OVERLOAD SURGE INVESTIGATIONS AT A FRANCIS TURBINE POWER PLANT

Alexandre D’ Agostini Neto  
University of São Paulo, Escola Politécnica, Mech. Eng. Department, Av. Prof. Mello Moraes 2231, São Paulo, Brazil  
alexandre.dagostini@usp.br

Prof. Dr. Fábio Saltara  
University of São Paulo, Escola Politécnica, Mech. Eng. Department, Av. Prof. Mello Moraes 2231, São Paulo, Brazil  
fsaltara@usp.br

Abstract. Francis Turbines operating at overload condition may have their overall dynamic behavior affected either by forced or self-excited pressure pulsations in the hydraulic system. Self-excited pressure pulsations may be triggered by the existence of a typical cylindrical axisymmetric cavitating vortex rope inside the conical region of the draft tube induced by the swirling flow leaving the Francis runner. This vortex might act as a destabilizing element in the hydraulic circuit depending on its compliance and mass flow gain factors. This work presents an overload surge investigation based on prototype measurement results for a medium head Francis Turbine Power Plant. Firstly, the prototype measurement results are presented. A stability analysis based on the eigenvalues of a linearized 1D model of the hydraulic circuit as a function of the vortex rope parameters is then performed. Steady State biphasic CFD simulations are used to estimate the vortex rope properties for the same prototype operating points. Upcoming CFD simulations based on a 1D/3D coupled system should improve the numerical prediction of the phenomenon.

Keywords: Overload Surge, System Analysis, Vortex Rope, CFD

1. INTRODUCTION

Hydraulic turbines have allowable operating regimes defined by a range of net heads and mass flows. Although a Francis turbine can continuously operate inside the allowable range defined by a manufacturer, its efficiency and dynamic behavior can change considerably depending on how far an operating condition is from the optimum or design point. For a turbine installed on a simple hydraulic circuit (dam, feeding pipes or penstocks and tailrace), the head variations occur typically on a slow time scale, depending normally on the hydrological regime of a river. The water mass flow, on the other hand, can change always when the electrical system demands more or less power. This means that frequently the turbine can run to off-design conditions to keep the network stability.

Francis runners operating with water flows below or above the optimum one can produce a high amount of swirl on the draft tube inlet section once the blades angle are fixed (D’ Agostini Neto et al., 2012). At partial load condition the circumferential components of the absolute velocity have the same direction than the runner circumferential velocity. This means that the flow and runner are co-rotational. In overload condition they have opposite directions, which mean that the flow rotates in opposite direction than the runner. The different swirl levels and directions associated to different velocity profiles on the draft tube inlet section can trigger the occurrence of vortex ropes on the draft tube that can result on power unit’s shaft displacements, pressure pulsations (or surges), power oscillations and additional dynamic loading on the main turbine’s components. When the pressure on the center of the vortices achieves the vapor pressure of water they become filled with vapor in a bi-phasic flow.

Partial load vortex ropes have a helical structure and rotate around the draft tube axis with a frequency of about 25% of the turbine speed induced forced pressure oscillations (Aschenbrenner et al., 2006). Overload vortices have a cylindrical axisymmetric shape on the center of the draft tube axis at least at the upper conical section of the draft tube and under certain conditions can provide a mechanism of energy transfer from the stationary flow which leads to self-excited pressure oscillations on the overall hydraulic circuit (Koutnik and Franke, 2007). The amplitude of the oscillations is limited by non-linear characteristics of the system. Figure 1 presents the typical structures observed at the draft tube flow for partial and overload conditions, resulting from variations on the water mass flow only (same net head). The mass flow is presented on the figure as the discharge coefficient ϕ.
The exact mechanism of destabilization is not yet known, but the phenomenon can be modeled by including the vortex rope parameters on a lumped component model that considers the waterways elements as one-dimensional components. Such kind of model has very low computational costs and allows the full characterization of the waterways stability based on parametric studies of the vortex properties. The model adopted for the vortex rope description is based on the total derivatives of the vortex rope volume (Koutnik, 2006a). The following development is done according to (Koutnik and Pulpitel, 1996) and (Flemming et al., 2008).

\[
\frac{dV}{dt} = Q_2 - Q_1
\]
By combining Eqs. (1) and (2) and introducing two new parameters, one can obtain:

\[ Q_1 - Q_2 = C \frac{dH_2}{dt} + \chi \frac{dQ_2}{dt} \]  

The adopted parameters are defined as:

- A Cavity Compliance as in Eq. (4).

\[ C = -\frac{\partial \nu_C}{\partial p_C} \text{[m}^2\text{]} \]  

- A Mass Flow Gain Factor as in Eq. (5).

\[ \chi = -\frac{\partial \nu_C}{\partial Q} \text{[s]} \]  

Another assumption to be made is that inertia and friction loss effects of the gas volume are negligible, leading to

\[ H_2 = H_1 \]  

The driving parameters can be also understood as (Koutnik et al., 2006b):

- A stiffness parameter as in Eq. (7).

\[ K = \frac{1}{C} \]  

- A damping coefficient as in Eq. (8).

\[ B = \frac{\chi}{C} \]  

The stability criterion, according to (Koutnik et al., 2006b), is that the B parameter must be smaller than the dissipation damping of the hydraulic system to keep it stable. The mass flow gain factor has also been appointed as responsible for other related cavitating flow instabilities, as firstly reported by (Brennen and Braisted, 1980).

2. OVERLOAD SURGE EVENT CHARACTERIZATION

An overload surge event was experienced at a hydroelectric power plant located in Brazil during tests for evaluating the possibility of an increase in the operating range of the turbine prior to the execution of the current study (VOITH Hydro Ltda., 2015). Two medium head Francis turbines are installed at the power plant. The specific speed \( (n_q) \) of the turbines is approximately 44 where:

\[ n_q = n \frac{Q^{0.5}}{H^{0.75}} \]
Figure 3 presents a time domain signal analysis of pressure and power oscillations measured during an overload surge event. Each point presented on the figure corresponds to an operating condition achieved for a constant guide vane opening. The characteristic peak to peak value from the signals normalized by the measured net head is adopted for presenting the pressure pulsations level. The peak to peak values of the power signal normalized by the generator’s rated output is adopted for evaluating the power oscillations. The pressures were measured at the turbine’s draft tube cone at the upper water side and from the tail water side (UWS and TWS respectively) with sensors mounted flush to the water flux. For flows above 20% of the optimum the pressure starts to oscillate continuously. The highest amplitudes were achieved at 30% of the optimum flow. A small difference is observed when comparing amplitudes measured on both sides of the draft tube. The power oscillations present the same tendency than the pressure.

Figure 4 (a) presents frequency spectra of measured draft tube pressure pulsations for increasing mass flows. One can see that on the beginning of the phenomenon the dominant frequency was 50% of the rotating speed of the machine. There is a tendency of fundamental frequency reduction with the increase of the mass flow. On the fully developed phenomenon the dominant frequency drops to 25% of the rotating speed and higher harmonics become relevant on the signal. Parts (b) and (c) of Fig. 4 present the time signals for two different flow rates. No phase shift between the signals measured on upper and tail water sides was observed which indicates that a precession movement of the vortex was absent. The higher frequency components visible on the TWS signal might be related to the pressure transducer fixation point and structure.
3. HYDRAULIC SYSTEM MODELING AND STABILITY ANALYSIS

In order to evaluate the unstable system described, the hydraulic circuit of the power plant was modeled using a commercial solver (Nicolet et al., 2001) in which every hydraulic element is modeled in terms of a hydraulic resistance, impedance and capacitance. This means that every element is converted to an equivalent electrical circuit component and a system of differential equations is obtained through the application of the Kirchhoff’s law.

The first step of the evaluation is a modal analysis of the model including an equivalent vortex rope element. The model is composed by individual modular elements such as up and downstream reservoirs, turbines, cavities, surge tanks, tunnels and pipes. The cavity element represents the vortex rope and was only included on the draft tube of the operating turbine. The power unit which remained in standstill condition does not have a cavity. Figure 5 presents the modeled waterways.

Figure 4.(a) Frequency spectrum for the draft tube TWS pressure pulsations for different flow rates; Time signals for UWS and TWS pressure pulsations for (b) 123% Qopt and (c) 130% Qopt
For each pipe element of the hydraulic circuit the distributed head loss coefficients, cross sectional area and sound speed were specified based on original design data. The adopted sound speed in the water was 1000 m/s. Additionally the characteristics of the turbine were extrapolated from reduced model data and scaled for fitting the optimum point and runaway speed. The turbine element is also connected to mechanical inertias.

The system was fully initialized considering that only the power unit #1 was in operation (which corresponds to the measurements conditions) and the measured turbine output was achieved. The deviations between the calculated and the measured net head and mass flows for the power unit in operation were respectively 0,03% and 1,59%.

The system to be solved is presented on Eq. (10). The matrices A and B contain the characteristics of all the modeled elements which are based on their physical properties. The matrix C corresponds to the boundary conditions of the system. In this case, up and tailwater levels of the measured conditions were applied as boundary conditions. The vector corresponds to the system's state space containing all pressures and flows of all elements.

\[
[A] \frac{d\vec{x}}{dt} + [B]\vec{x} = \vec{c}
\]  

(10)

The stability analysis should be performed by calculating the system's complex eigenvalues. A parametric modal analysis was performed in order to obtain the vortex parameters related to unstable modes (Eq. (11)).

\[
det([I]s + [A]^{-1}[B]) = 0
\]  

(11)

Figure 6 presents the unstable eigenvalues of the modeled system. The lowest frequency identified corresponds to 0.94 times the rotational speed of the machine. Taking into account that the measured frequency was of 0.5 times the rotational speed, the deviation still needs to be clarified. Figure 7 presents the pressure mode shape corresponding to the mentioned natural mode. It becomes very clear that this mode is related to a breathing of the vapor phase cavity.
4. VORTEX ROPE CHARACTERIZATION

In order to provide more precise parameters to the stability analysis the proper vortex rope parameters must be obtained. (Koutnik and Franke, 2007) present a methodology based on a single phase 3D CFD model in order to obtain the vortex parameters. (Flemming et al., 2008) compare the vortex structures obtained through single and two-phase CFD calculations. According to the author, the two-phase CFD was the one that produced overload vortex contours that qualitatively fitted to the one observed in reduced model test rigs.

A 3D CFD model based on the finite volume method was built in order to evaluate the vortex rope structure for the studied prototype. The turbulence and governing equations of the flow have been solved using the commercial software Ansys CFX 16.1, which is a very robust solver for the hydraulic machinery industry. As the runner is indispensable to provide proper inlet conditions to the draft tube flow, in steady state calculations a 1-channel runner and tandem cascade were modeled and a mixing plane interface was used to evaluate the flux between stationary and rotating parts. In this kind of approach, circumferential averaging of the flow field properties over layers is performed to update the boundary conditions along the two zones of the interface.

The CFD calculations were run in prototype scale. Firstly, single phase calculations were performed in order to calibrate properly the prototype operating conditions. The nominal blade profile was adopted for the simulation. A cavitation model based on the Rayleigh-Plesset equations was later included on the model in order to compute properly the vortex rope in vapor phase. The actual measured tailwater level was considered as boundary condition at the turbine outlet.

On this first step no vapor phase vortex rope structure was identified at the draft tube. After extensive revision of all the simulation parameters, a geometrical modification was included at the runner blade trailing edge. This modification was performed in order to include the effect of a chamfer existent on the prototype blade in order to avoid dynamic excitation by von Karman vortices. This modification was sufficient to increase the swirl level at the runner outlet to a level in which the vortex core pressure was below the water vapor pressure and the vortex rope was then finally obtained as presented on Figure 8. The average Y+ values of the model are 3.7 at the runner blade and 67 at the draft tube walls. Another model with average Y+ close to 1 was also build and no significant variation on the runner outlet angle was observed.
Figure 8. Water-vapor interface obtained for the biphasic CFD simulations

Figure 9 presents qualitatively the pressure fields obtained for the same operating condition for mono and biphasic flow models. The dark-blue region on the right hand side of the figure corresponds to the vortex rope core region (vapor phase).

Figure 9. Pressure contours at the draft tube for the same operating point considering (a) 1 phase flow and (b) 2 phase flow

In order to further determine the vortex parameters, additional simulations with different mass flows and turbine offset levels were performed and further post-processing is still going on during the current research.

5. CONCLUSIONS

The experimental results allowed a well-done characterization of the phenomenon regarding spatial distribution, time and frequency domain analysis. The amplitude continuously increased with the turbine mass flow and the frequency, which starts from 50% of the rotating speed of the machine reduces to 25% at the last measured point.

A stability analysis showed that for certain vortex rope parameters unstable modes are present on the system. The lowest eigenfrequency identified with the model, however, is 50% above the highest measured frequency. Deviations are still being evaluated.

Single and two phase CFD simulations were performed. Incorporating the real prototype blade features, such as the trailing edge chamfer, was decisive for the success in the simulations. Vortex rope structures obtained during this phase were qualitatively similar to the ones observed in the model test.

Further post processing is still being performed in order to determine the vortex parameters.

Figure 10. Vortex rope cavities for (a) different mass flows and (b) different offset levels
6. REFERENCES


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