained experimentally. The experimentally obtained head and overall efficiency of one stage of the machine operating in the reverse mode shown in Figure 4 are obtained from the data measured with the four stage machine. That is to say that the CFD analysis gives the hydraulic efficiency while the experimental data represent the overal efficiency including disc and mechanical losses. So the difference of about 13 % between the calculated and measured efficiency corresponds very well with the estimation of losses which are not included into the theoretical study. The shift of the measured optimal flow rate to the right (comparing to the calculated data) can be explained by the volume losses in the wearing rings.

Though the presented analysis based on the pump middle stage is suitable for the estimation of the pump (or turbine) head and hydraulic efficiency as well as for the visualization of flow phenomena in the middle stage, it is not suitable for the prediction of pump (or turbine) cavitation properties, because these properties depend strongly on the design of the pump inlet part. That is why we do not analyse the cavitation properties like cavitation inception or NPSH curves in this study.

To make the analysis of the flow phenomena inside the stage easier, two planes perpendicular to the axis of revolution have been introduced. The first one ('Plane1') is located at mid-height of the return guide channels of the first stator; the second plane ('Plane2') can be found between the hub and shroud discs at the outlet of the second impeller (Figure 5).

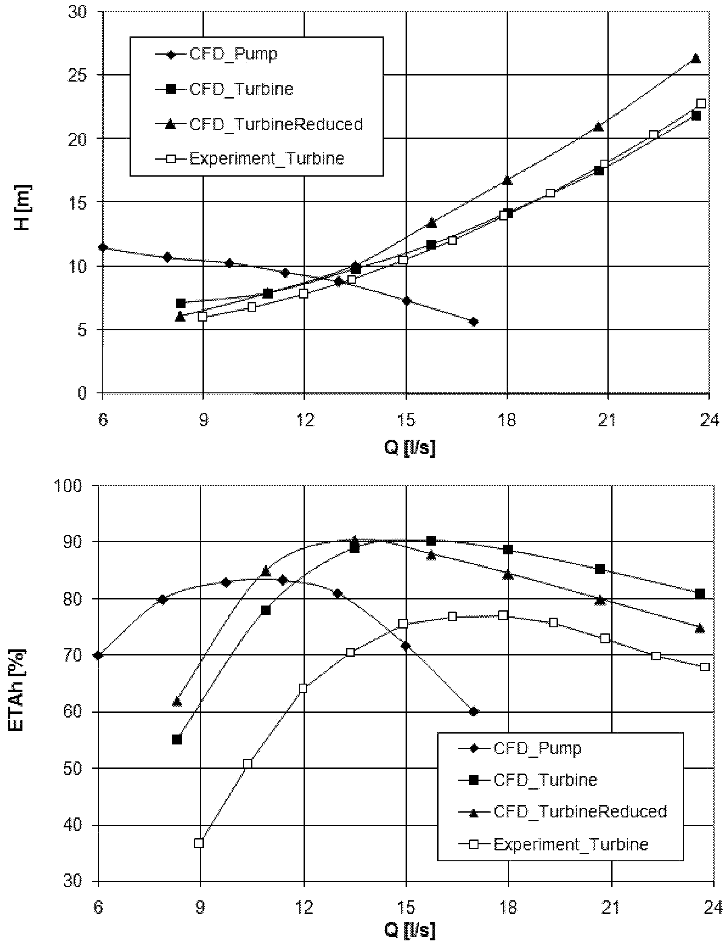

Fig.4: Calculated head and hydraulic efficiency in the pump and turbine regimes and measured head and overall efficiency in the turbine regime

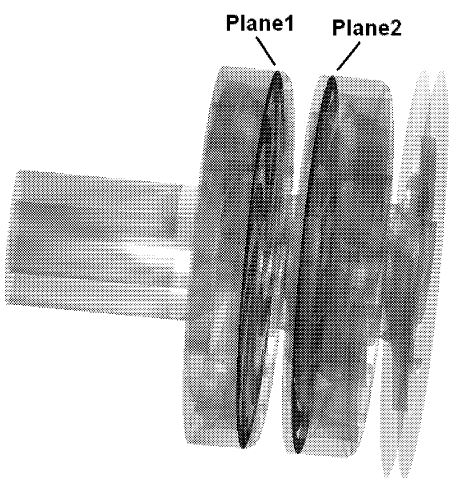

Fig.5: Definition of Plane1 and Plane2

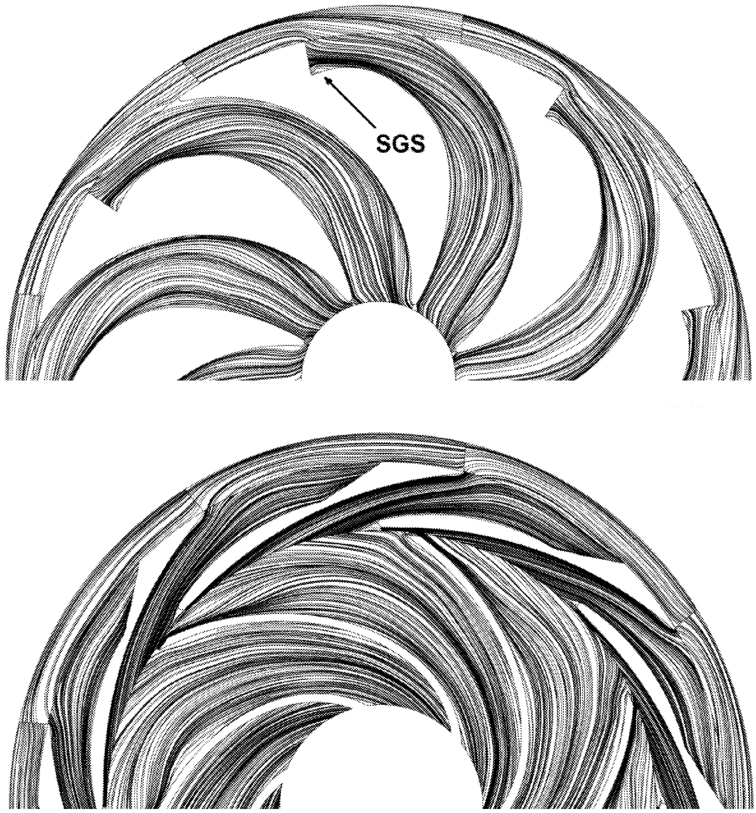

Fig.6: Streamlines on Plane1 and Plane2; turbine mode, full impeller, $Q = 15.75\,l/s$; SGS means a small global separation

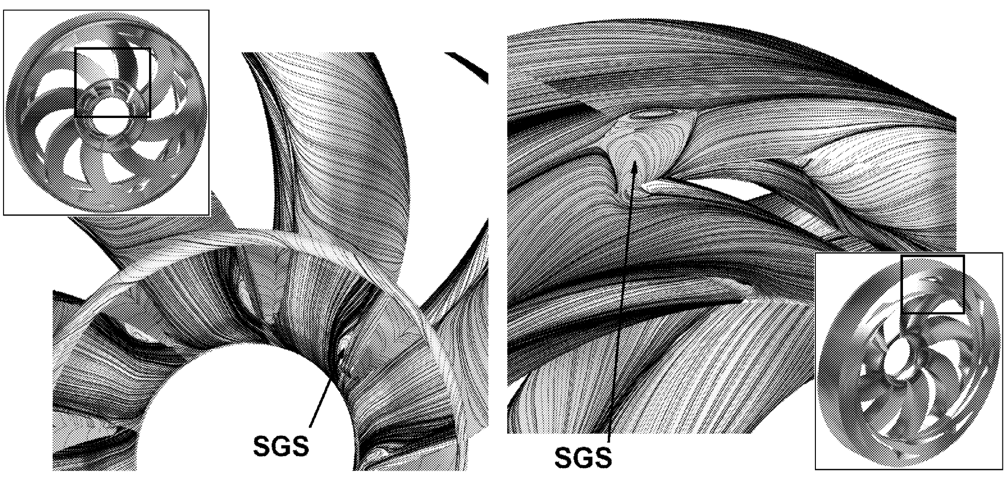

Fig.7: Separations in the stator; turbine mode, full impeller, $Q = 15.75\,l/s$; SGS means a small global separation

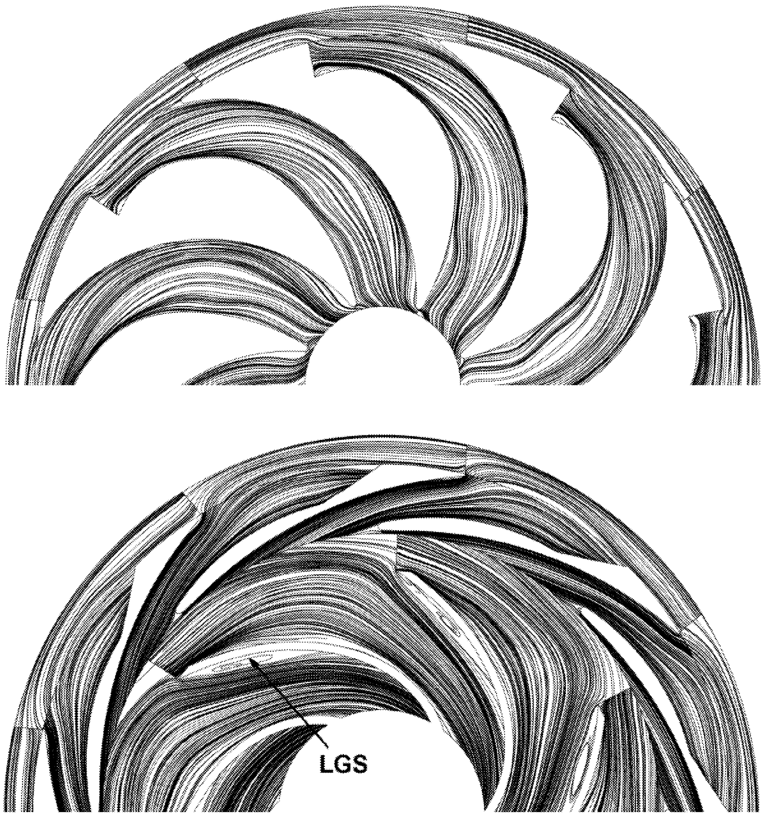

Fig.8: Streamlines on Plane1 and Plane2; turbine mode, reduced impeller, $Q = 13.5\,l/s$; LGS means a large global separation

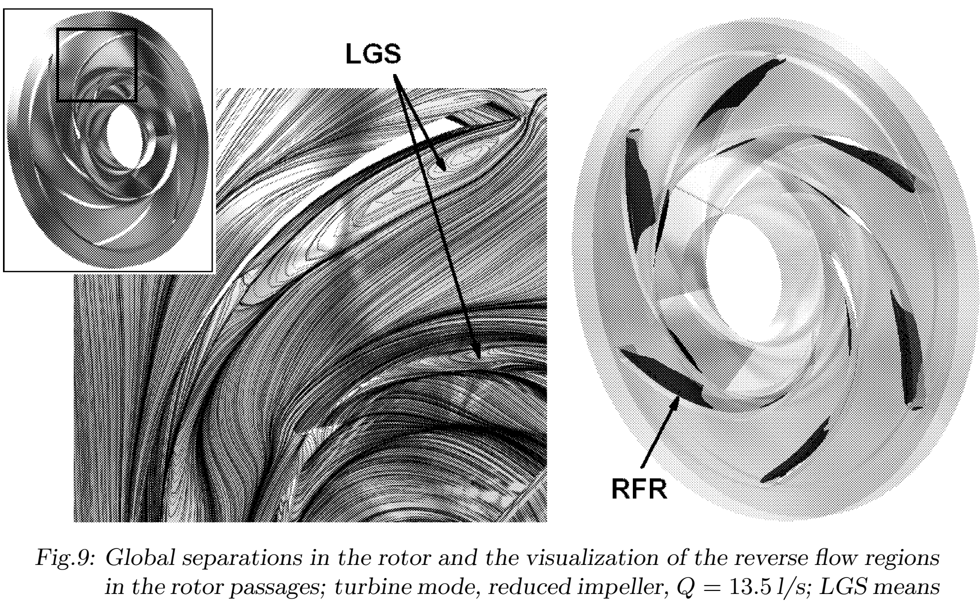

Fig.9: Global separations in the rotor and the visualization of the reverse flow regions in the rotor passages; turbine mode, reduced impeller, $Q = 13.5\,l/s$; LGS means a large global separation, RFR means a reverse flow region

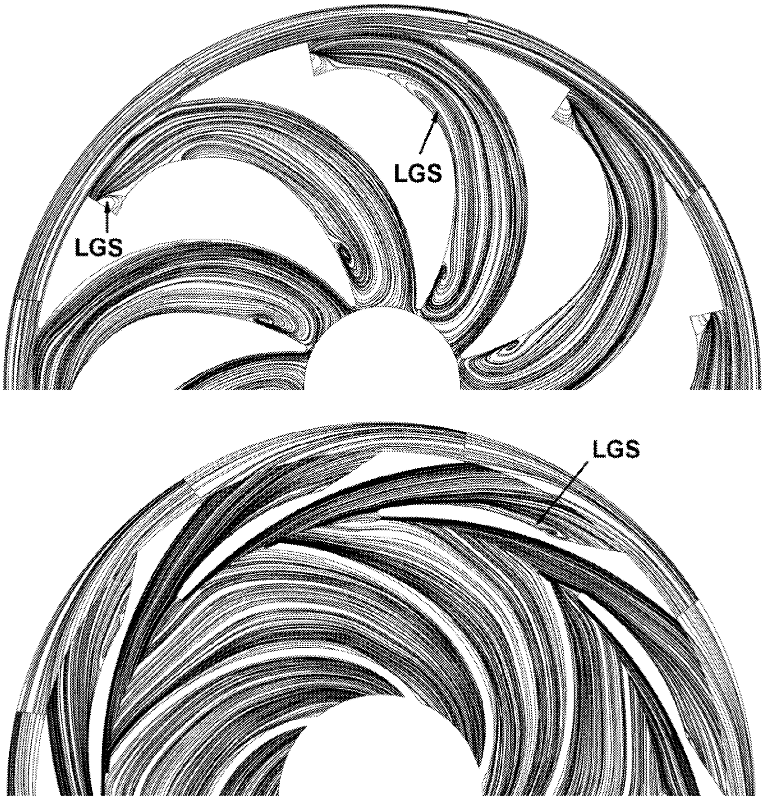

Fig.10: Streamlines on Plane1 and Plane2; pump mode, $Q = 11.4\,l/s$; LGS means a large global separation

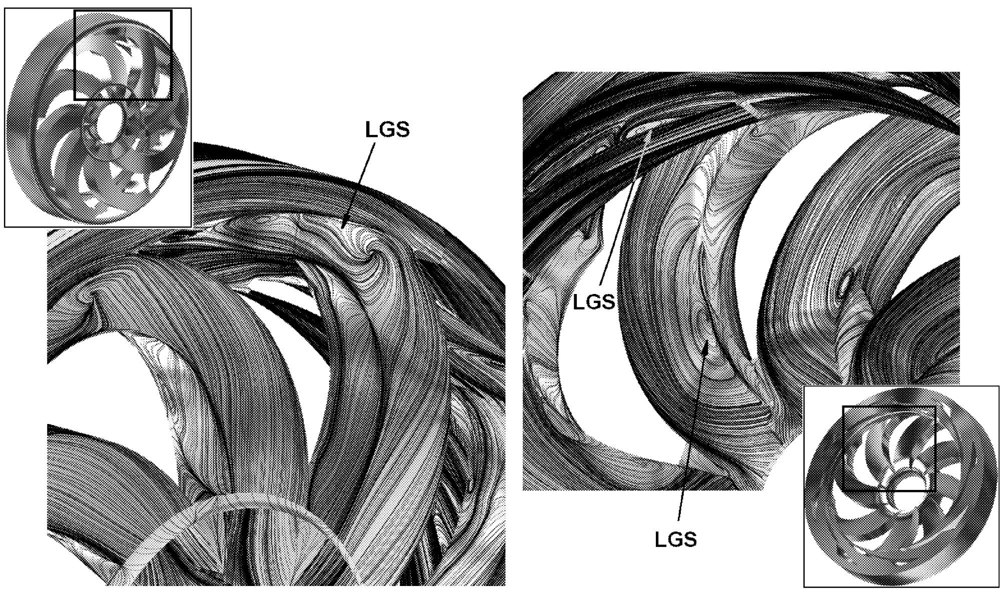

Fig.11: Separations in the stator; pump mode, $Q = 11.4\,l/s$; LGS means a large global separation

Figures 6–7 show the flow pattern for the stage with the full scale impeller in the reverse mode and the flow rate $Q = 15.75\,\mathrm{l/s}$. In Figure 6 the streamlines of the relative velocity can be seen on Plane1 and Plane2. No separation can be found in the impeller passages. The situation is a little bit more complicated in the case of flow inside the stator of the channel type. For all flow rates including the optimal one we can find the local separations in the stator. They are linked to a very complicated pattern of secondary flows. Often we can also observe the global separations in the stator which produce high energy losses. All these separations are difficult to visualize; the most direct way is using the surface streamlines. The global separations are characterised by the presence of the saddle point in combination with a nodal point of attachment, accompanied often with foci forming horn-type dividing surfaces [9]. Usually we can identify the reverse flow regions corresponding to the global separations. Figure 7 shows details of the stator surface streamlines and relatively small global separations in the highly curved parts of the stator channel.

Figures 8-9 show the flow pattern for the stage with the reduced impeller in the reverse mode and the flow rate $Q = 13.5\,\mathrm{l/s}$. The character of flow in the stator is very similar to the flow in the stage with the full scale impeller (Figure 7). As expected, the main differeces between the flow pattern in the stage with the full scale impeller and the flow in the stage with the reduced one can be found inside impeller passages. Here we can see large global separations which are probably the reason of the lower hydraulic efficiency of this stage. In Figure 9 details of the rotor surface streamlines can be seen as well as the visualization of the reverse flow regions in the impeller passages.

The flow in the stage operated in the pump regime and at the flow rate of 11.4 l/s is shown in Figures 10–11. Though there is no separation inside the impeller passages, we can find large local as well as global separations in the stator parts. These separations are typical for the pump regime and it is very difficult to suppress them.

4. Conclusions

This study indicates that the hydraulic efficiency of the multistage pump in the reverse mode can be quite high, even without any expensive corrections of the manufactured parts. The flow inside the analysed pump with the full scale impeller in the reverse mode is practically without large separations for the flow regimes close to the optimal rate of flow (which is in contrary to the pump regime). When operating at off-design conditions, critical is the highly curved part of the stator channel as well as the part where the return guide vanes interact with the rotor blades. The cut-off trailing edge of the impeller blades can be also another source of losses. The hydraulic design with the reduced impeller diameter shows, that we can shift the optimal rate of flow slightly to the left but the hydraulic efficiency will decrease with this modification.

Acknowledgments

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