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Uticaj pojedinih konstruktivnih parametara radijalnih lopatica sprovodnog aparata na hidrauličke karakteristike Fransisovih turbina sa promenljivom brzinom

Influence of Particular Design Parameters of Radial Guide Vane Cascades on Their Hydraulic Performance at Variable Speed Operated Francis Turbines

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Rezime - Projektovanje radijalnih lopatica sprovodnog aparata turbina treba da zadovolji potrebe radnog kola turbine da dostigne određene projektne parametre. Za režime rada konvencionalnih turbina, hidrodinamički parametri u prostoru predkola mogu se relativno lako proceniti. Za rad turbine sa promenljivom brzinom pri konstantnom naporu, ovi parametri se menjaju na način da se ne mogu lako predvideti korišćenjem konvencionalnih tehnika. U ovom radu, sprovodni aparat, kao ključni turbinski podsistem koji treba da izvrši efikasan dovod vode radnom kolu u ovim radnim režimima, numerički je testiran u različitim geometrijskim oblicima i konfiguracijama, da bi se posmatralo ponašanje kaskade, na način da se proceni koja konfiguracija zadovoljava određena hidrodinamičke zahteve i zahteve efikasnosti radnih tačaka od interesa. Kriterijumi koje sprovodni aparat treba da zadovolje zasnivaju se na hidrodinamičkim parametrima i proširenju radnog opsega turbine.

Ključne reči - sprovodni aparat, CFD, Fransisove turbine, rad sa promenljivom brzinom (VSO)

Abstract - The design of radial blade cascades, with the intent to become turbine guide vanes, lies in the basis of satisfying turbine runner needs for a particular turbine design point of interest. For conventional turbine operation, the hydrodynamic parameters in the pre-runner space can be relatively easily estimated. For variable speed turbine operation at the constant head, these parameters change in a way that cannot be easily predicted using conventional techniques. In this paper, the guide vanes, as crucial turbine sub-system which needs to perform efficient runner flow feeding for these operating ranges, are numerically tested in various geometrical shapes and configurations, to observe the cascades behaviour, in a way of estimating which cascade configuration satisfies certain hydrodynamic and efficiency considerations of the operating points of interest. The criteria which the guide vanes need to meet are based on hydrodynamic parameters and expansion of the turbine operating range.

Index Terms - Guide Vanes, CFD, Francis Turbines, Variable Speed Operation (VSO)

I Introduction

The design of guide vanes, more or less, so far has been developed for constant synchronous turbine runners, to meet some hydraulic criteria considering the turbine power output at given head/discharge conditions [1-3]. For variable speed operating conditions, things changes drastically when the guide vanes design is considered, where multiple points of interest shall be met [4, 5]. The variable speed operating physics has been previously explained [6, 7] and this research represents a statistical upgrade towards definition of the influential geometrical constraints that the guide vane cascade has to the hydrodynamic and efficiency characteristics of the turbine.

The variable speed operated (VSO) Francis turbine is developed under the circumstances of constant head. Several methods of operation can be examined, and by those, the developing procedure of designs can be established and further optimized. First method is that the VSO of a turbine is strictly depending from the turbine universal characteristic chart (hill chart), by tracking local most efficient zones in a certain range of rotational speed. In that way, variable-speed operation gives the opportunity to adjust and optimize the rotational speed of the runner according to the available head for each guide vane opening.

Second observed method, which is, is focused on the result obtained from VS operation, i.e. expanding the operating range of the turbine, overall, and what hydrodynamic conditions has to be present in the pre-runner space to obtain operating range expansion. The definition of the hydrodynamic conditions for different rotational speed of the turbine runner directly influences the design of the guide vane cascade and the blades in the cascade.

The VS operation, the hydrodynamic conditions and the blades design are interacting between each other, so strict correlation from one to another cannot be made easily, and that is why, several partial analysis have been made: how the geometry influences the hydrodynamic conditions and how the changes of hydrodynamic conditions, due to VS operation, influences the geometry. In this paper, the first approach has been analysed, defining which cascade configuration gives shrinkage or expanding of the turbine operating range.

II Variable Speed Operation Interpretation

The VSO turbine, as it was mentioned, can be examined in two ways. First, according to the universal hill chart of turbine, where tracking of the local most efficient zones is done.

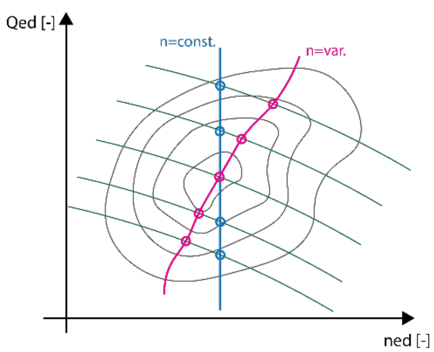


Figure 1. Variable speed operated turbine – example [5]

From fig.1, it can be seen that the guide vanes design is not influencing the turbine operation, as the variable speed method taken here can be achieved with ordinary guide vane cascade. Thus, the hill chart shape also is obtained for a selected guide vanes configuration, and the situation looks like this.

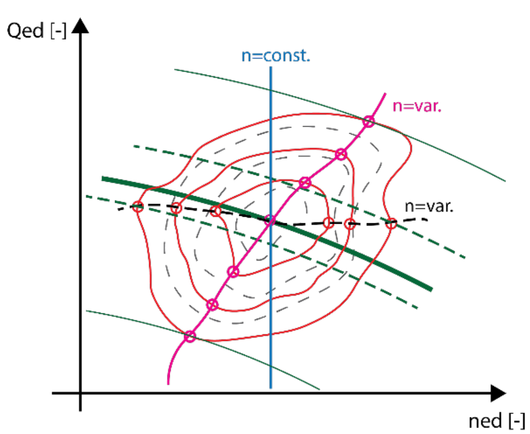


Figure 2. Guide vane reconfiguration and VSO

If we observe the guide vanes, how they influence on the general shape of the hill chart, the main goal is to expand the turbine operating region, i.e. how the guide vanes configurations and influence on the hill chart shape. The following scheme is expected (fig.2).

From the chart on fig.2 it can be seen that the guide vane curve characteristics are wider and it is expected to expand the operating region of the turbine (red iso-lines). Obtaining a guide vane configuration which will expand the operating region, than, by variable speed operation as in fig.1, the benefits are evident and the VSO effect is more dominant (magenta curve or black dashed curve).

This interpretation of the VSO turbine by operating schemes and regions is only sufficient for observing the expected overall turbine behaviour. The hydrodynamic parameters which leads to obtaining such expansions, are mainly defined from the turbine runner inlet flow conditions when VSO is present, and how they can be achieved with the guide vanes, which means that they influence the guide vanes design and vice versa. The velocity parallelograms in front of the turbine runner for VSO are presented on fig.3.

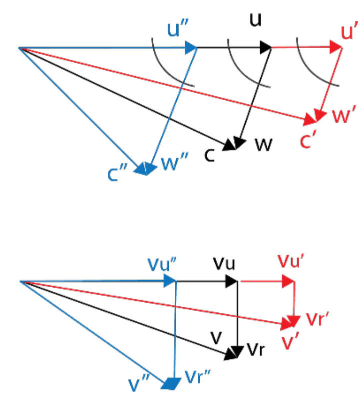


Figure 3. Runner inlet velocity triangles and guide vanes outlet velocity parallelograms

From the schematic charts from fig.1 and 2, it is evident that for constant head and change of rotational speed, the discharge of the turbine changes, either it increases or decreases. On basis of that, the velocity triangles were developed (fig.3) and the velocity parallelograms at the guide vanes outlet. The following velocity vector configurations show that the guide vanes shall meet these velocity ratios for VSO turbine. It can be noticed that the runner inflow angle is kept constant, as the inflow conditions is desired to remain as more efficient as possible, by providing a shock-free (zero incidence) inflow conditions to the runner [1].

III Guide Vanes Cascades Numerical Tests

As the last statement was to ensure shock-free inflow conditions to the runner, taking into account this criteria, several geometries were developed and tested through CFD analysis, to obtain how particular geometrical parameters influence on the overall turbine characteristics, when operation region expansion is demanded by VSO. The tested geometrical parameters of the cascades are the density *L/t [-]*, the relative opening of the guide vanes *a/L [-]*, the inlet cascade radius relative to the outlet C_{RI} *[-]* and the relative angular positioning of the blade chord length regarding the turbine centre of rotation φ_N *[-]*. These geometry parameters are general for description a various types of radial cascades, where some detailed parameters such as blade thickness distribution or maximal thickness location etc. are neglected for further analysis.

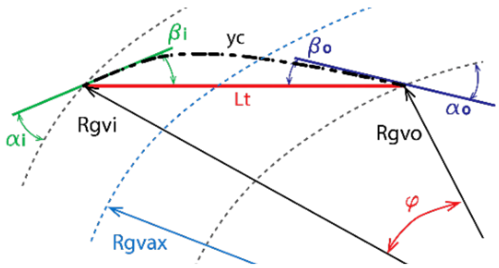


Figure 4 Blade in the cascade description

All the configurations are developed for shock-free inflow conditions to the cascade and the trailing edge bending angles are developed for shock-free flow entrance into the runner, for the given turbine inputs (tab.1).

Table 1. Turbine input parameters

Net head	Hn [m]	11,2
Design flow rate	Qd [m3/s]	0,2
Design rotational speed	nd [rpm]	333,33
Runner inlet diameter	Dr1 [m]	0,62
Runner outlet diameter	Dr2 [m]	0,349
Guide vanes height	Bgv [m]	0,06
Speed Factor (IEC60193)	ned [-]	0,185
Discharge factor (IEC60193)	Qed [-]	0,1567

The configurations geometries were developed in MATLAB and transferred to ANSYS Workbench, where the flow domains were created. The mesh was built in ANSYS TurboGRID and assembled with previously meshed runner of the Francis 99 turbine. The numerical model is reduced, consisted only from the guide vanes, the runner and the draft tube cone. The numerical mesh is consisted of approx. 1.5 million cells, where approximately 550 to 650k cells were used in the guide vanes, the runner is consisted of 810k cells and the draft tube cone from 150k cells.

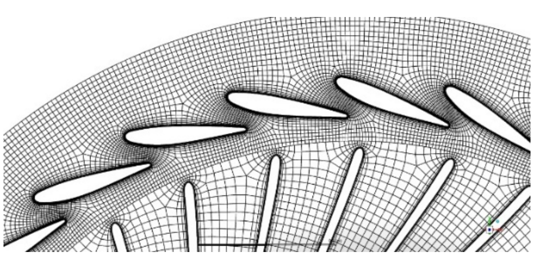


Figure 5. Mesh preview – example of the simulated models

The boundary conditions of the model are inlet and outlet total pressures, to obtain the value of the turbine design head, where at the inlet, a cylindrical components of the vector were adjusted. The turbulence model used is standard *k-ε*. The runner of the Francis 99 turbine is consisted of 15 full blades and 15 splitter blades, which are assumed as “moving walls” along with the runner hub and shroud surfaces, with no slip conditions. The runner domain frame is given motion around the vertical ‘z’ axis. The guide vanes models are consisted from 24 up to 32 blades. The frames are connected between each other via interfaces, which allows further easiness of transient simulations using the sliding-mesh technique. The simulations were guided as steady, changing the runner rotational speed in the range of ±15% of the nominal speed.

IV Results

The guide vanes configurations are developed in one position, according to the turbine design point i.e. the best efficiency point. The same configurations are tested for off-design point of ±15% of the nominal speed.

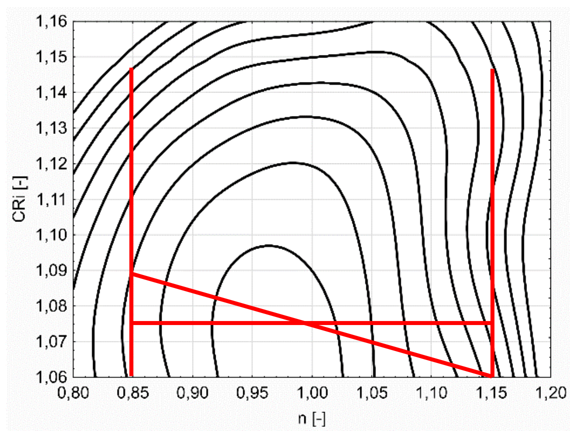


Figure 6. Influence of cascade inlet radius on operating range expansion at different rotational speeds

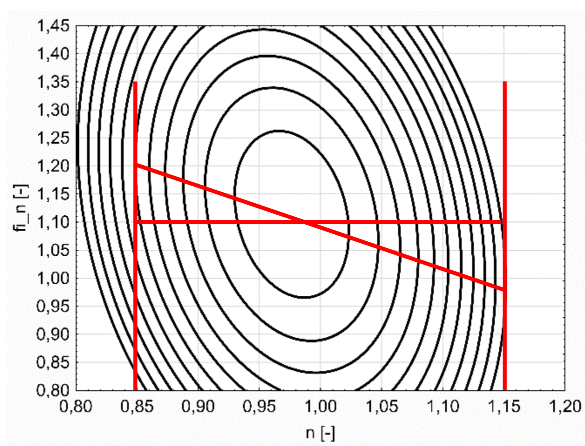


Figure 7. Influence of cascade blades chord line angular position on operating range expansion at different rotational speeds

It can be concluded from fig.6. that the zone of optimal operation lies at the turbine design rotational speed n=1 [-], and with a cascade inlet radius from 5% to 10% larger than the outlet radius. Regarding the expansion of the operating range, it can be concluded that the iso-lines are most stretched at 7.5% of inlet radius ratio, which can be the guidance for further development of the "optimal" cascade.

From fig.7, it can be seen that the biggest extension of the range is when a variable angular position of the chord is given. As this, technically is plausible if the blades are "profiled" and have a changeable angular shape, will be analyzed further. For standard radial cascades of fixed blade geometry, the weighted angular position of the chord shall be around 1.11 [-].

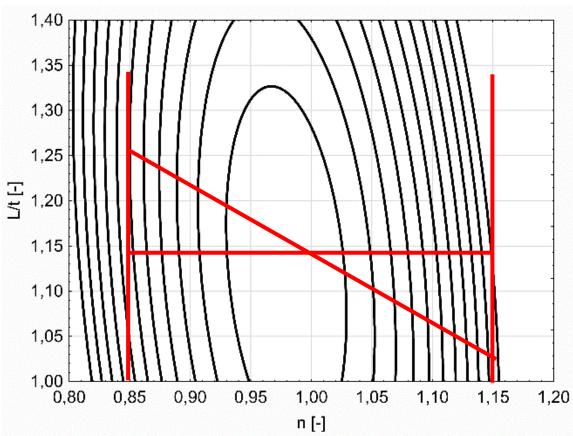


Figure 8. Influence of cascade density on operating range expansion at different rotational speeds

On the chart at fig.8, it can be easily concluded that the cascade density i.e. the overlap between the blades influences on the expansion of the operating range. Most expand region is obtained between 10%-25% of blades overlap, i.e. cascade density of *L/t=1.1 - 1.25* [-].

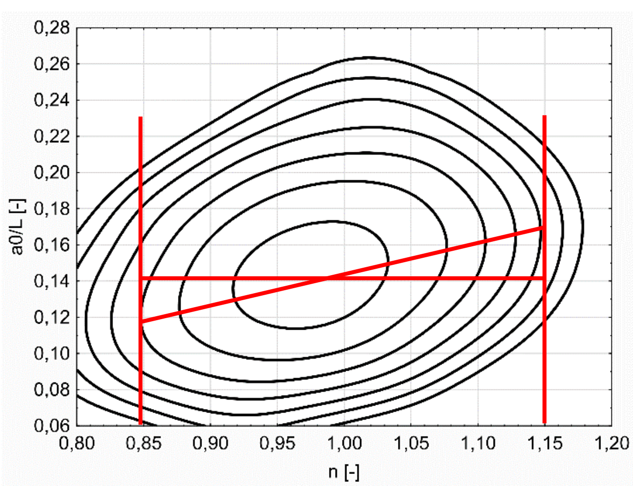


Figure 9. Influence of blades relative opening to their length on operating range expansion at different rotational speeds

The blades relative opening shows that maximal operating range expansion is obtained when there is a change of the opening from 12% to 17% (fig.9). If the cascade is built from constant blade configurations, the weighted value of the relative opening shall be in the range around 14%-16% of the blade length.

V MODELS COMPARISON

By taking the previous assumptions and calculations, how the cascade types behaves when variable speed is needed, for obtaining an expanded hill chart, previous analysis were guided to test and prove this approach for further development.

A strict comparison was made with the existing guide vanes on the observed turbine model Francis 99 which is in the Waterpower Laboratory at NTNU. The guide vanes are consisted of 28 blades, and the cascade has the following geometry parameters (tab.2). The developed model is geometrically very close to the existing guide vanes, as the turbine model has very tight geometrical constraints where the guide vanes can be positioned and examined. The following analysis was carried out for further development of this cascade and to further perform model tests.

Table 2. Model configurations compared

	CRi [-]	L/t [-]	a0/L [-]	fi_n [-]
FR99	1,126	1,324	0,166	1,23
MODEL1	1,127	1,3	0,148	1,25

The main difference between the cascades is the relative angular position of the chord lines of the blades and the relative opening of the blades. The developed Model 1 has slightly smaller opening, related with the conclusions from fig.10. The other geometry parameters are selected to ensure that the developed cascade can be fitted in the turbine model test rig.

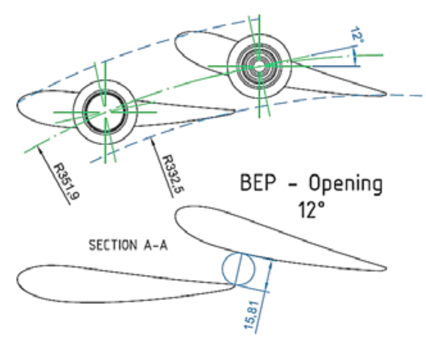


Figure10. Developed guide vanes model

For several guide vanes openings and rotational speeds of the runner, for equal operating conditions, the turbine hill charts were obtained and plotted on relative efficiency scale from 0.9-1 [-].

From the results obtained in the hill charts, it can be easily concluded that, by respecting only one of the previous derived criteria (in this example the relative guide vane opening), an expansion of the turbine operating range is obtained, especially in the range of reduced rotational speed and reduced flow rates. This is significant increase of the operating range which intuitively shows that this turbine can operate with good efficiency out of the previous described operating limits.

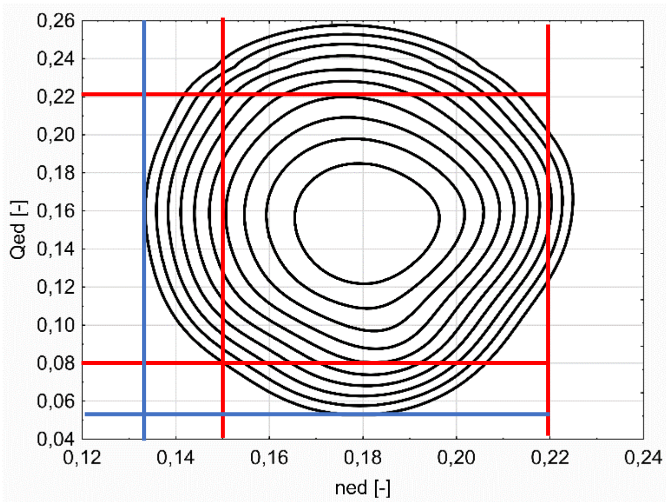


Figure 11. Developed guide vanes (Model 1) – turbine hill chart (pre-described operating limits and extensions)

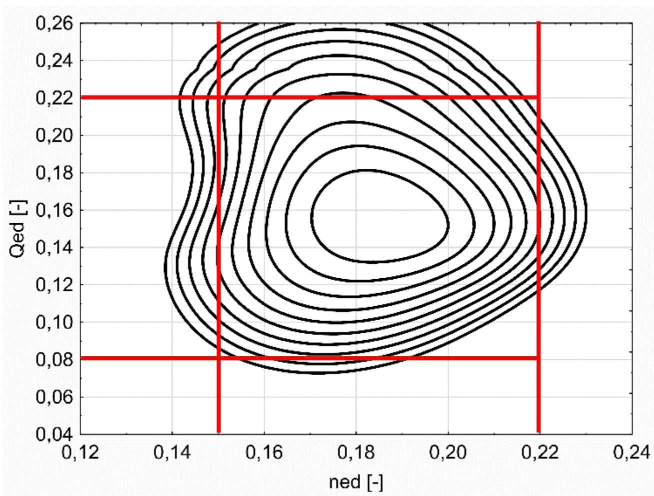


Figure 12. Francis 99 guide vanes model – turbine hill chart (pre-described operating limits)

According to fig.11 and 12 it can be concluded that the turbine characteristics compared for the 2 models, extends. Model 1 gives better performance compared to FR99 guide vanes. Extended characteristics is obtained for reduced rotational speed, compared as the ratio between the limitations:

$$\frac{n_{edFR99}}{n_{edM1}} = \frac{0.15}{0.132} = 1.364\ [-]$$

For reduced flow rates, the characteristics extends compared with the previous limitation:

$$\frac{Q_{edFR99}}{Q_{edM1}} = \frac{0.076}{0.052} = 1.462\ [-]$$

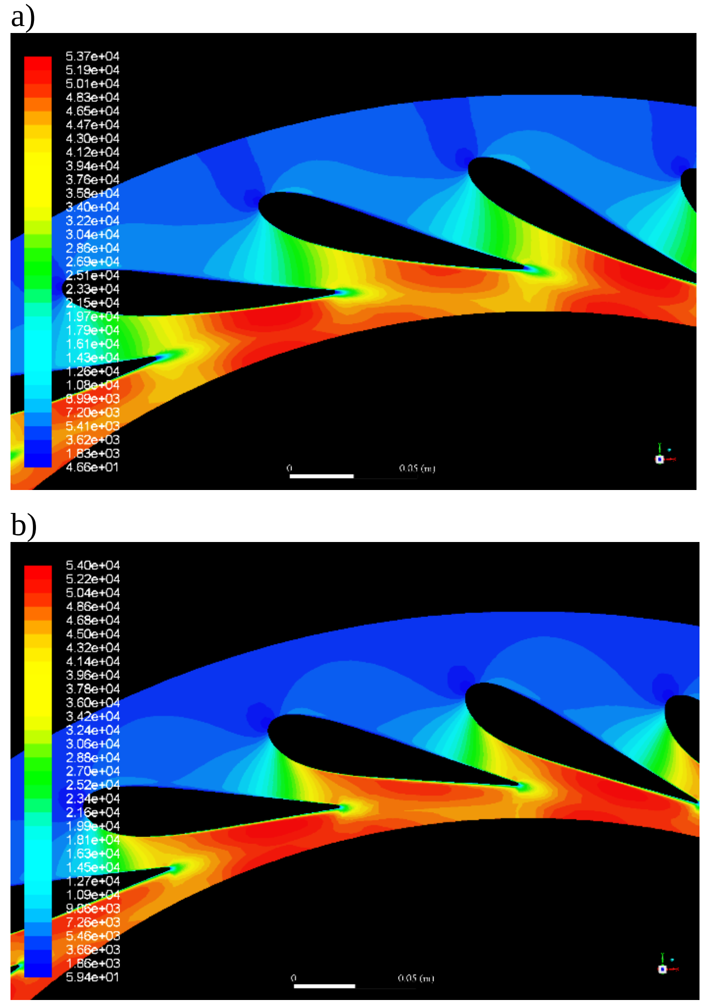
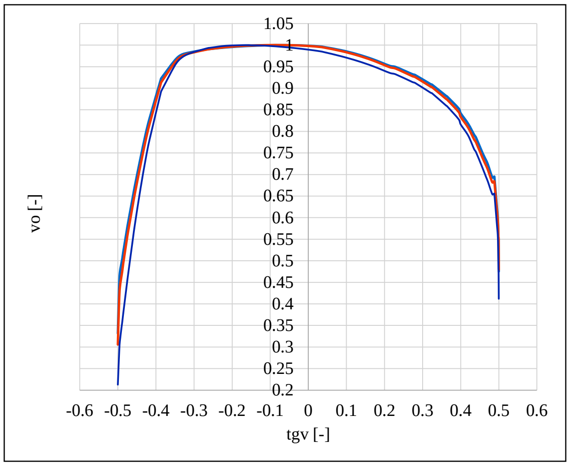


Figure 13. Dynamic pressure contours
a) (Developed Model 1), b) (Francis 99 Guide vanes)

The velocity profiles at the outlet pitch of the blades is compared to show the plausible hydrodynamic reason for extension of the operating range. The velocity profiles are analyzed also in relative manner, to obtain not the intensity, but the uniformity (asymmetry) of the profile, given on equal blade pitch.

From the chart on fig.14 it can be concluded that the velocity profile obtained for the developed model 1 remains almost equal for off-design rotational speeds, compared to the existing guide vanes, which for increases rotational speed of the runner, the velocity profile deforms and losses its symmetry.

According to these results, a decision was made to further develop a model of the guide vanes, to perform further measurements and mode tests, to prove these characteristics of operating range expansion and how does the guide vanes, overall, can re-shape the operating hill chart of the turbine.

a) 

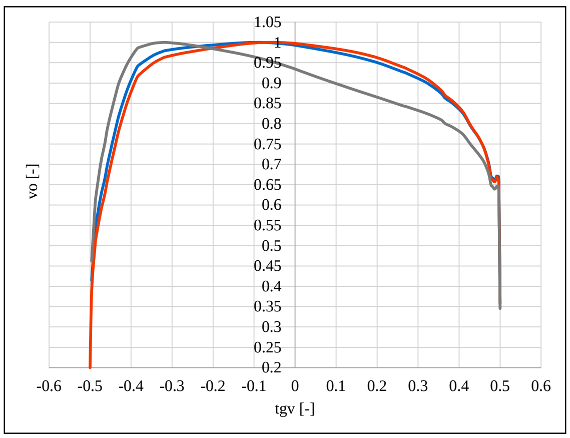
b) 

Figure 14. Velocity profile
a) Model 1, b) Francis 99 Guide Vanes

VI CONCLUSION

In this paper, a brief analysis was carried out for determining some of the influential geometry parameters of guide vanes, when variable speed operation is expected of the turbine runner, to cover more operating zones. Primarily, the guide vanes cascade is multiple times described and geometrically parameterised in the previous researches done, so a continuing to those researches was made here to the next step.

Several cascade configurations (particularly 21 configurations) were examined, with various geometry parameters differences, from different cascade inlet radius, different angular position of the blades chord line, different cascade density, opening etc. All these geometry parameters were tested with CFD simulations on a reduced numerical model of the Francis 99 turbine model from the NTNU Waterpower laboratory. All the cascades were developed for the best efficiency operating point of the turbine. The tests were carried out at constant head and opening, for variable rotational speeds of the runner. The results were plotted as contours (iso-lines) of 3 variables, the geometry parameter observed, the rotational speed and the turbine efficiency, to observe how the operating range extends and the optimal zone shifts with the change of rotational speed or with the change of the particular geometry parameter.

According to the primary results, it was obtained how certain geometry parameters influence the extension of operating range of the turbine. A comparison was made for a developed guide vane cascade (Model 1) with the existing guide vane cascade of the Francis 99 turbine model, to observe how the blades opening change influence the range extension. The results showed extension of the range, so a decision was made to further build a model of the guide vanes to prove these operational extensions, by performing model test at the laboratory.

From this analysis, generally, it can be concluded simply that the design of the guide vanes cascade is not unambiguous, but quite opposite. They represent a matter which needs to be evolved iteratively by certain combinations, which can give reasonable and favourable hydrodynamic conditions in the pre-runner space, and on the overall turbine efficiency.

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