Numerical optimization of a centrifugal pump impeller with splitter blades running in reverse mode

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Abstract – *In this paper the numerical modeling and optimization of a centrifugal pump impeller running in reverse operation is presented. In order to determine the operating envelope of the turbine as well as validate the numerical results, simulations were initially performed at normal pump operation. By comparing the numerical with the experimental values, a relatively good correlation could be achieved. The turbine operating point was determined using empirical correlations, which were also validated by the numerical performance curves obtained from multiple operating point simulations. This information is used to validate the numerical approach and assumptions in order to be applied for the optimization of the impeller in turbine operation. Subsequently, a parametric study is performed comparing the relative effect in performance of various design modifications, such as the inlet edge shape, the meridional channel width, the number of vanes, and the effect of splitter blades. Using the previous results as guidelines, the numerical optimization of the runner is performed using two optimization algorithms. In the first, only the blade shape was allowed to change while the rotational speed was also included while meridional contour of the runner was maintained constant in order to accommodate the available spiral casing dimensions. In the second optimization, the runner diameter was also included. From the results of the optimization, an overall increase in turbine efficiency of approximately 2.7% could be achieved.*

***Keywords**: Hydropower, Pump as Turbine, Impeller Design, Numerical Modeling, Optimization*

I. Introduction

The continuous increase of cost of conventional fuels has made the development of small and mini hydroelectric plants financially viable. However, the specific cost per kW of such units can still be relatively large. A popular alternative, for the limitation of this cost has been the use of centrifugal pumps as turbines (PAT) [1], [2]. The advantage of such an approach is the simple construction, the ability to use standard of the shelve components with minimal modifications and the benefit from decentralized electricity generation [1]-[4]. These characteristics have increased their popularity over the years, especially when used in mini hydro electric schemes in rural and developing regions, in which the success of a project strongly relies on the initial costs [3].

Although, substantial benefits are associated with the use of pumps as turbines, there are several important disadvantages that shouldn't be ignored. Most importantly, the efficiency of such machines is significantly lower compared to the equivalent in pump operation, which may amount up to -8.5% of pump efficiency [1]. Another issue with using PAT is the difficulty to predict the actual performance of a standard pump in turbine mode. Several efforts have been made in the past using empirical correlations [1], CFD or experimental measurements [4]-[7], but still not enough information is available. This lack of information may discourage the use of pumps as turbines for a cost effective energy generation solution.

A significant disadvantage of PAT is the inexistence of guide vanes and of a flow control mechanism, which limits the operating envelope and inhibits the ability to efficiently control the operating point. This highlights the importance of improving their design in order to maximize energy productivity over an extended operating range. One way to achieve this is by increasing the hydraulic efficiency of the runner at a desired operating point. A recent approach that has been utilized in high head Francis turbines and centrifugal pumps has been the incorporation of splitter blades in the runner. In various studies, improvements on the head and efficiency curves have been reported [8]-[10], while an improvement of the unsteady characteristics has

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also been shown to be achieved [10], [11]. The cavitation performance of the impeller has also been reported in [12]. The use of splitter blades has been shown to favorably modify the pressure distribution on the blades and at the same time delay the flow separation in certain operating conditions, which could facilitate the inception of cavitation [13]. In pump operation the increase of blade number, reduces the blade loading and improves cavitation characteristics, while also leads to an efficiency increase as the secondary flows within the flow channel are reduced. In high head machines, in which the impeller widths are small, compared to the diameter, the increase of blade number may lead to an overall reduction in efficiency due to an increase of hydraulic friction losses. However, in high head machines the relative velocities are larger, while additional losses can be induced due to flow chocking. The use of splitter blades allows the above to be achieved with a simultaneous limitation of these unwanted characteristics. A second approach to improve the performance of centrifugal pumps in turbine operation is by using an appropriate optimization procedure [14]-[16] aiming at the maximization of the hydraulic efficiency of the individual components of the PAT, and especially the runner.

This paper deals with the performance improvement and optimization of a pump impeller, operated as a turbine. A standard of-the-shelf low specific speed pump was provided by a local manufacturer and studied numerically in order to identify potential areas of improvement and subsequently, to optimize the performance characteristics of the pump in turbine operation. The aim was to develop an efficient and, at the same time, cost effective solution for low output applications, such as energy recovery etc. Initially, the pump performance characteristics were obtained numerically and compared against the manufacturers' performance data in order to identify the flow behavior within the impeller channel at various operating conditions, to identify potentially problematic areas of the initial design and to validate the numerical approach. Subsequently, the initial design was operated in reverse and simulations were repeated for various operating conditions. The effect of various design alterations were investigated parametrically, and finally the design optimization of the initial design was performed. Two optimization approaches were used, in which the parameters defining the geometry were allowed to vary, while in the second, the rotational velocity was also used as a free parameter. In all optimization cycles a constant meridional profile was considered, which was a constraint imposed by the geometry of the available spiral casing. Through this process it was possible to identify the geometric parameters leading to improved performances and increase the hydraylic efficieny of the PAT by 2.7%.

II. Model Description and Design Variations

The numerical model used in this paper is based on a low specific speed centrifugal pump with a design head of 38mWc, flow rate 18m^3/h, a rotational speed of 2900 rpm and 5 impeller blades. In the following table, the main geometric parameters of the pump are summarized.

TABLE I
Geometric characteristics of centrifugal pump under investigation

Parameter	Value
Pump Type	LDP 32-160
Rotational Speed [rpm]	2900
Impeller width [mm]	5
Impeller diameter [mm]	177
Suction diameter[mm]	53.1
Discharge diameter [mm]	41.1
Motor power [kW]	5.5

As mentioned above, in order to identify performance improvements through simple modifications of the existing geometry, five additional design variations, as listed in table 1, were considered. Specifically, the effects of the inlet edge blade profile rounding, the increase of blade number, the increase of the meridional flow channel width and the use of splitter blades on turbine performance were investigated.

TABLE II
Design variations under consideration

Case#	Design variation
0	Initial Geometry
1	Inlet edge blade rounding
2	20% width increase
3	6 blades
4	7 blades
5	Splitter blades

III. Blade Parameterization

In order to effectively perform a numerical design optimization procedure, a suitable means of parametrically describing the geometry under investigation, is necessary. The chosen parameterization must allow for an accurate description of the runner, with the minimum number of parameters. In this application the conformal mapping method was used for the parametric description of the blades. By assuming a fixed meridional channel contour (Fig. 1), the mean blade surface was obtained by using two 5-control point bezier curves that define the blade angle distribution along the hub and shroud profiles, respectively.

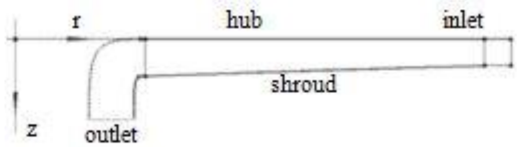

Fig. 1 Meridional channel of runner under investigation

The angle distributions are defined as functions of the normalized meridional length m/r, as shown in fig. 2. The final pressure and suction surfaces were obtained after imposing a thickness distribution on the mean surface of the blade. For the thickness distribution, a five control point bezier curve was used fig. 3. As the blade is radial and the runner width small, one defining curve is enough to fully describe the blade shape. Each control point was allowed to vary within a suitable range to allow feasible design variations to be obtained. The control points used to describe the blades are presented in table 3.

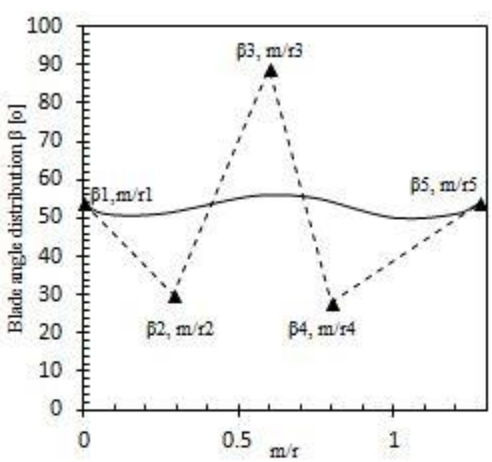

Fig. 2 Blade angle distribution along the normalized meridional length

In the optimization study, the blade number is maintained constant, equal to the initial one (5 blades), while the splitter blades are positioned at the intermediate pitch position between the main blades (Fig. 4). The design of the splitter blades follows the main blades and a constant length ratio of 0.75 is chosen.

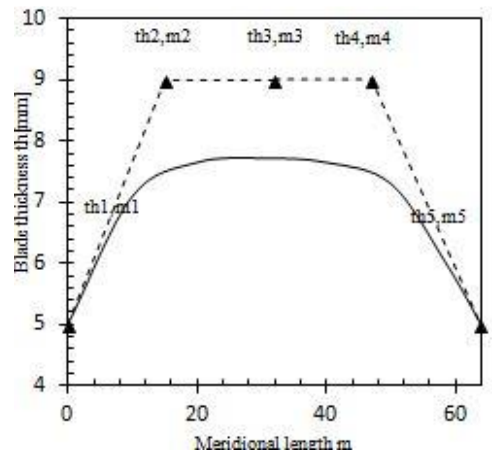

Fig. 3 Thickness distribution along the meridional length

TABLE III
Design parameters of initial geometry

Parameter	1	2	3	4	5
β	54	30	89.6	28	54
m/r	0	0.29	0.6	0.8	1.28
th	5	9	9	9	5
m	0	17	32	47	83.93

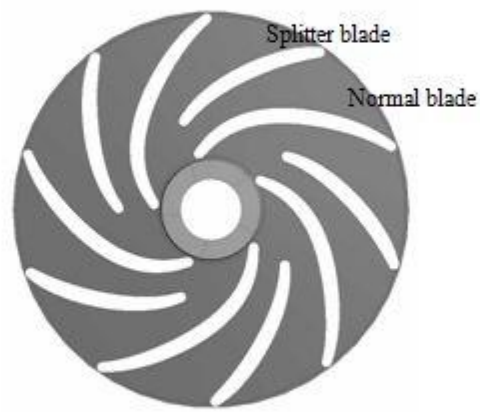

Fig. 4 Runner with splitter blades

IV. Numerical Method

The computational domain under consideration consists of a single blade-to-blade passage (Fig. 5), as well as of the full centrifugal pump model, including the impeller and the spiral casing (Fig. 6). For both cases, an unstructured tetrahedral mesh was used, while for the flow analysis the steady state RANS equations using a 2nd order discretization scheme and the realizable k-e turbulence model were solved. The boundary conditions were derived by using a spiral casing flow simulation where the absolute and tangential velocity components were obtained and from suitable empirical correlations that relate the design flow in turbine mode to the operational characteristics in pump mode [eq.1], as presented in [1]. The corresponding boundary conditions were: inlet flow rate of $29m^3/h$, zero outlet static pressure, and rotational speed of 2900 rpm. The commercial CFD software Ansys fluent was used for all simulations.

$$Q_T = Q_P \cdot 1{,}2 \cdot (\eta_P)^{-0{,}55} \tag{1}$$

An important technical detail of this specific pump that is included in the numerical model is the abrupt contraction of width from 12 mm to 5 mm at the inlet of the blade to blade domain, to account for the transition between the spiral casing and the runner geometries (figs. 5, 7-top).

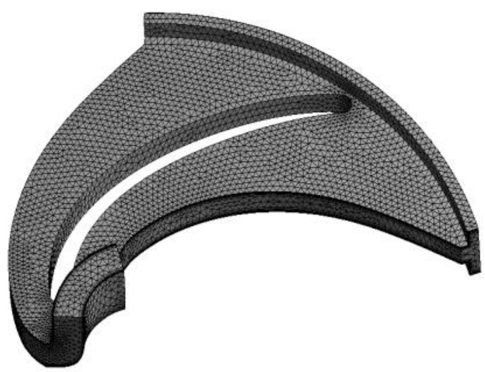

Fig. 5 Blade to blade domain and mesh used

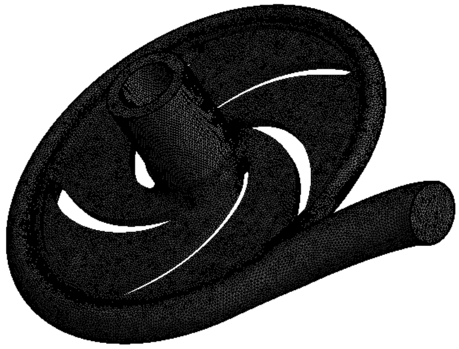

Fig. 6 Numerical mesh of full model

Furthermore, local refinement (fig.7-bottom) close to all walls was applied in order to accurately capture the boundary layer and ensure the y+ constraints are met [17]. For the evaluation of the numerical approach and the determination of mesh independence, three different mesh sizes were considered, i.e. 125k, 200k and 497k cells were considered. From the results, a mesh of approximately 200k tetrahedral cells was considered as an acceptable compromise between accuracy and computational cost. The same mesh configuration was also used in the optimization.

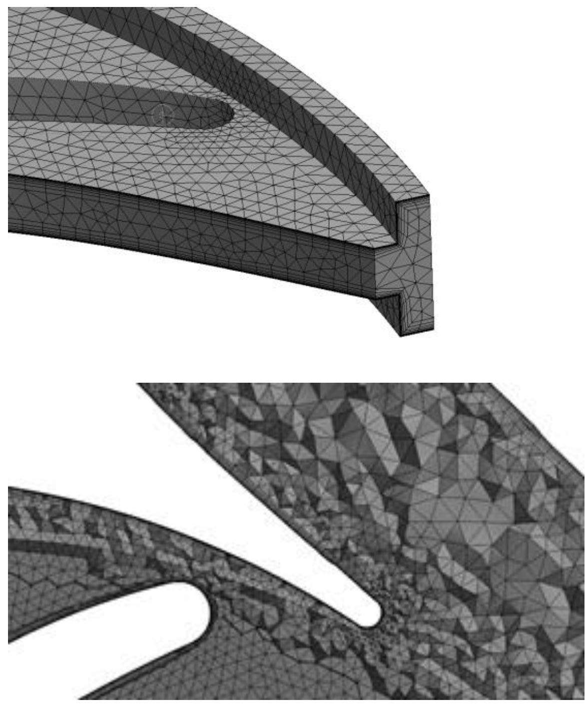

Fig. 7 Mesh details: top: Expansion at inlet; bottom: Mesh refinement close to spiral casing tongue

V. Runner Optimization

The numerical optimization of the runner was performed with the use of the stochastic optimization software EASY based on genetic algorithms, developed by LTT-NTUA [18]. A second optimization was performed using the optimization routines available in Ansys Design Explorer package. The difference between the two, apart from the software used, was also the number of design parameters. In the first optimization apart from parameters defining the angle distribution of the blade, the runner diameter was also taken as free variable. The variation range of each design variable was selected around its initial value. Specifically, the range for outlet and inlet blade angles, β_1 and β_2, were set 25º – 35º and 30º – 40º respectively and outlet and inlet radius, R_1 and R_2, at 24,5 mm – 30 mm and 80 mm – 86,4 mm, respectively. The intermediate control points of the hydrofoil mean line ranged between 1/3 and 2/3 of their allowed length, in order to locally avoid high curvature. Finally the wrap angle was allowed to vary between 30º and 45º, which is typical for centrifugal pumps. Fig. 8 summarizes the optimization procedure using EASY.

A similar approach is implemented by the Ansys optimization package [19]. Initially, a design database is formed by using design variations consisted of different parameter combinations across their respective range. Each of these individual designs is evaluated numerically and the performance parameters of each variant are stored in the database. Using these evaluations, a surface distribution is calculated that interpolates the design parameters with the cost functions. By increasing the population of the initial database, the interpolated surface can better capture the sensitivity of the objective functions to variations in the design parameters. Subsequently, the optimum in the interpolation surface is located. By using a genetic algorithm approach the model relating the design parameters can be refined until convergence is achieved.

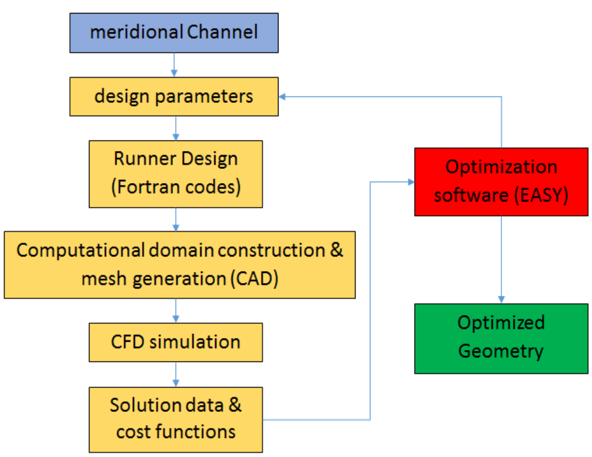

Fig. 8 Runner optimization procedure using EASY

Energy efficiency was chosen as primary evaluation criterion, representing the real project efficiency and including kinetic energy losses at runner outlet. Energy efficiency is computed from the torque M produced at axis of rotation and net head across the turbine according to (eq.1) in which Pt corresponds to the total pressure at the inlet and outlet of the domain, respectively. As a secondary evaluation criterion the percentage of total head absolute deviation from the respective value computed for turbine operation of the initial impeller, was set. Cost functions CF_1 and CF_2 for feeding the optimization algorithm were set accordingly. Finally, the optimization procedure was performed for 1000 iterations of evaluated runner geometries, separated in 10 consecutive generations.

$$\eta_h = \frac{M \cdot \omega}{(P_{t,in} - P_{t,out}) \cdot Q} \quad (2)$$

$$CF_1 = 1 - \eta_h \quad (3)$$

$$CF_2 = \frac{|H - H_{T,init}|}{H_{T,init}} \quad (4)$$

VI. Results

VI.1. Design variation results

In the following figures the performance curves for each case are shown, normalized by the theoretical operating point of the initial geometry ($Q=29\ m^3/h$). Fig. 9 shows the variation of predicted head against turbine flow, and as can be observed, cases 4 and 5 show similar performances, while for all other cases the same flow is achieved at lower heads. On the other hand, the increase in impeller width reduces the head across the flow range studied.

A comparison between the corresponding efficiency characteristic curves is shown in Fig.10. Comparing cases 0 and 1 it can be seen how the rounding of the inlet edge improves the overall turbine performance, especially at larger flow rates, though a small effect in the head characteristic appears (figs 9, 10). At higher flows, the flow velocities increase and the potential mismatch in the flow and velocity angles leads to increased losses. An increase in blade number leads to a reduction in hydraulic efficiency of the runner (cases 4 and 5), due to the increased friction losses. Furthermore, it is evident that an increase in impeller width (case 2) shifts the maximum efficiency point to higher flow rates, while a substantial increase of its value can be achieved. The effect on hydraulic performance of the leading edge rounding can also be seen, where approximately a 2% increase in efficiency can be seen compared to the initial shape. This increase is less evident for the BEP flow rate, but becomes significant at higher flows. Also, according to these curves, the increase in blade number appears to reduce substantially the hydraulic efficiency of the runner (cases 4 and 5) at the theoretical best efficiency flow rate.

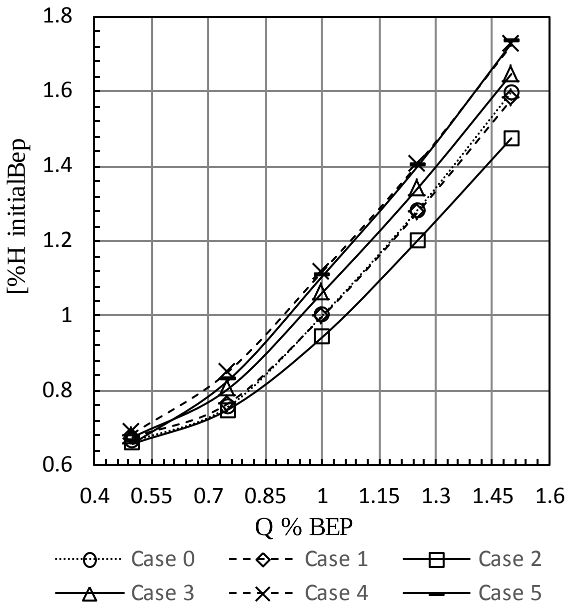

Fig. 9 Head variation at various flow rates for each design variation

To further evaluate the results shown above, cases 1, 2, 4 and 5 were further studied considering the full model, including the spiral casing, for the nominal flow rate in turbine mode.

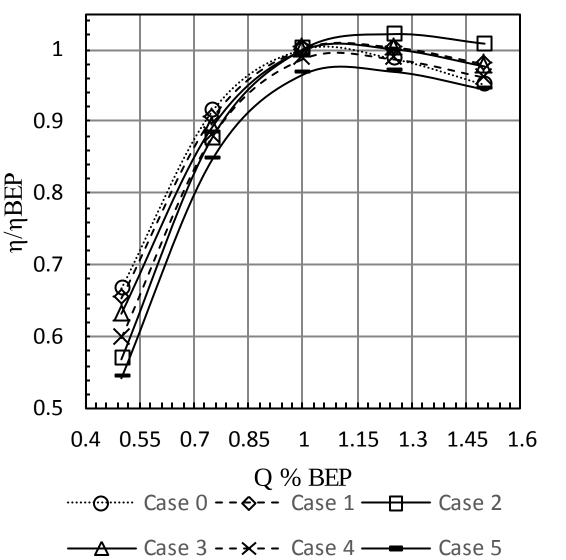

Fig. 10 Efficiency variation at various flow rates for each design variation

TABLE IV Relative Performance results of different design variations

Case #	Head[%]	Power[%]	Efficiency[%]
1	100	100	100
2	93.0	90.1	96.7
4	104.5	109.9	105.3
5	101.6	106.5	104.8

The obtained operational characteristics are compared in table 4. The use of more blades leads to a reduction of the specific speed of the pump/turbine due to the increase of the operating head (cases 4 and 5), while the opposite is observed when increasing the impeller width (case 2).

The results show that in cases 4 and 5 the overall efficiency is increased by 5.3 % and 4.8 %, respectively, compared to the reference case 1. This is an opposite trend than that of the impeller alone, and reveals that the use of more blades achieves a more effective and efficient energy transfer in the turbine, and reduced overall losses.

The increased actual losses, compared to the blade-to-blade simulation, can be seen in Fig.11, where the velocity vectors at the mid span of the turbine are plotted. Large vortex regions are predicted for the initial geometry, as a result of the small blade number and the increased blade pitch-wise distance. These can be further reduced by adjusting the blade angles, while further performance improvements can be achieved by reducing the blade thickness and thus minimizing the flow velocity and friction losses within the flow passages. By doing so, the cavitation characteristics of the impeller can also be improved.

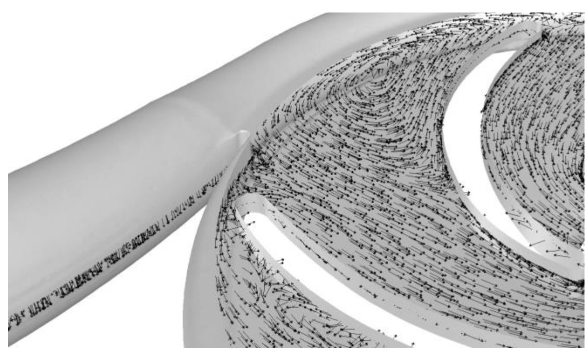

Fig. 11 Velocity vectors at the mid-span of the runner for case1

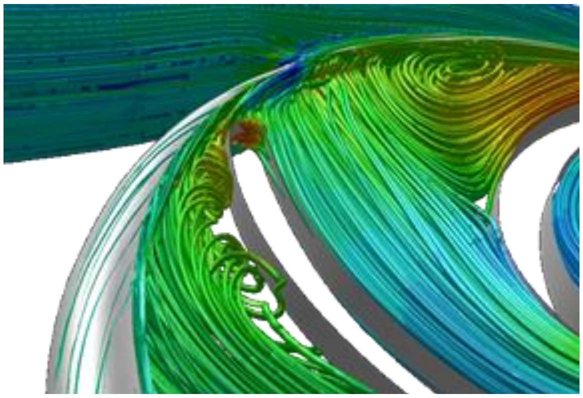

Fig. 12 Flow lines for case 1

In fig.12 the interaction effect of the last blade as it approaches the spiral tongue, can be observed. In the pressure side of the blade, a tip vortex is formed as a result of the local velocity gradient in that region. As the runner blade approaches the tongue, the area restriction causes a sudden localized increase in velocity, which in turn results in a modified flow angle relative to the blade leading angle and hence to higher impact losses.

Fig. 13 shows the effect of the splitter blades in the velocity field, as well as the corresponding flow stream lines. It is evident that the use of the splitter blades substantially reduces the size of blade- to-blade vortices, improves the poor runner-spiral tongue interaction effects, and inhibits flow separation near the blade exit.

Some other important features of the developed flow field are discussed below. Fig. 14 shows the relative velocity vectors at the mid-span of the runner, where a flow separation is observed at the suction side towards the trailing edge of the initial runner. A correction of this flow feature may be achieved by an increase in the blade outlet angle. In the same figure, the positive effect of the splitter blades on the flow field in the outlet edge region is shown, where the recirculation region appears to be reduced.

Fig.15 shows the flow field at a given cross section of the spiral casing. It is evident that severe secondary flows are formed, which are mainly caused due to the sudden contraction in the transition from the spiral casing towards the runner. This flow pattern causes the flow to choke, increasing thus considerably the energy losses, and at the same time leads to a distortion of the velocity profile at the inlet of the runner. This irregularity, seen by the circumferential averaging of the velocity vectors in the axial direction, intensifies the energy losses at the runner inlet and leads to the observed large discrepancy between blade-to-blade and full model simulation results.

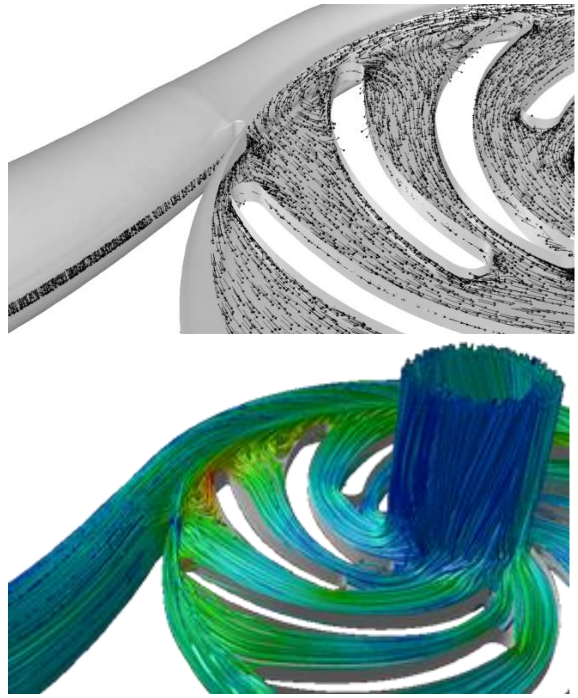

Fig. 13 Relative velocity vectors (top) and flow lines (bottom) for case 5

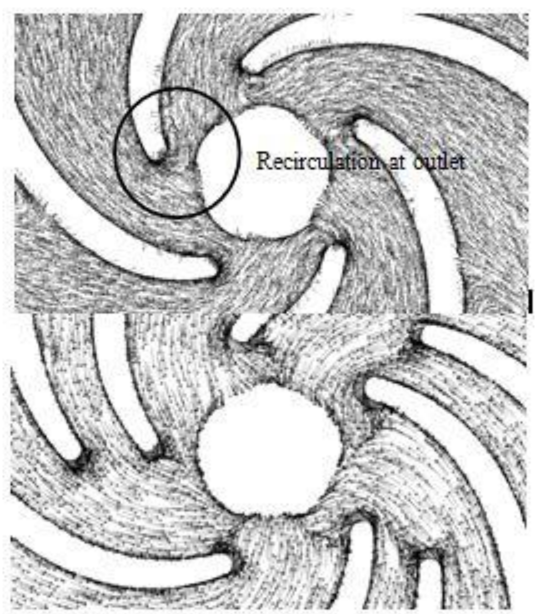

Fig. 14 top: Velocity vectors at mid-span of the initial runner showing flow separation at the suction side near the outlet edge of the blade; Bottom: Recirculation reduced with splitter blade introduction

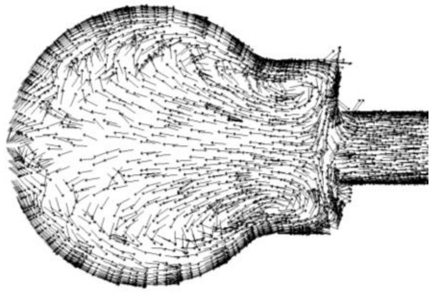

Fig. 15 Secondary flows at a cross section (y=0) of the spiral casing

Finally, in fig. 16 the corresponding relative velocity vectors and flow lines for the various cases studied are shown. In all cases the blade-to-blade fixed vortex can be observed, however, at a lesser extent with increasing blade number. In terms of flow uniformity, we can see that case 5 using the splitter blades produces the best results, and should be investigated further. The results presented above were used as design guidelines for the subsequent application of the optimization study.

VI.2. Optimization Results

The comparison of operating curves between the initial impeller and the optimized runner is performed for a flow rate range of 25% to 125% of the estimated best efficiency point for turbine operation. As previously mentioned, two separate optimization runs were performed using two optimization algorithms and different design variables. In the first, only the blade shape is modified and two optimization runs were performed using initially a fixed rotational speed, and subsequently including the rotational speed as a design parameter. In the second optimization the runner diameter was also considered as a design variable. In both cases, the hydraulic efficiency at the BEP and the design head were used as the objective functions. In table 5 the optimization results are summarized and compared with the initial data, where the optimization cycle with a fixed rotational speed is denoted as O1, including the rotational speed as a design variable as O1b, while the second optimization is denoted as O2. At the same time, in fig.17 the normalized performance curves are compared for each case.

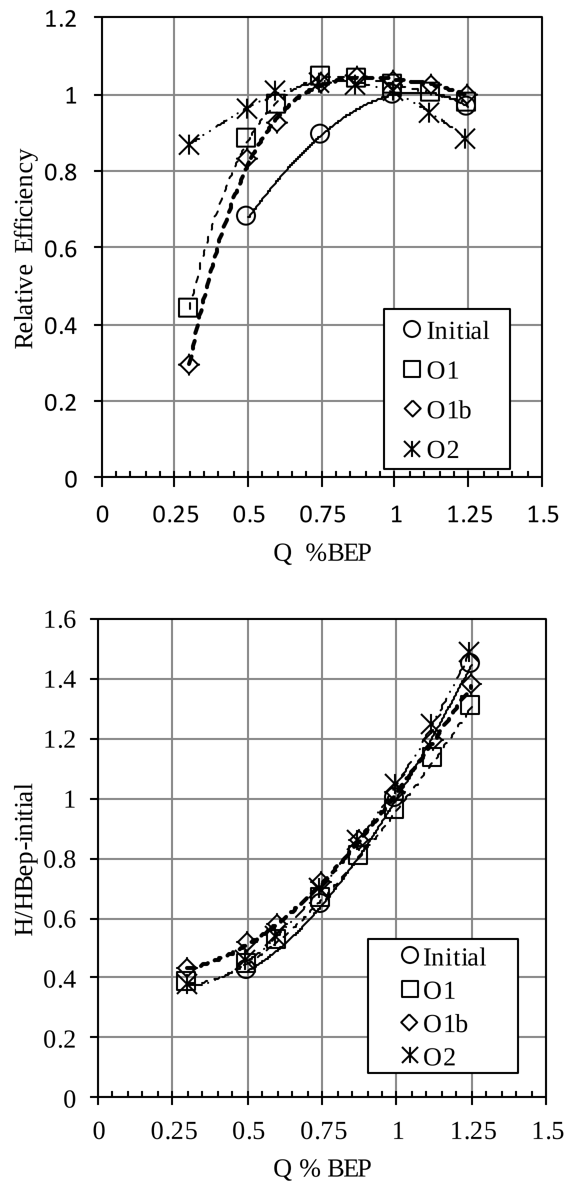

Fig. 16 Performance curves comparison of optimized designs

From these curves we can see that the efficiency curves assume a flatter shape for all the optimization cases considered. Moreover, the BEP is shifted to lower flow rates as a result of the increased number of blades. An average of 2% increase of efficiency at the design flow point can be seen. On the other hand, the head curves appear identical for the initial and the splitter blade case, up to the 75%Q point, and then slightly increasing for higher flows. From a comparison between the initial shape and after the optimizations it appears that an improved performance is achieved as a result of the

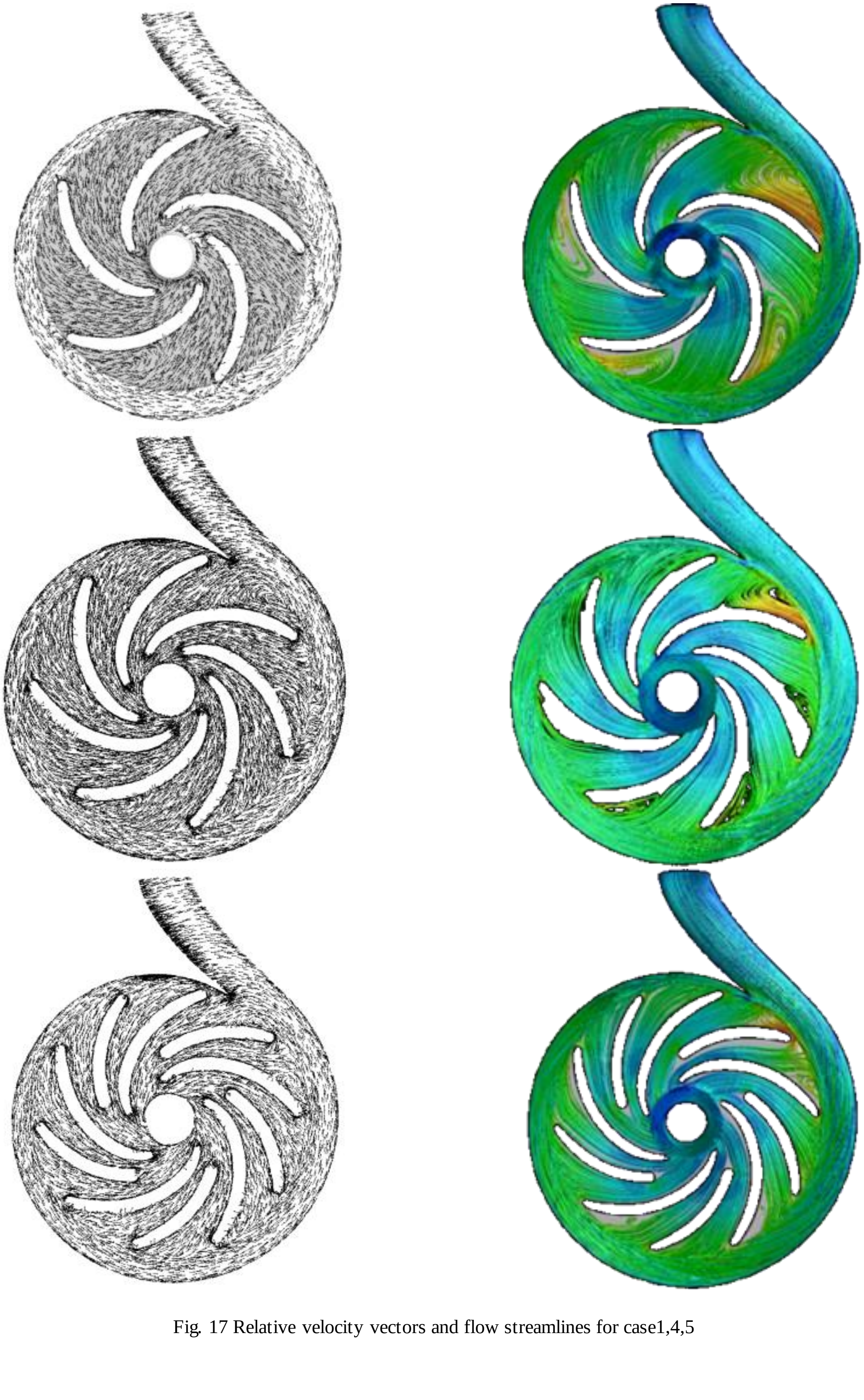

Fig. 17 Relative velocity vectors and flow streamlines for case1,4,5

change in geometry, which better conforms to the design flow conditions. The optimized geometries show an improved performance across an extended operating range showing a 2.3%-2.7 efficiency increase at the design flow point (table 4) and up to a 4.4% increase at a flow rate of 125%Q. In fig.17, O2 shows a better performance variation for lower flow rates, while a sudden drop in performance is observed for flow rates higher than 110Q. Furthermore, it should be pointed out that in all optimization cases even higher efficiencies could be achieved for partial flow conditions.

In fig. 18 the pressure distribution for the initial and the final optimized cases are compared. The optimized geometry shows a more favorable distribution in the stream wise direction, as well a reduced pressure loading. This is due to the increase of blade number and the reduction of the pitch wise distance between successive blades, which also leads to an increase in the minimum pressure value.

Table V Optimization results

case	Q[m^3/h]	H[m]	ω[r/s]	ηrel[%]
initial	29	109.89	303.69	100
O1	29	110.41	303.69	102.3
O1b	29	110.85	333.66	102.7
O2	29	113.23	303.69	101

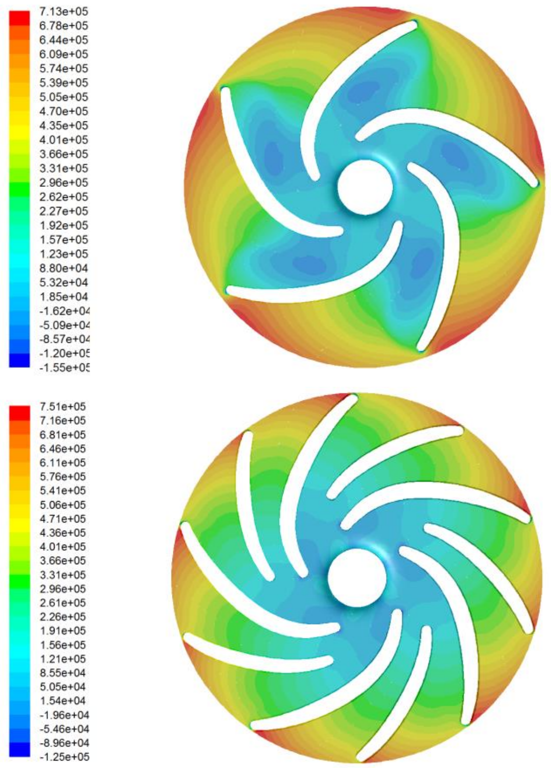

Fig. 18 Static pressure contours of initial and final runners

Finally, fig.19 shows a comparison between the initial runner design and the optimized one. It is evident that the smaller wrap angle blade shows better performance at the desired operating point.

VII. Conclusions

In this paper the numerical design of a centrifugal pump in reverse operating mode was presented. The effect of various design modification on turbine mode operation of a centrifugal pump were shown, and simple performance improvement strategies were identified.

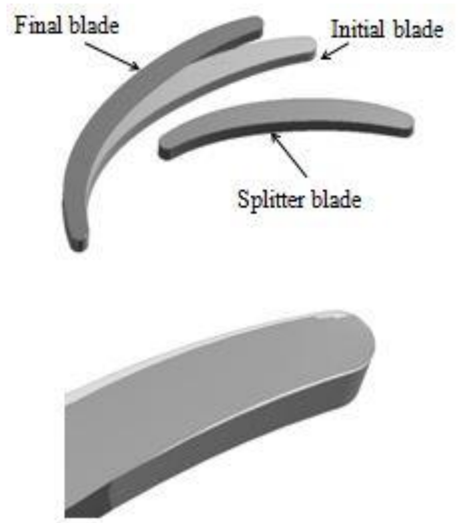

Fig. 19 Comparison of initial and final geometries

The results presented here, were later used for the determination of proper design parameters ranges as well as to indicate which parameters were more important. The results suggested that the small number of blades together with the abrupt change of width at the inlet of the runner caused a severe recirculation in the flow passage and increased losses. These losses could not be predicted when the interaction of the spiral casing and the impeller was ignored. Furthermore, using a larger number of blades lead to a shift of the performance curve to larger heads as well as higher efficiencies for the same flow rate. Following the results of the flow analysis of the initial geometry, an optimization study was performed. In the final geometry the blade outlet angles were modified to reduce flow recirculation and at the same time the blade thickness was also reduced leading to a reduction the outlet edge velocity magnitude. An overall increase of 2.7% in efficiency in turbine operation was achieved at the design flow operating condition. The results of this study show that an efficient low cost turbine can be obtained through a careful redesign of a pump impeller to be operated as a turbine.

Acknowledgements

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