Numerical investigation of draft tube pressure pulsations in a Francis turbine with splitter blades

I Kassanos, J Anagnostopoulos and D Papantonis
Laboratory of Hydraulic Machines, School of Mechanical Engineering, National Technical University of Athens, Heroon Polytechniou 9, Zografou, 15780 Athens, Greece
E-mail: ikassanos@gmail.com

Abstract. Operation of Francis turbines at part load conditions is related to the appearance of the draft tube helical vortex rope. Splitter blades have been employed in high head Francis turbines in order to improve performance as well as their unsteady characteristics. In this work the draft tube unsteady characteristics of a Francis runner with splitter blades are investigated numerically. Two different splitter designs were analysed, and the performance results were compared to the baseline runner with no splitter blades used. The amplitude of pressure pulsation caused by the precessing vortex rope as well as the related frequency was compared for all cases, for two different operating conditions. From the results a relationship between the pulsation frequency and splitter blade geometry was observed.

1. Introduction

Francis turbines are the most popular reaction turbines due to their high efficiencies, increased reliability and extended lifetime. In order to maximize energy generation, Francis turbines are often required to operate at part load conditions where the precessing vortex rope forms in the draft tube potentially resulting in unwanted pressure fluctuations, power swings and structural vibration [1]. For these reasons, this phenomenon has been the focus of significant research over the past years aiming towards understanding the flow patterns associated, predict the appearance of the phenomenon, understand and quantify its effects on the turbine and at the same time identify corrective measures.

Through experimental measurements the physical mechanism of the precessing vortex rope has been studied at various operating conditions with an effort to understand the underlying flow physics. By obtaining velocity measurements downstream of the runner the evolution of the vortex and its boundary as function of time and cavitation number has been studied [2,3]. These analyses provide useful insight on the behavior of the vortex rope and at the same time can pose a baseline facilitating the development of numerical techniques and prediction models.

With the development of computation fluid dynamics (CFD) tools, it has become possible to perform reliable simulations for the analysis of the of the flow patterns inside the turbine. Several studies have used this tool to simulate the flow in the draft tube in conditions where the vortex rope appears aiming towards the validation of the numerical approach against experimental data [4,5]. As draft tube performance is more important for low head machines, there has been little research on the flow observed in the draft tube of high head turbines. In high head turbines splitter blades have been employed to improve the efficiency of the turbine, minimize secondary flows and reduce pressure
fluctuation amplitudes [6,7,8]. Although the use of splitter blades has been popularized in high head machines, there is limited work studying the effect of the splitter blade geometry on the characteristics of the draft tube vortex rope at part load conditions. In this work the effect of the splitter blades on the draft tube pressure pulsation characteristics of two different splitter blade designs was investigated numerically.

2. Numerical Model

In the present study a turbine of specific speed equal to \( v_o = 0.26 \) as specified by the IEC standard [9] was investigated numerically with a reference runner diameter of \( D_{2e} = 210 \text{ mm} \) and 13 runner blades. In order to minimize the computational resources for this study, the distributor and spiral casing were neglected. The numerical model shown in figure 1, consisted of the runner, the draft tube suction cone, as well as a straight extension downstream of the suction cone. The cone angle was 4.2\( ^\circ \) while all the draft tube dimensions are given in figure 1 non-dimensionalised by the runner reference diameter \( D_{2e} \). The numerical grid used consisted of 2.11 million tetrahedral elements in total, of which 30\% were used in the runner while the draft tube was modeled using approximately 1.5 million elements. Two operating conditions were considered corresponding to 0.9 \( Q_{\text{BEP}} \) and 0.72\( Q_{\text{BEP}} \) at the design head. The inlet boundary conditions consisted of a constant radial and tangential velocity component corresponding to the guide vane opening for each flow condition, while a zero average static pressure was assumed at the outlet. In table 1 the corresponding boundary conditions and inlet turbulence quantities considered are summarized.

Previously it has been shown that the ke model is associated with a damping effect leading to an under-prediction of the pressure fluctuation amplitudes, while a better agreement with experimental amplitudes and frequencies can be achieved using the SAS model [5]. Recently, the SAS model has been shown to slightly overpredict the pressure amplitudes at higher harmonics compared to experimental values and the RSM model [10]. However, since in this study the effect of different splitter blade geometries on the draft tube vortex is compared numerically and in order to minimise computational requirements, the SAS model was considered adequate and used in all calculations.

![Figure 1. Computational domain and mesh under consideration.](image)

| Operating Point | \( Q/Q_{\text{BEP}} \) | Flow rate parameter \( \Phi \) | Net Head Parameter \( \Psi \) | Radial velocity \( v_r [\text{m/s}] \) | Tangential velocity \( v_u [\text{m/s}] \) | Turbulence Intensity [%] |
|-----------------|----------------------|----------------------|----------------------|----------------------|----------------------|----------------------|
| 1               | 0.9 * \( Q_{\text{BEP}} \) | 0.26                 | 2.7                  | 1.88                 | 8.55                 | 6                    |
| 2               | 0.72 * \( Q_{\text{BEP}} \) | 0.207                | 2.7                  | 1.51                 | 9.15                 | 6                    |

For the velocity and pressure coupling, a fully coupled method was assumed, while for all equations a second order discretization scheme in space and time were considered. Finally, a time step corresponding to a 1\( ^\circ \) rotation per time step was used.
3. Results

3.1. Numerical results of the initial geometry

In all the simulations performed, two monitoring locations at an axial position of \( L_1=0.147D_2e \) and \( L_2=0.57D_2e \) downstream of the runner were considered (figure 1). On each section four monitoring points were placed around the circumference of the draft tube wall corresponding to the intersection of the monitoring planes and the \( x(x_0, x_1) \) and \( y(y_0, y_1) \) planes. An initial solution was obtained using a steady state computation and a frozen rotor approach, before the simulation was switched to an unsteady analysis. After switching to the unsteady model approximately 1500 time steps were required before time periodic results were achieved.

In the following figures the results are presented in dimensionless form using the runner rotational speed for the frequencies and the turbine head for the pressure fluctuations. Figure 2 compares the pressure monitor signals in the frequency domain for the two downstream pressure monitors of the initial runner geometry considered \((L_1 \text{ and } L_2)\) at OP1. Comparing the response between the two downstream locations, it can be seen that the pressure fluctuation is quickly attenuated as the flow moves towards the exit of the suction cone. At the same time, the pressure fluctuations appear at the runner rotation frequency and its harmonics, with the dominant frequency appearing at \( 2f_n \). At these flow conditions the residual swirl (table 2) entering the suction cone isn't strong enough to initiate the formation of the vortex rope.

![Figure 2](image)

**Figure 2.** Frequency spectrum of the monitor pressure data for OP1 using the SAS model.

In figure 3 the pressure monitor signal for OP2 in the frequency domain is shown, where the dominant frequency of \( 0.3f_n \) is identified. Contrary to OP1, the swirl at the suction cone outlet has increased to a point where a recirculation region is formed towards the axis (figure 5), the flow becomes unstable and the vortex rope is formed.

![Figure 3](image)

**Figure 3.** Frequency spectrum of the monitor pressure data for OP2 using the SAS model.

| Case # | Area ratio \((A_r)\) | Pitch position \((P)\) | \(m_2\) OP1 | \(m_2\) OP2 |
|--------|-------------------|---------------------|-----------|-----------|
| initial| -                 | -                   | 0.0189    | 0.031     |
| 1      | 0.67              | 0.5                 | 0.0135    | 0.0263    |
| 2      | 0.64              | 0.5                 | 0.01396   | 0.0266    |

A secondary frequency approximately 0.5 times the rotational frequency can be also identified. This frequency components is believed to be caused due to the self rotation of the vortex rope [3].
3.2. Effect of splitter blades on draft tube flow

Table 2 summarizes the two different splitter blade geometries considered in the present study. The geometries are characterized by the ratio of the surface area of the splitter blade to the surface area of the main blade, as well as the pitch-wise position in the inter-blade passage. In figure 4 the geometries of the two variants are compared with the main runner blade. It can be seen that the main difference of the two splitter blades is that the outlet edge of case 1 lies on a meridional plane, which is not the case for case 2 geometry. In Figure 5 the dimensionless velocity distribution at monitoring location L1 for all cases considered are compared. These distributions were used to compute the flux of moment of momentum at the runner outlet as a quantitative indicator of the residual swirl associated with a particular design in a similar manner as in [11]. The calculated fluxes are shown in table 2 where it can be seen that at OP2 both splitter blade cases lead to a reduction of the residual swirl of 15.2% and 14.2%, respectively.

Figures 6 and 7 show the frequency spectrum of the pressure monitor signals for the two splitter blade cases at OP1 and OP2. At OP1, comparing the initial runner with case 1, an increase in the pressure fluctuation amplitude can be seen for all harmonics up to 6fn. This increase is a result of the increase in number of blades, which have doubled compared to the initial geometry. The dominant frequency in this case is the runner rotation frequency.

For OP2, an analysis of the pressure signals shows that for case 1 the vortex rope frequency has shifted to 0.34fn, while three additional frequency components at 0.44fn, 0.53fn, 0.66fn can be identified, which are simultaneously captured by all four monitors at sections L1 and L2. For case 2, at OP1 the pressure amplitude has further increased in the first three harmonics as a result of the slightly increased residual swirl compared to case 1 (table 2). Similar to case 1, the dominant frequency is the runner rotation frequency. At higher frequency harmonics, similar amplitudes can be observed for all three geometries. At OP2 where the vortex rope has formed, the frequency components of 0.32 fn, 0.54fn, 0.75fn are dominant for case2, with the vortex rope frequency at 0.32fn. Similar to case 1 the amplitude of pulsation has been significantly reduced as also suggested by the reduction in the flux of moment of momentum. Compared to case 1, the frequency spectrum of case 2 at OP2 shows more resemblance to the initial geometry spectrum. The above suggest that small changes in the shape of the splitter blade can affect the characteristics of the vortex rope at part load conditions.

4. Conclusions

In this paper, the effect of the splitter blade geometry on the draft tube vortex rope was studied numerically. Two different splitter blade designs were compared against the initial runner with no splitter blades at flow conditions corresponding to 0.9QBEP and 0.72QBEP. The results showed that the introduction of the splitter blades alters the frequency response in the draft tube due to a reduction of the residual swirl downstream of the runner. It was shown that small geometry modifications can
affect the behaviour of the runner at part load conditions, suggesting that an optimum design could be obtained minimising the effects of the part load vortex rope. The effect of the splitter blades at lower flow conditions in relation to geometry modifications should be further investigated in order to further quantify the effects of the splitter blades on part load performance and the velocity distribution downstream of the runner.

Figure 6. Frequency spectrum of the monitor pressure data for the case2 (a) OP1 (b) OP2.

Figure 7. Frequency spectrum of the monitor pressure data for the case2 (a) OP1 (b) OP2.

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