CFD SIMULATION OF THE VELOCITY AND PRESSURE FIELDS WITHIN TWO-STAGE PUMP INTERIOR

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CFD simulation is aimed on the two-stage pump operation modeling in the pump and turbine regime. The flow analysis will be focused on the proper angle of wicket gates attack and on the separation localization. This investigation should bring the geometry modifications that will increase the hydraulic efficiency. The computational model was created in preprocessor GAMBIT and software Fluent will be used for the finite computation.

Keywords: *two-stage pump, turbine, angle of attack, modeling*

1. Introduction

The study of two-stage pump in turbine regime continues in this paper. The excessive head of gravity water systems is often wasted without use. That is the reason of the searching of improvement possibilities. Multi-stage pump can work with good efficiency and have a relatively low acquisition cost in comparison with the classic water turbine.

Before the CFD simulation there were finished the measurements and created the performance characteristics, see [1]. Evaluating the influence of the wicket gates will be the most important result of the flow analysis.

2. Geometry

The centrifugal two-stage pump is a standard product Sigma Group a.s, Fig. 1.

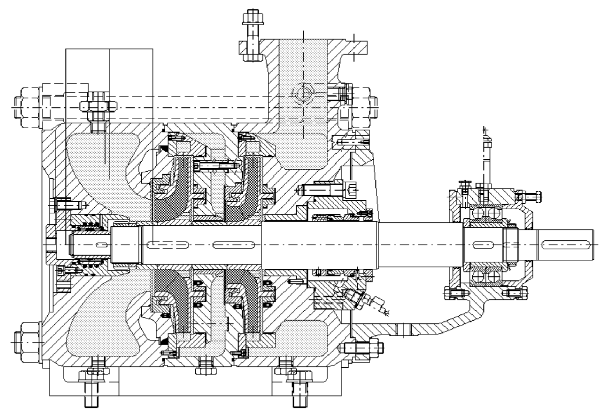

Fig.1

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The shrouded impellers consist of 6 and 7 blades (1st and 2nd stage) with the outside diameters $D_2 = 255\,\mathrm{mm}$. A computing model is complete to reach an optimal design (Fig. 1) and results. This is the only way of the fluid flow prediction trough the complicated geometry. An unstructured mesh consisted of around 8.8 millions predominantly hexahedral cells and was building in preprocessor GAMBIT r. 2.2.30. The main area of the flow analysis is the space between the distributor and the return channel. In this area there are a lot of interface possibilities with the pump interior, see Fig. 2.

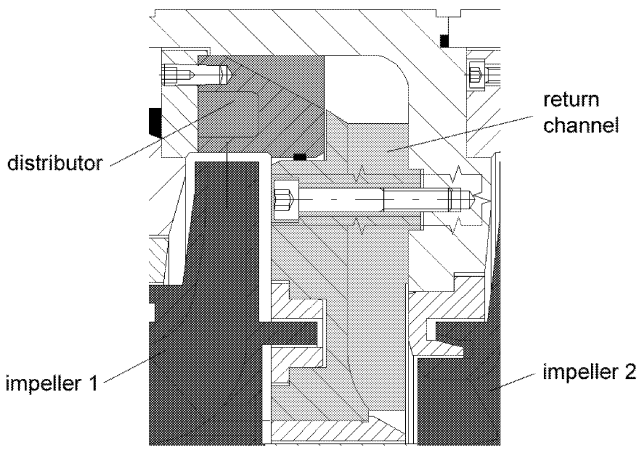

Fig.2

3. Basic characteristics

The most important operational parameter of the multistage pump were evaluated together with the flow features. Comparisons with experiment [1] were carried out, whenever it was possible. Specific energy curve is presented first, see Fig. 3.

The measurement revolutions were $n = 1450\,\mathrm{min}^{-1}$ as its visible on the right part of Fig. 3. The optimum working point were set for the flow $Q = 16.02\,\mathrm{l\,s}^{-1}$ and the specific energy $Y = 340.6\,\mathrm{J\,kg}^{-1}$. By using the mathematical simulation were gained for the optimal flow the specific energy $Y = 362.8\,\mathrm{J\,kg}^{-1}$.

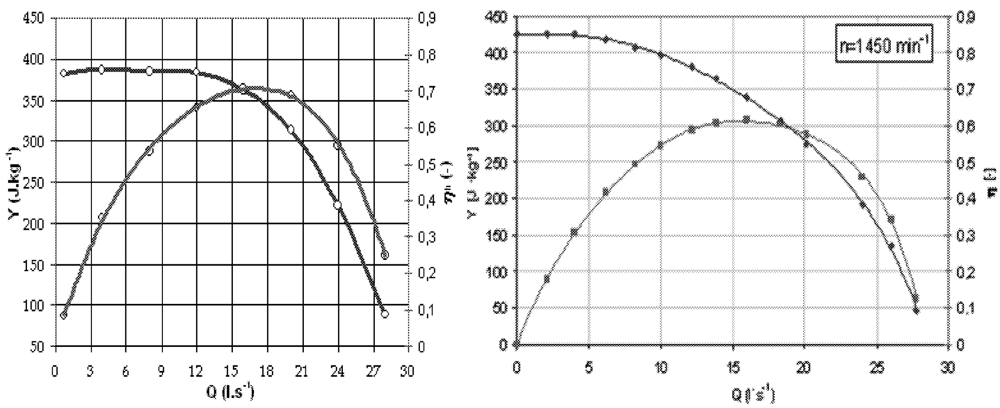

Fig.3

The biggest difference between the numerical simulation and the experimental results is visible especially for the lower flow values, where the computed specific energy is stagnating but then the measured specific energy is increasing.

For completeness' sake Fig. 3 shows the efficiency dependence, which was on the measurement $\eta = 61.4\,\%$ and computed hydraulic efficiency as $\eta_\mathrm{h} = 70.9\,\%$.

The last pump regime characteristic that shows the comparison of the measured and computed results is the power input behavior as it is visible in the Fig. 4.

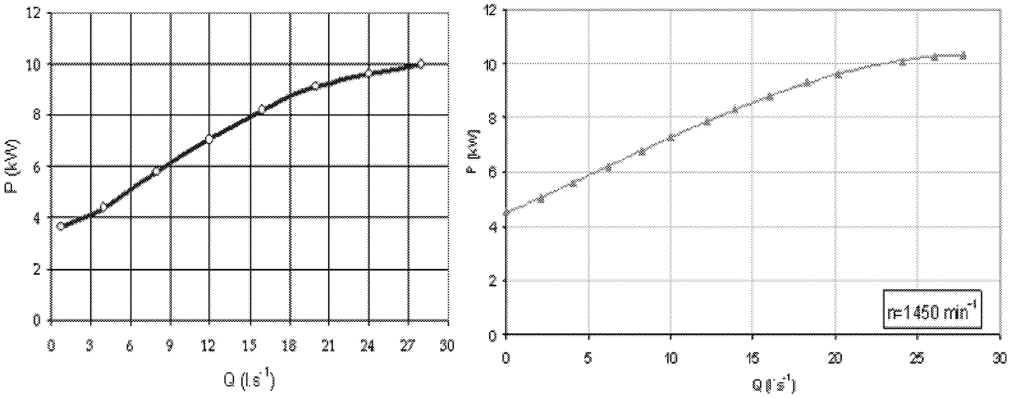

Fig.4

Experimental result : $P_\mathrm{opt} = 8.8\,\mathrm{kW}$. Computed result : $P_\mathrm{opt} = 8.2\,\mathrm{kW}$.

There are some differences if we compare just these two values, but the curves trend from the Fig. 4 shows some similar character.

The power input increasing is approximately linear but in the higher flow area its increase is falling. Now let's look on the reverse and preface the normal net head and flow dependence for the turbine regime, see Fig. 5.

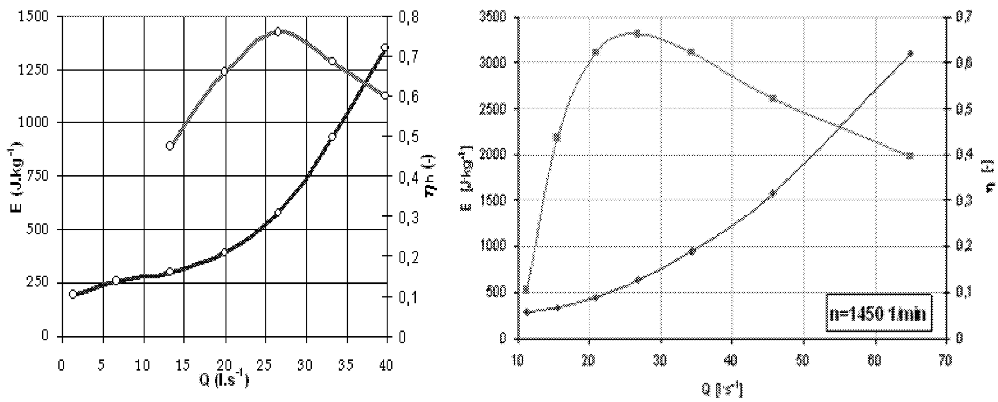

Fig.5

For the weak convergence there are missing data of the higher flows in the left part of the Fig. 5. Out of the measurement goes the optimal working point for the flow $Q = 26.6\,\mathrm{l\,s^{-1}}$ and specific energy $E = 632\,\mathrm{J\,kg^{-1}}$. The efficiency is : $\eta = 66.8\,\%$. For the same flow value there is the specific energy $E = 576\,\mathrm{J\,kg^{-1}}$ and hydraulic efficiency $\eta_\mathrm{h} = 75.9\,\%$ from the mathematical simulation – see Fig. 5.

4. The flow analysis

The flow rate analysis between the distributor and return channel goes to the interior changes of the two-stage pump (Fig. 2). Changes in the runners are not assumed, so we focused on the area, where the flow is curved a lot and can be the place of the higher losses. The informative preview on the rotating parts with the guide channels and the complete model is shown on Fig. 6.

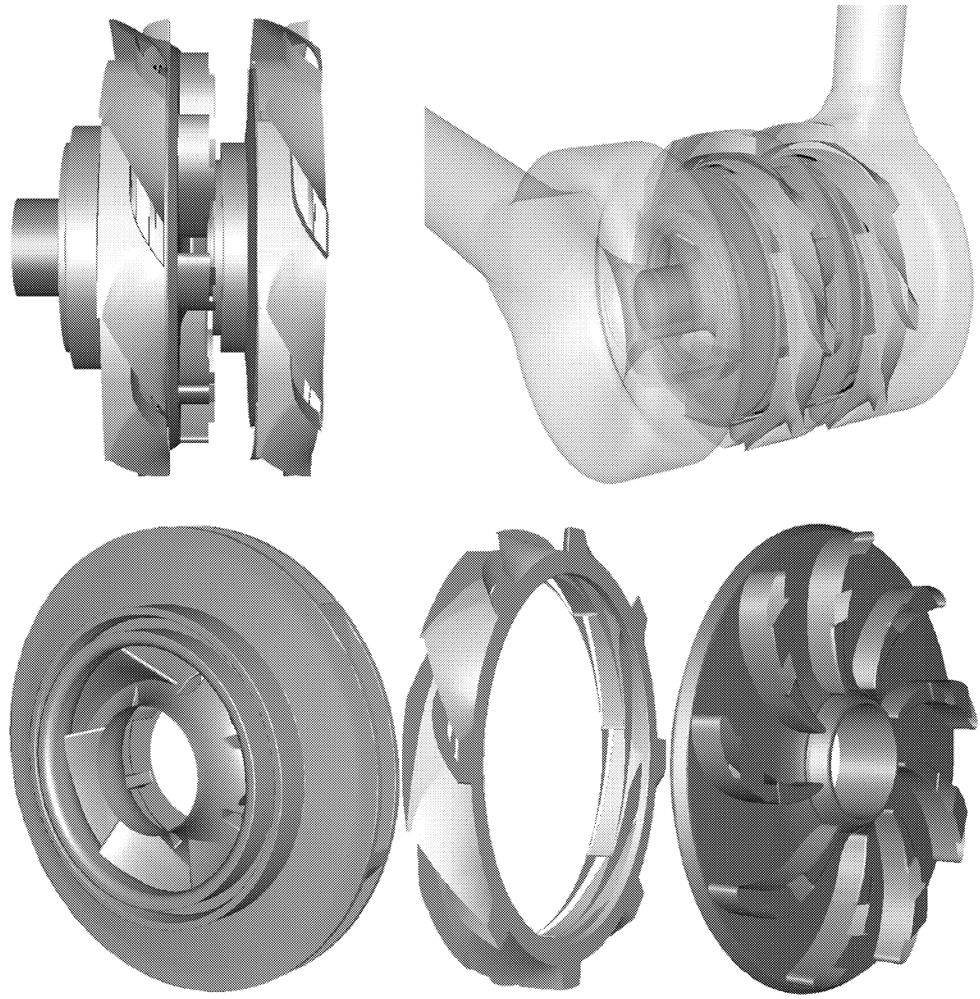

Fig.6

Through the shaft is guided the z axis. The sections where the flow through the pump will be monitored lies within the return channel plane (for $z = \mathrm{const.}$), next on the cylindrical plane passing through the distributor and through the plane of the z axis itself.

The most interesting will be the turbine regime for the flow $Q_\mathrm{opt} = 26.6\,\mathrm{l\,s^{-1}}$, additionally will be shown the rate display in the pump regime ($Q = Q_\mathrm{opt} = 16.02\,\mathrm{l\,s^{-1}}$).

4.1. The turbine regime

The absolute velocities field in the return channel is visible in the Fig. 7 close to the pump body wall ($z = -50$ mm) and Fig. 8 close to the return channel disc ($z = -40$ mm).

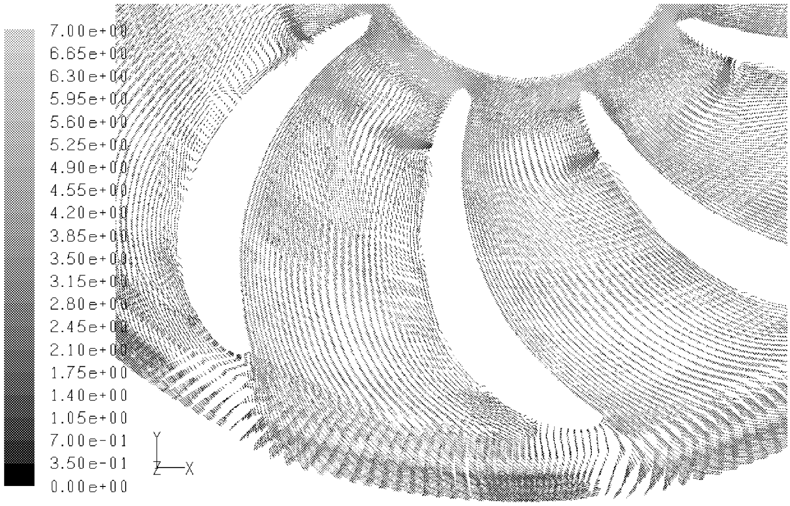

Fig.7: ($z = -50$ mm)

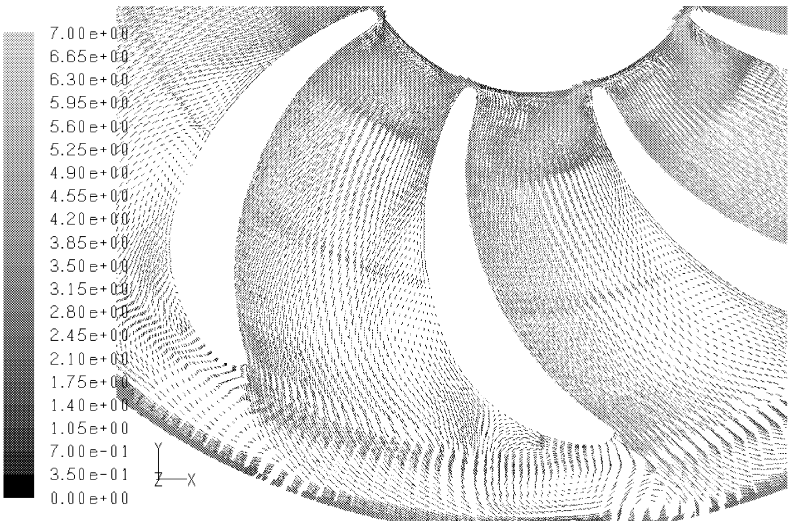

Fig.8: ($z = -40$ mm)

It is lovely visible, that the flow is markedly random and the main flow part is realized close to the pump body wall (Fig. 7). The blades are flow around one of the sides and on the trailing edge (from the turbine point of view) became vortex. Further will be shown the flow field in the entering and trailing edge areas for the sections $z = -40$ and 50 mm, Fig. 9 and Fig. 10.

The situation on the runner enter is rather correct and the stagnating point lies almost in the middle of the entering edge; applies to all blades. It was mentioned above that there

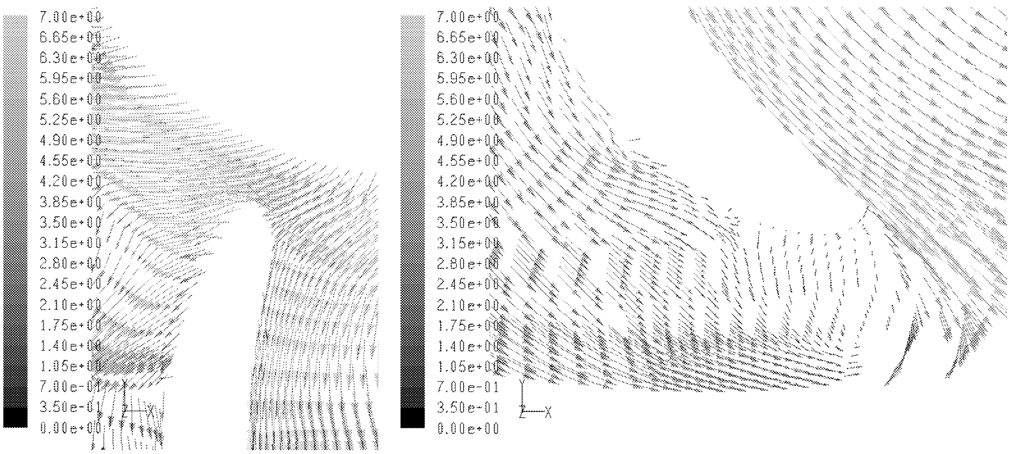

Fig.9: (z = −50 mm)

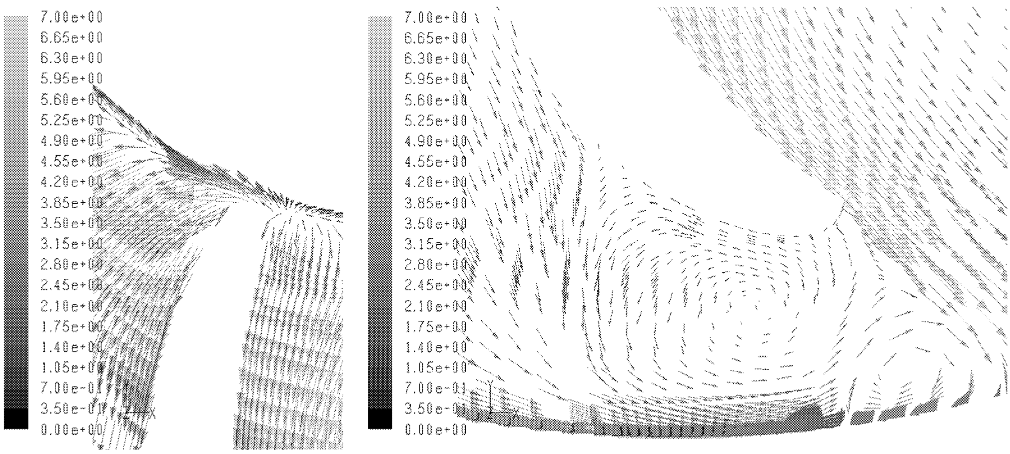

Fig.10: (z = −40 mm)

is a vortex on the trailing edge of the blade that is more significant in the short distance from the return channel disc. The explanation offers the Fig. 11, plane $x = 0$.

Fig. 11 shows the liquid flow through the return channel to the first runner distributor. Even like the above mentioned cases are the absolute velocities observed. Expressively appears the liquid flow round the wall of the pump body, where the velocities are essentially higher than close to the wall of the return channel disc. On the eve of the entering edge to the distributor there creates the strong vortex. Similar phenomenon was signed for the right and left terminal of the specific energy characteristic.

As the last one it will be shown the input to the guide channel on the cylindrical plane segment, Fig. 12.

The trailing edges of the guide channel blades are not made for the turbine regime entering. The liquid reaches the wide trailing edges of the pump blades, divides and brake away. The channel filling is on the other hand proper and satisfactory.

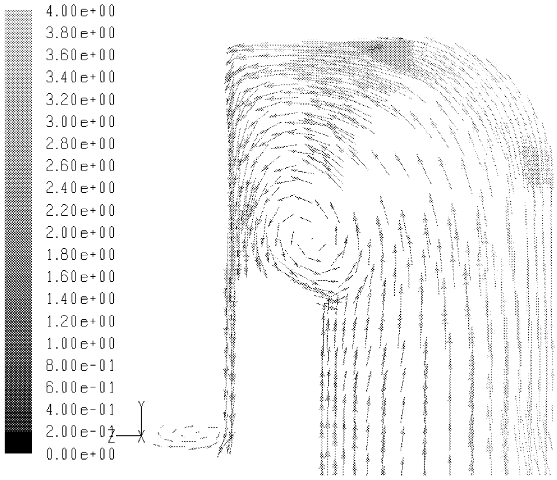

Fig.11

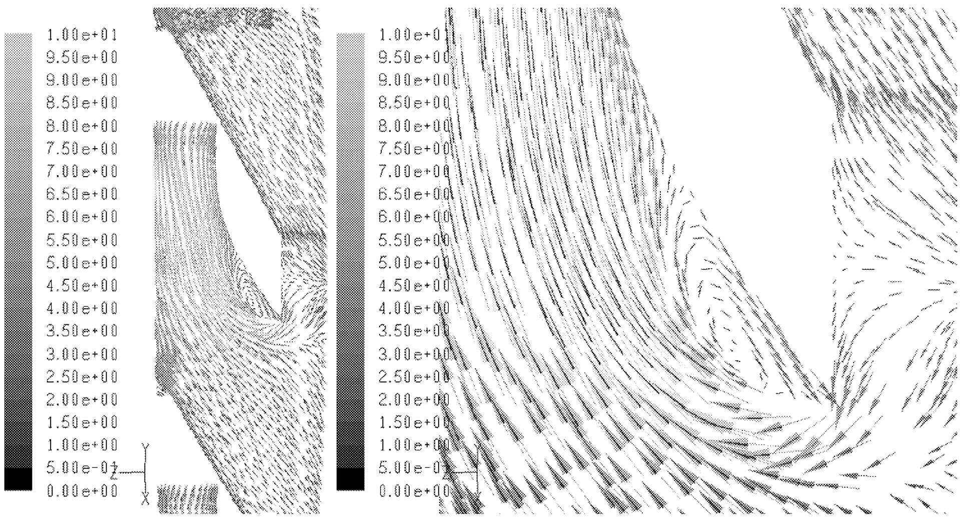

Fig.12

4.2. The pump regime

The analysis is not as complete as the turbine regime case. It is focused just on the main flow characteristics from the considered interior changes point of view $Q = Q_{\mathrm{opt}} = 16.021\,\mathrm{s}^{-1}$.

Fig.13 represents the return channel section for $z = -50\,\mathrm{mm}$. At first sight catches the blades entering edge that causes dramatic flow separation and vortices practically in the whole width of the channel. Beside the fact that it is the optimum working point the same picture is through the whole pump characteristic range.

Against the turbine regime the plane $z = -40\,\mathrm{mm}$ is omitted and the velocity field is draw in plane $x = 0$, Fig. 14.

The liquid gets of the guide channel and stream down the sloping edge of the return channel. In the moment it ends the 90° bending has almost the same velocity in the whole

Fig.13

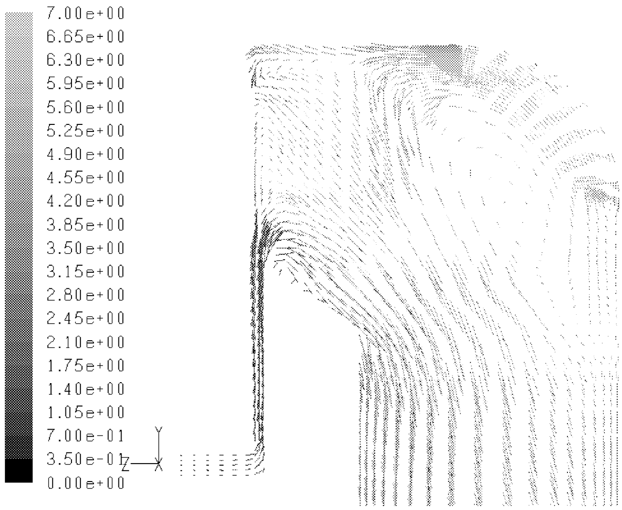

Fig.14

width of the clear opening. Before it happens the suddenly channel above the return channel caused the flow separation and creation of the large vortex.

Further it will be shown the velocity field on the cylindrical plane segment, Fig. 15.

The diffusive character of the distributor channels is the main problem in the pump regime. Accompanying problem is the velocity decrease in the middle cross-section part. If the clear opening were contracted it will mean big troubles in the turbine regime. The turbine optimal working point is set in the higher flow ratio area and the contracted channels bring the wider guide vanes, see Fig. 12. That's why it's better not to interfere into the distributor.

5. Conclusions

At the very beginning has been said that the two-stage pump modification will not include the runners' replacement. The distributor will be connected to them by way that implies

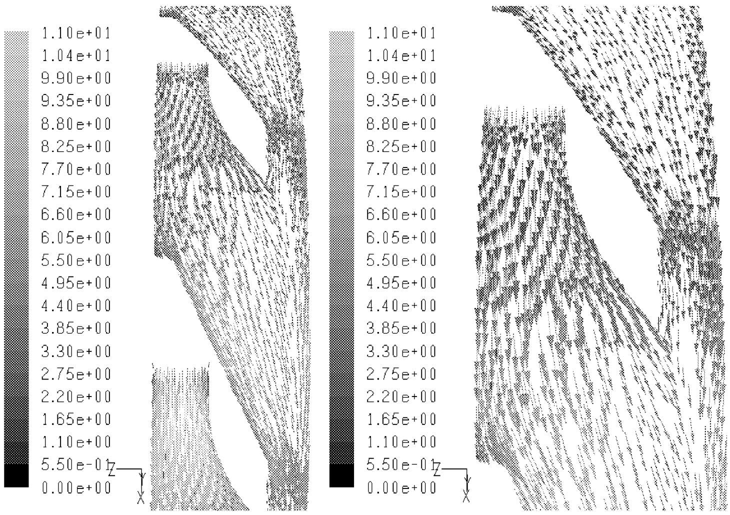

Fig.15

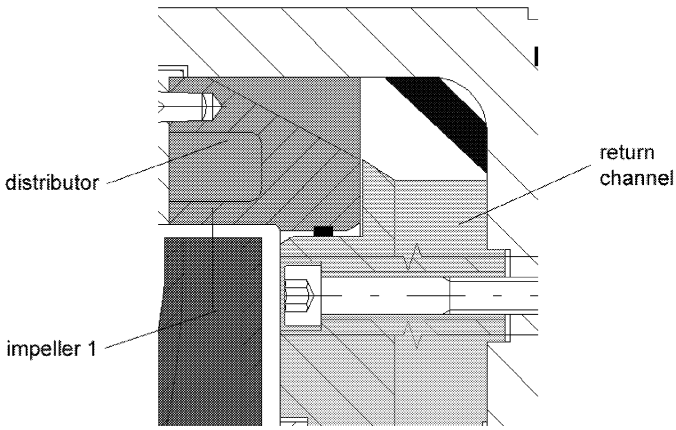

Fig.16

from the flow analysis, see Figs. 12 and 15. For the turbine as well as the pump regime is the velocity field in principle acceptable.

Changes will touch the return channel (Fig. 7–10, Fig. 13) and the chest between the guide and return channel (Fig. 11, Fig. 14).First it will be better to change the chest – it is easier and less exacting. The goal is the vortex originating restriction and better stock inlet between the runners creation. How it will look like is outlined on Fig. 16.

The ring fixed by screws reduces the chest size and eliminate the clear opening area diffusion, which causes the flow separation. This happened in the turbine as well as in the pump regime.

The second modification should repair the return channel blading. It depends on the fact if the pump will work especially in the turbine regime. Then it is necessary to entertain such interference. It can be said the return channel flow in the pump regime is inferior.

The above mentioned modifications and their suitability will be the next scope of employment.

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References

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