NUMERICAL ANALYSIS OF THE EFFECT OF SPLITTER BLADES ON DRAFT TUBE CAVITATION OF A LOW SPECIFIC SPEED FRANCIS TURBINE

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ABSTRACT

The formation of the suction cone vortex cavitation imposes constraints in the operating envelope of Francis turbines. Splitter blades have been used in the past in low specific speed turbines to increase efficiency and improve pressure pulsation characteristics at part load conditions. In this paper, the draft tube cavitation performance of a low specific speed Francis turbine with splitter blades was investigated. The steady and unsteady RANS equations were solved using commercial CFD software to analyze the flow and obtain the performance and draft tube cavitation characteristics of the original runner, while the predicted efficiencies were compared against experimental data of a model Francis turbine installed in the laboratory. The flow field and blade pressure distributions were compared using different splitter blades with different blade area ratios. An analysis of the flow using both a single phase calculation and a two phase model to capture the part load vortex core at the runner outlet was also performed. Subsequently, the effect of splitter blades with various area ratios and circumferential positions on the inception, form and evolution of the draft tube vortex was compared. The results showed that by a careful selection of design variables, an improved efficiency and cavitation behavior can be achieved.

KEYWORDS

Francis turbines, draft tube cavitation, splitter blades, numerical analysis, experimental comparison

1. INTRODUCTION

Over the years the requirement for increased energy generation has driven the need for improvements in the performance of Francis turbines over extended operating conditions. However, operation in off design conditions is associated with poor performance and the appearance of the draft tube cavitation in part load conditions. This has highlighted the importance of improving the efficiency of new designs and the refurbishment of existing plants through careful hydraulic design of components and, more importantly, runners. For high head machines, such an improvement can be achieved by introducing splitter blades [1, 2].

Recently, more interest has been shown in hydraulic machines with splitter blades. Several studies have been published demonstrating the advantages of such an approach. The addition

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of splitter blades leads to a modification of the pressure distribution both in the suction and pressure sides of the runner blades improving the cavitation characteristic of the turbine [3] Furthermore, by reducing the large blade to blade area present in high head runners, secondary flows within the flow channel can be significantly reduced, leading to an improvement of the operational characteristics of the turbine [3]. A significant parameter that affects the performance of the runner with splitter blades is the blade length ratio. In [4] a study was performed to determine the best combination of length ratio and pitch position on the overall performance of the runner. The authors conclude that a substantial improvement can be achieved if the length ratio is carefully selected. On another work [5], the authors show that from a careful selection of length ratio, the unsteady characteristics of the turbine call also be reduced. They show that apart from performance improvement, the addition of the splitter blades can also reduce the pressure pulsation amplitudes within the turbine. Similar conclusions were drawn from the work presented in [6], where removal of the splitter blades caused an increase in pressure pulsations in an experimental Francis turbine. All the above, further support the beneficial effect of splitter blades, leading an overall improvement in the performance characteristics of high head Francis turbines.

In this work, the effect of the introduction of splitter blades on the performance of a low specific speed runner is investigated. The numerical results of the initial design without splitter blades were compared to experimental data, showing good correlation. Subsequently, the effect of cavitation modeling on the determination of the pressure distribution and flow field at the outlet of the runner was compared for three operating conditions, in order to determine the qualitative possible effect on the flow field. Finally, the effect of the splitter blades on the efficiency and draft tube vortex was compared.

2. NUMERICAL MODEL

The numerical model consisted of both a blade to blade domain for the steady state calculations, and the complete runner with the draft tube suction cone, for the unsteady calculations. The outlet diameter of the runner is 210mm with 13 blades. In fig. 1 the computational domains for both configurations and computational meshes used are shown, respectively.

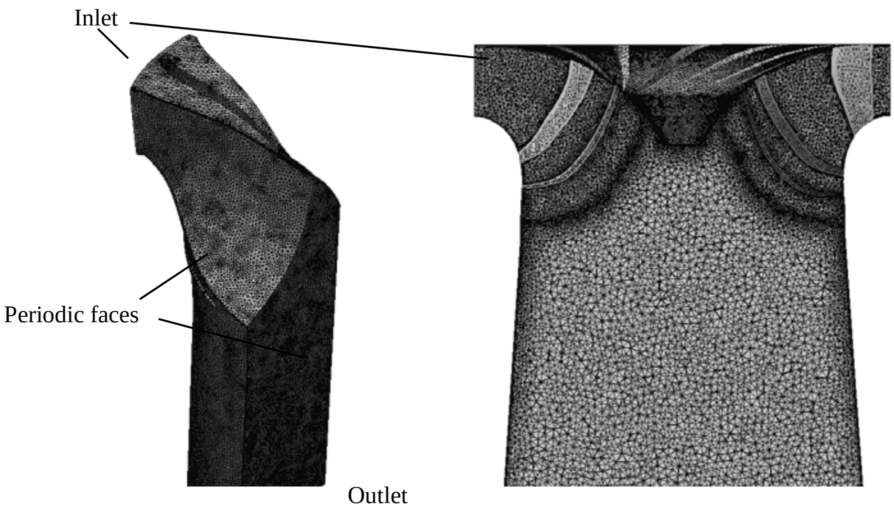


Figure 1 Computational domains and meshes used in current calculations

The numerical model corresponds to a ns=122 model Francis turbine installed in the laboratory with a reference diameter of 215mm, inlet width and diameter of 46.6mm and 261mm, respectively. For both steady and unsteady calculations, the realizable k-e turbulence

model was used and a second order discretization in time and space was employed. The boundary conditions used in the present calculations were extracted from full turbine steady state simulations by setting the inlet flow rate, while at the outlet a zero static pressure boundary condition was set. For cavitation modeling the 2-phase mixture model was used, while for the unsteady calculations a time step equal to 4^o rotation was set. The steady state simulation results were used as initial conditions. With this set up at time periodic solution was obtained after about 8 runner rotations.
The flow conditions that were considered for the comparison of steady, unsteady and cavitation simulations are shown in table 1

Point (OP)	Net Head Parameter Ψ	Flow rate parameter Φ
1	2.7	0.26
2	2.7	0.2
3	2.7	0.14

Tab. 1 Simulation parameters for unsteady and cavitation investigations

3. RESULTS

3.1 Comparison of stead, unsteady and experimental measurements

In fig.2 the numerical results of the present model are compared with experimental measurements. Overall, we can see that a good agreement between experiment and measurements is achieved as the numerical calculations closely capture the envelope curve of the turbine. Therefore, the present model can be considered valid for further investigations.

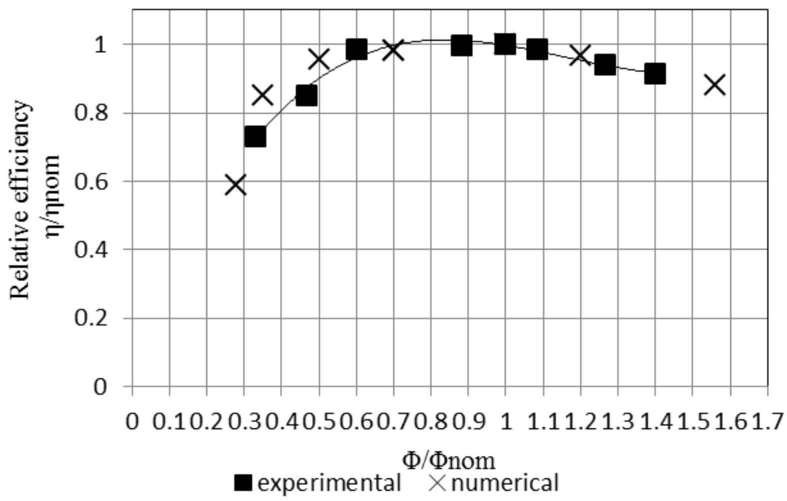


Figure 2 Comparison of experimental and numerical data of a model francis turbine

In order to compare qualitatively the flow downstream of the runner, the vorticity magnitude in three downstream locations is used. In [7], the pressure, vorticity and helicity distributions were used to analyze the vortex flow in the draft tube of a Francis runner. As the vorticity represents the amount of rotation of a fluid particle as it flows within the turbine, for comparison purposes, the mass averaged vorticity magnitude will be used as a measure of flow rotation. In fig. 3 the pressure distribution on the runner blade for OP1 and the flow velocity evolution at various cross sections downstream of the runner, are shown. From this figure it can be seen that substantial swirl is present at the runner outlet, which causes the flow reversal near the axis of the runner.

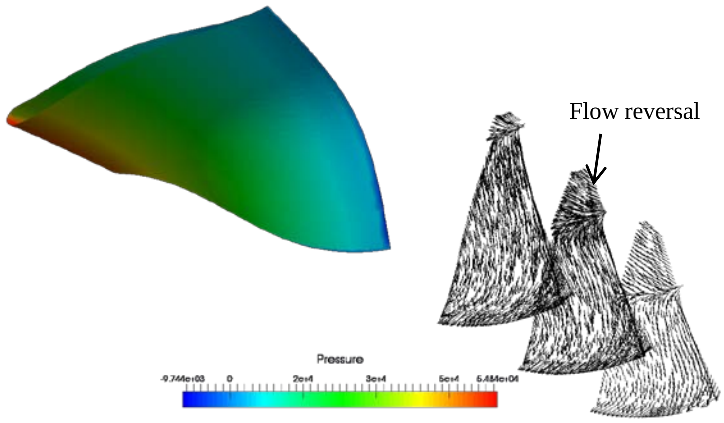
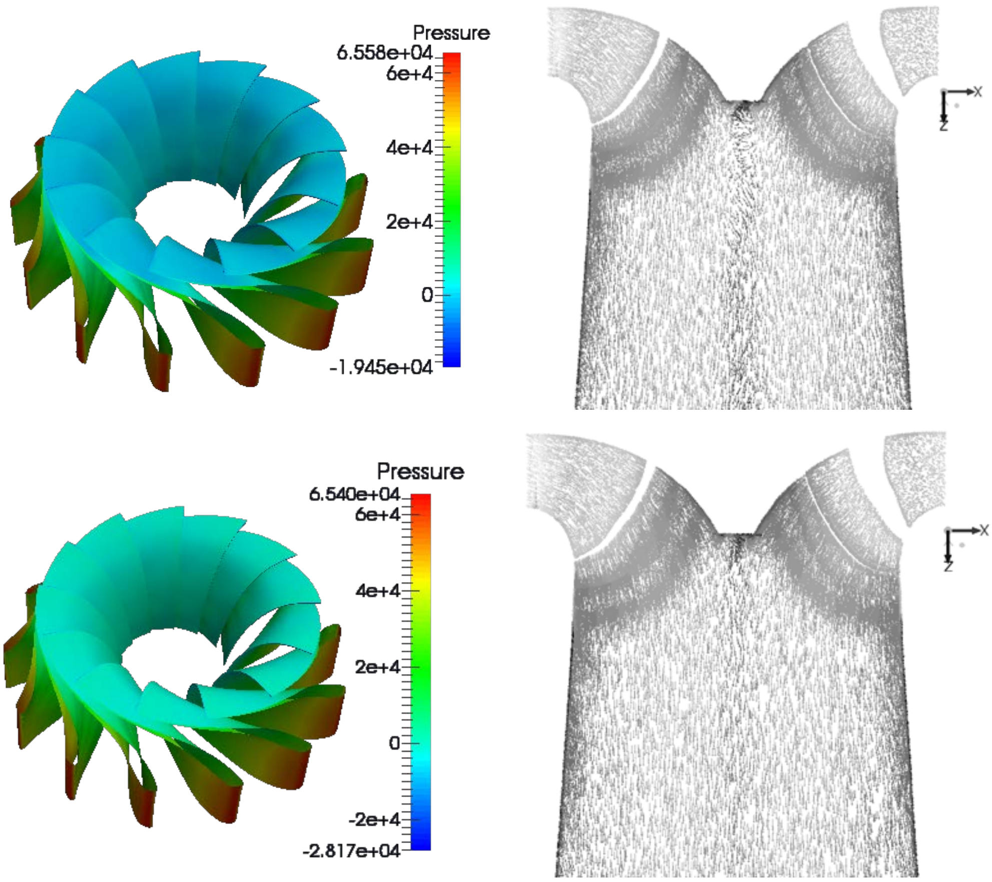


Figure 3 Pressure distribution on the blade of the initial runner and velocity field evolution downstream of the runner



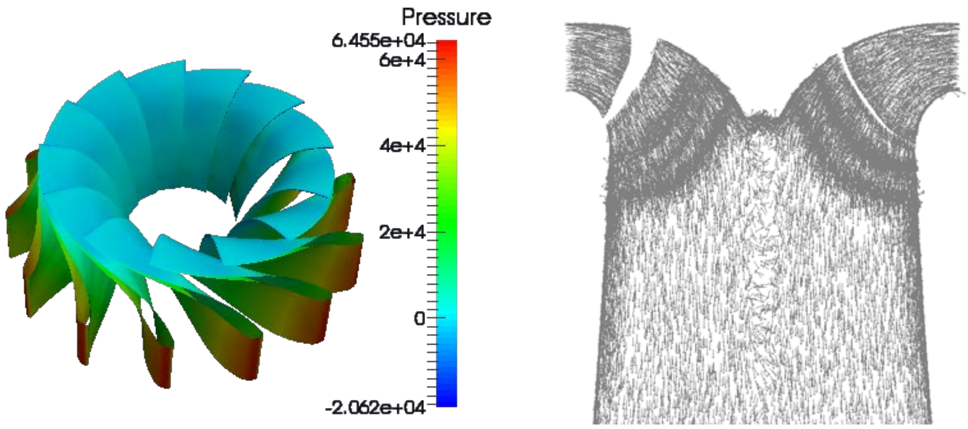


Figure 4 Comparison of runner pressure distribution and draft tube flow for OP1 under steady, cavitation and unsteady simulations

Table 1 shows the vorticity magnitude calculated at three downstream locations for OP1-3 of the full turbine and suction cone simulations. From fig. 4 it can also be seen that areas where the vortex core appears corresponds also to areas of large vorticity.

Case	Vorticity magnitude at z==0.84Dref[1/s]	Vorticity magnitude at z=1.16Dref [1/s]	Vorticity magnitude at z=1.49Dref [1/s]
Steady OP1	65.85	53.86	49.345
Cavitation OP1	50.66	42.49	38.68
Unsteady OP1	64.72	52.68	48.23
Steady OP2	73.7387	72.28	80.55
Steady OP3	149.67	140.19	146.1

Tab. 2 Vorticity magnitude for three runner downstream locations for OP1-3

OP1 is near 0.92Qnom, which near the theoretical point where the vortex core phenomenon appears [6]. The appearance of the vortex core in the steady state simulation well agrees with the unsteady calculations. This can also be seen from table 1, where the vorticity magnitudes for all monitoring positions are almost identical. On the contrary, when cavitation modeling was included in the steady state analysis, the vortex core could not be captured. This was caused due to the fact that the operating point chosen was near the nominal flow rate where no residual swirl is present in the turbine and numerical the inclusion of cavitation modeling has an impact on the flow evolution.

From figs. 4 and 5 the formation of the vortex core as the flow rate is reduced can be seen. Although with steady state simulations it is not possible to quantify the impact of the unsteady characteristics of the vortex core on the operation of the turbine, we can see that from a qualitative point of view, important information can still be obtained without needing to simulate the vortex core, which can be computationally costly. This can of great importance in the early stages of the design of a new runner, as it can aid in reducing the lead time required.

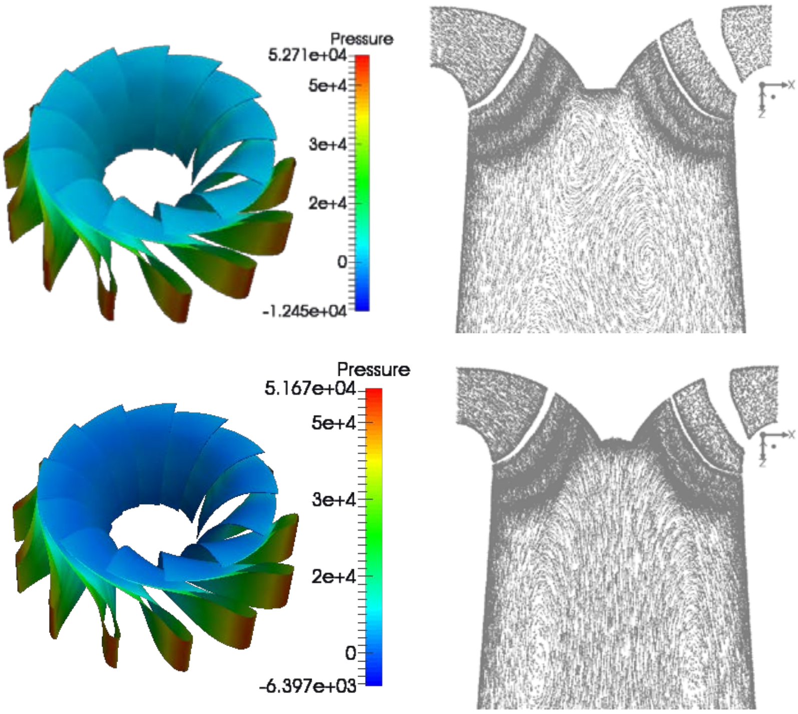


Figure 5 Formation of the draft tube vortex with reducing flow rate and pressure distribution on the runner blades

3.2 Effect of splitter blades on turbine performance

In table 3 the splitter blades used in the current analysis are shown. Usually, the main geometric parameter describing splitter blades is the length of the blades' meridional projection, compared to the respective length of the original blade, defined as the length ratio. However, this parameter does not take under consideration the wrap angle of the blade, which depending on the design, may be significant and may lead to misleading conclusion regarding the design of the runner. For this reason, the total blade area ratio is chosen as a distinguishing parameter. Apart from two different area ratios, an additional pitch-wise position of the splitter blade was chosen.

Case #	Area ratio (A_r)	Pitch position (P)
1	0.64	0.5
2	0.67	0.5
3	0.67	0.4

Tab. 3 Splitter blades design variations studied

Case#	Splitter design	Φ	Ψ	η
initial	-	0.26	2.6	0.943
1	Ar=0.64, P=0.5	0.26	2.59	0.961
2	Ar=0.67, P=0.5	0.261	2.58	0.922
3	Ar=0.67, P=0.4	0.259	2.59	0.895

Tab. 4 Effect of different splitter blade design on runner performance

From table 4 it can be seen that the addition of a splitter blade with smaller area ratio leads to a significant increase in the hydraulic efficiency of the runner. Specifically, approximately a 2% increase was observed. On the other hand, using a longer splitter blade has the opposite effect, and a 2% reduction in efficiency is observed compared to the initial runner. The negative effect is further amplified by reducing the circumferential distance of the splitter blade by 10%, as seen in the same table. This drop in efficiency is a result of the flow choking that takes place, as the flow passage area is reduced. The effect of the splitter blade on the blade pressure distribution can also be seen in fig.6. As the flow leaves the inter blade area, it is accelerated, leading to a pressure drop on the pressure side near the exit of the larger blade. The same effect occurs in the suction side of the blade in case 3, where the splitter blade is placed closer to the main blade. In terms of the flow evolution, case 1 shows the least vorticity downstream of the runner, compared to all other cases. However, from fig.6, it can be seen that the inclusion of splitter blade reduces the flow rotation in all cases and reduces the at low flow conditions.

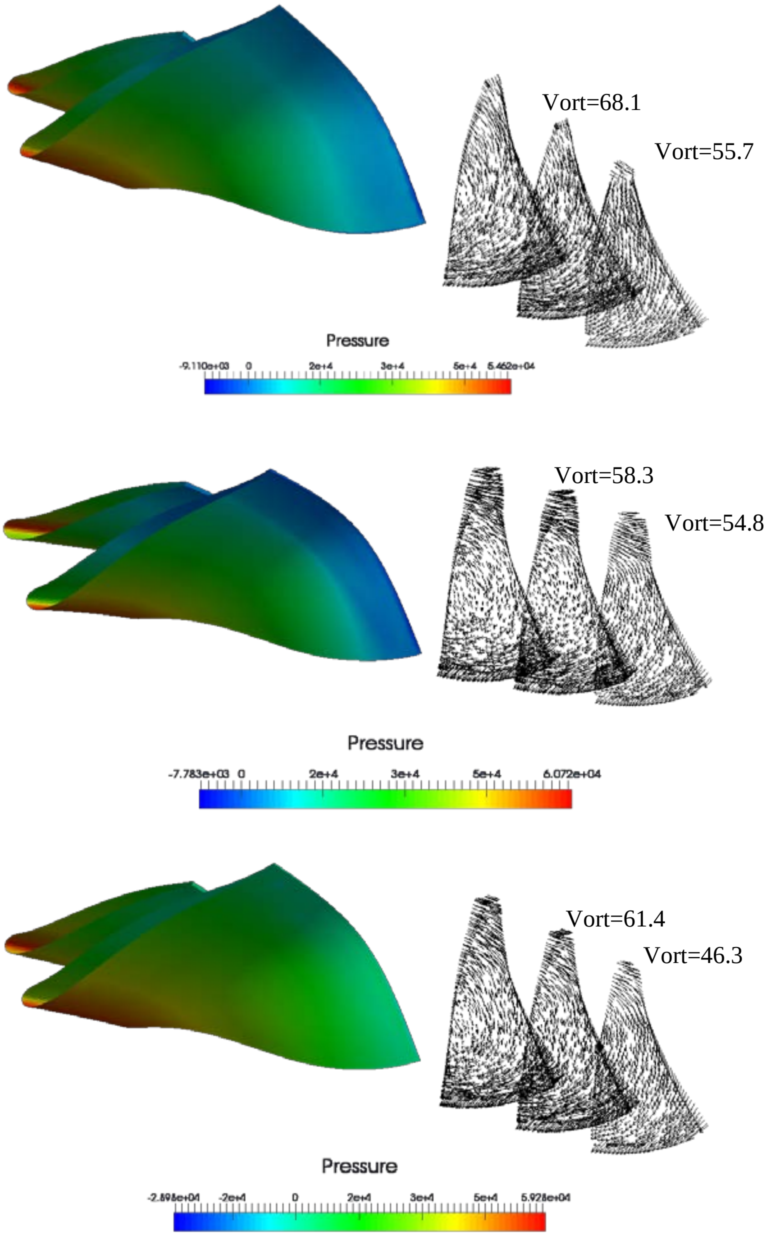


Figure 6Effect of different splitter blade design on the blade pressure distribution and the flow evolution downstream of the runner

4. Conclusions

In this paper, numerical simulations comparing the steady and unsteady simulation results for three operating conditions of a model Francis turbine were presented. The effect of simulation approach on the prediction of the main flow characteristics during part load operation, where the helical vortex core appears was shown. From the results it was shown that with steady state calculations, the main flow characteristics can be qualitatively obtained, which can be useful in the early stages of the design of new runners. Subsequently, the benefit of incorporating splitter blades on runner performance as well as part load performance was shown. The careful selection of splitter blade design variables can lead to important improvements in the efficiency and off-design operation of the runner.

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