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EXPERIMENTAL INVESTIGATIONS INTO A FRANCIS TURBINE WITH LOW SPECIFIC SPEED

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The paper presents our experimental investigations into Francis turbine with low specific speed. First, the energetic and cavitational performances are computed based on experimental data recorded in hydropower plant. Next, the unsteady pressure is measured on the draft tube cone wall in order to identify the hydrodynamic instabilities at off-design operating conditions.

Keywords: Francis turbine, low specific speed, experimental investigations, energetic and cavitational performances, pressure fluctuations

1. Introduction

The variable demand on the energy market, as well as the limited energy storage capabilities, requires a great flexibility in operating hydraulic turbines. As a result, turbines tend to be operated over an extended range of regimes quite far from the best efficiency point. In particular, Francis turbines operated at off-design operating conditions have a high level of residual swirl at the draft tube inlet as a result of the mismatch between the swirl generated by the guide vanes and the angular momentum extracted by the turbine runner [1, 5]. Further downstream, the decelerated swirling flow in the draft tube cone often results in vortex breakdown, which is recognized now as the main cause of severe flow instabilities and pressure fluctuations experienced by hydraulic turbines operated at part load [8].

In this paper, a Francis turbine with low specific speed $n_s = 78.77$ ($\nu = 0.166$) is considered. The Francis runner geometry with low specific speed is shown in Figure 1.

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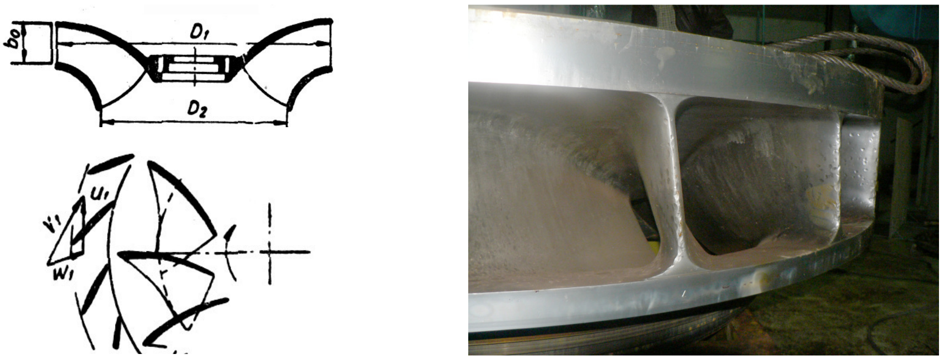

Fig. 1. The Francis runner geometry with low specific speed: sketch (left) and photo (right).

The low specific speed Francis turbine consists of 12 stay vanes and 24 guide vanes whilst the runner has 19 blades with the reference radius $D_{2e} = 1.776\,\mathrm{m}$. Figure 1 shows the Francis turbine cross view with parameters from Table 1 while the three-dimensional geometry of the Francis runner is presented in Figure 2.

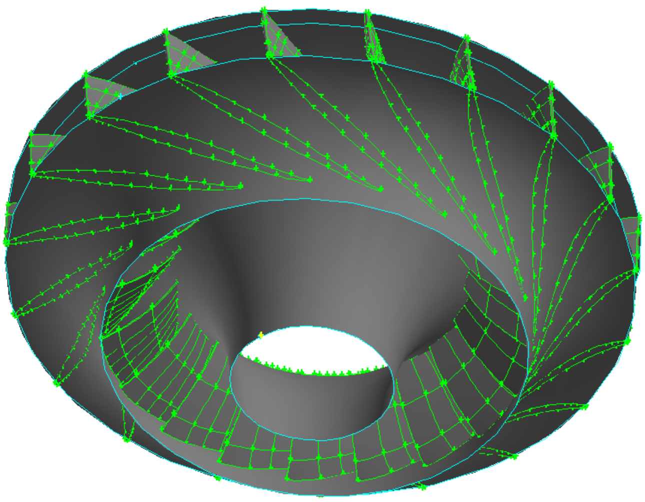

Fig. 2. The three-dimensional geometry of the Francis runner with low specific speed.

Table 1

Parameters of the Francis turbine with low specific speed.

Parameters	Value	Eqs. according to IEC [9]
characteristic speed n_s	79	$n_s = nP^{0.5}H^{-1.25}$
discharge coefficient φ	0.223	$\varphi = Q\left(\pi\omega R_{2e}^3\right)^{-1}$
energy coefficient ψ	4.027	$\psi = 2E\left(\omega R_{2e}\right)^{-2}$
hydraulic power coefficient λ	0.897	$\lambda = 2EQ\left(\pi\omega^3 R_{2e}^5\right)^{-1}$
dimensionless characteristic speed ν	0.166	$\nu = \varphi^{0.5}\psi^{-0.75}$

2. Experimental investigations into the hydropower plant

First, the equipments are installed in hydropower plant in order to record the mechanical and electrical data: head water and tail water levels as well as the static pressure upstream and downstream to the turbine in order to compute the head (H); the pressure drop on the Winter-Kennedy taps in order to compute the discharge (Q); pressures on the piston of the guide vane servomotors as well as guide vanes servomotor stroke (S_{AD}) in order to compute the guide vane opening (a_0); the generator power as well as the hydro unit power in order to compute turbine power; line voltages and phase currents at the generator and excitation voltage and currents [3]. The hill chart of the Francis turbine prototype is reconstructed (Fig. 3) based on Francis turbine model using the IEC procedure with an in-house application software [6]. Consequently, the data computed according to IEC procedure [9] are validated against experimental data measured in the hydropower plant.

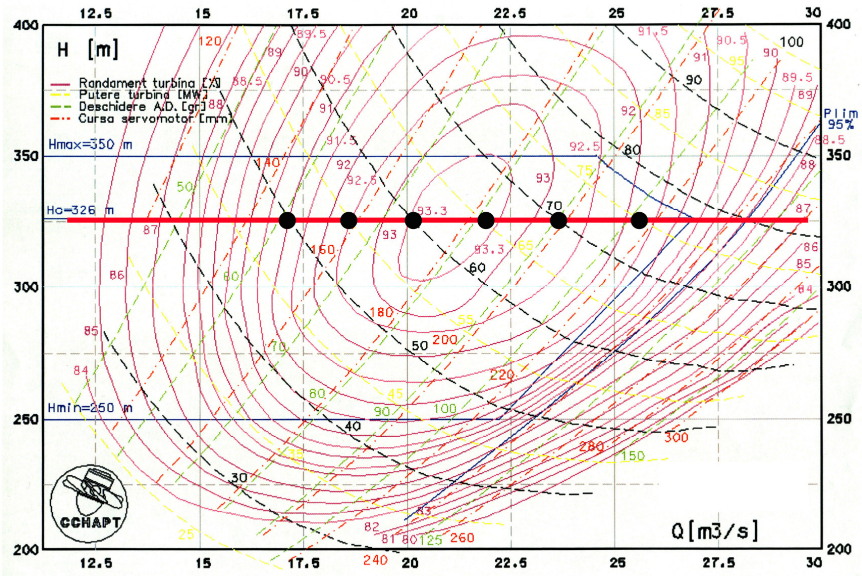

Fig. 3. Hill chart of the Francis turbine prototype with low specific speed. The unsteady pressure was measured on the cone in six points displaced on constant head.

3. Energetic and cavitational performances of the Francis turbine

The turbine hydraulic efficiency is computed according to following equation

$$\eta_T = \frac{P_T}{\rho g H Q}, \tag{1}$$

The measurement uncertainties are computed according to the Gauss law:

$$\varepsilon_\eta = \pm\sqrt{\varepsilon_H^2 + \varepsilon_Q^2 + \varepsilon_P^2}\ , \tag{2}$$

where efficiency uncertainty (ε_η) depends on the head (ε_H=±0.37%), discharge (ε_Q=±1.03%) and power (ε_P=±0.3%) uncertainties. Consequently, the efficiency uncertainty (ε_η=±1.13%) is computed according to the equation (2).

The turbine hydraulic efficiency (η_T) of the Francis turbine prototype evaluated based on the experimental data (plotted with magenta solid line in Fig. 4) is compared with the values computed with IEC procedure using the data from experimental test rig (with blue solid line). As a result, an excellent agreement is obtained certifying to the experimental procedure used in the hydropower plant. The hydraulic turbine efficiency is higher 93% at best efficiency point and above 90% on the extended region around BEP, see Figure 3.

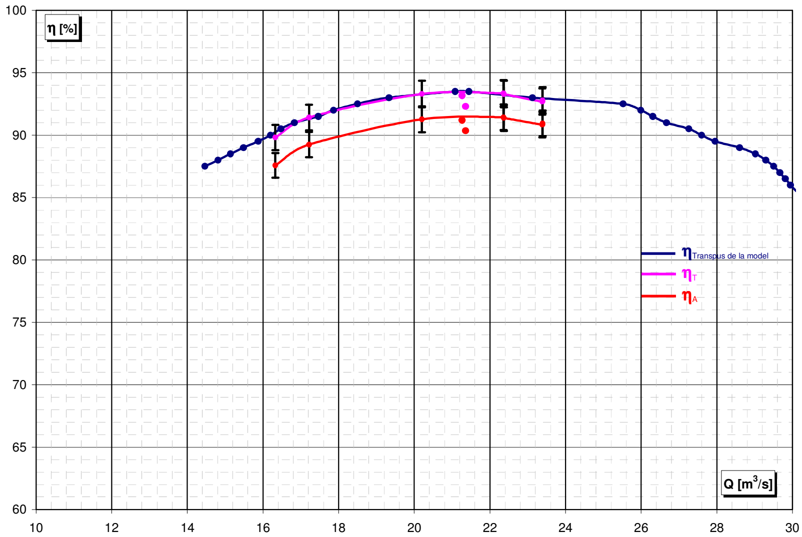

Fig. 4. The turbine hydraulic efficiency η_T computed based on the experimental data (with magenta line) and the hydro unit efficiency η_A (with red line) of the Francis turbine prototype with low specific speed at nominal constant head.

Among all the aspects of machine operation, cavitation inception and development plays a fundamental role with respect to the erosion risk [7]. Obviously, it is economically preferable to have a cavitation free operation as long as the efficiency is unaffected and the erosion is limited. This explains why the cavitation behavior problem receives greater attention in the case of hydraulic machines [2].

The cavitational performances of the Francis turbine are evaluated through the critical operation cavitation number σ_{cr} against Thoma cavitation number (so called power plant cavitation number σ_{pl}). The Thoma cavitation number σ_{pl}, defined by the IEC standards [9] as,

$$\sigma_{pl} \equiv \frac{p_{at} - p_{va} \pm \rho g H_s}{\rho g H}, \tag{3}$$

is not a quantity specific to the machine. In order to locate and to evaluate the presence of cavitation inside the turbine, one computes the *reserve cavitation number*, $\Delta\sigma$, as

$$\Delta\sigma = \sigma_{pl} - \sigma_{cr}, \tag{4}$$

where σ_{cr} is the *critical operation cavitation number* computed .

Cavitation development occurs in all the zones where the local static pressure is equal or less than the vapour pressure. Therefore, the rezerve cavitation number allows one to detect the incipient cavitation points as well as the cavitation and supercavitation regimes, as follows

$\Delta\sigma > 0$	$\sigma_{cr} < \sigma_{pl}$	without cavitation
$\Delta\sigma = 0$	$\sigma_{cr} = \sigma_{pl}$	cavitation inception
$\Delta\sigma < 0$	$\sigma_{cr} > \sigma_{pl}$	cavitation
$\Delta\sigma << 0$	$\sigma_{cr} >> \sigma_{pl}$	super-cavitation

Equation (3) introduces σ_{cr} which is a coefficient specific to the turbine and it is computed based on experimental investigations on the test rig. In doing so, we separate the turbine cavitation analysis from the plant cavitation evaluation. One can see that $\sigma_{cr} = \sigma_{pl}$ only at cavitation inception, otherwise there are two completely different quantities. The power plant cavitation number σ_{pl} is computed based on the experimental data using equation (3). In this case, it is used the following formula

$$H_s = z_R - z_e, \tag{5}$$

where z_R is the reference level of the Francis turbine relative to the Black Sea level and z_e the level of the tailing water. The power plant cavitation number σ_{pl} and critical operation cavitation number σ_{cr} for nominal constant head are plotted in Figure 5 and σ_{pl} values are shown in Table 2.

Table 2

Power plant cavitation number σ_{pl} at nominal constant head.

n_{11} [rot/min]	Q_{11} [m^3/s]	Z_e [m]	H_s [m]	σ_{pl}
61.76	0.13384	267.130	-3.130	0.040511
61.76	0.14116	267.130	-3.130	0.040511
61.76	0.15229	267.210	-3.210	0.040755
61.76	0.16562	267.220	-3.220	0.040786
61.76	0.17434	267.288	-3.288	0.040995
61.76	0.18331	267.288	-3.288	0.040995
61.76	0.19173	267.288	-3.288	0.040995
61.76	0.20050	267.288	-3.288	0.040995

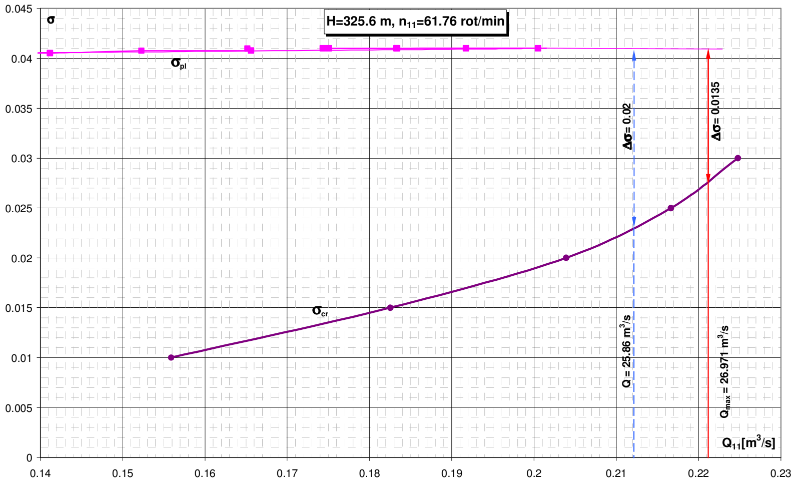

Fig. 5. Thoma cavitation coefficient σ_{pl} (with magenta solid line) and turbine cavitation coefficient σ_{cr} (with brown solid line) of the Francis turbine prototype with low specific speed at nominal constant head.

Based on the criterion defined above $\Delta\sigma > 0$, the Francis turbines operates at nominal constant head without cavitation, see Figure 5. However, the Francis runner cavitation sensitivity is increased near to the maximum discharge.

3. Unsteady pressure measurement in situ on the draft tube cone

Operating Francis turbines for all discharge is often hindered by the development of the helical vortex (so-called vortex rope) downstream the runner, in the draft tube cone. The unsteady pressure field induced by the precessing vortex rope may also lead to hydro- acoustic resonance.

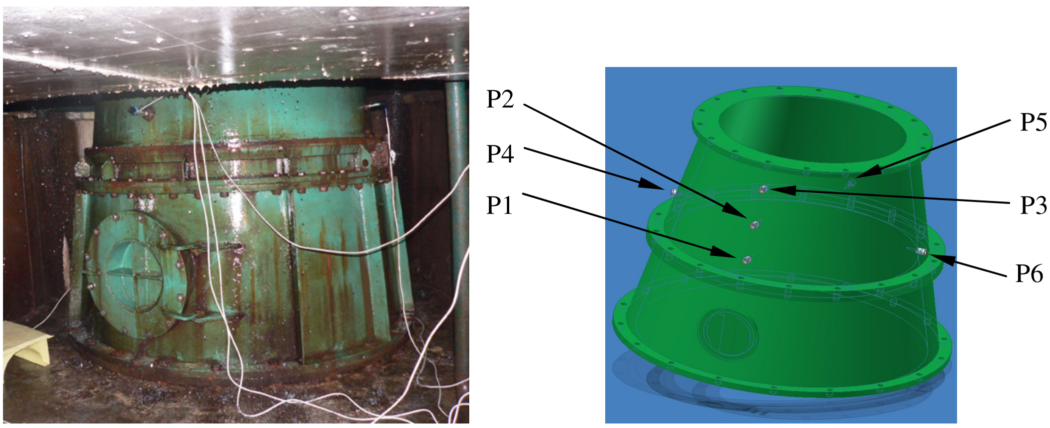

Fig. 6. Photos of the unsteady pressure transducers installed on the cone of Francis turbine prototype with low specific speed (left) and 3D view of the unsteady pressure transducers located on the cone of the draft tube (right).

Three unsteady pressure transducers are displaced along to the element of cone (P1, P2 and P3) while four unsteady transducers are mounted at the same radius (P1, P4, P5 and P6) at the middle section of the cone, see Figure 6, in order to evaluate the pressure fluctuation associated to the vortex rope. The unsteady pressure is recorded into all six transducers when Francis turbine operates with and without air admission. The Fourier spectra are obtained from unsteady pressure signals. In order to analyze the pressure fluctuations it is selected the pressure transducer P3 near to the Francis turbine throat based on our previous investigations [5]. The Fourier spectra are plotted in Fig. 7 without and with air admission when Francis turbine operates at 77% partial discharge. The dimensionless values are marked with ^ and defined according to:

$$A\wedge = A/(\rho g H)\ [-] \tag{6}$$

$$f\wedge = f / f_R\ [-] \tag{7}$$

where A is the amplitude of the pressure fluctuations, H head, ρ water density, g gravity and f the runner frequency, respectively. In this case, it is not distinguish the pressure fluctuations associated to the vortex rope without or with air admission. As a result, the vortex rope is not developed enough at 77% partial discharge in order to indentify the pressure fluctuations associated its. However, the Fourier spectrum for air admission is clearer than without air admission, Figure 7. The same conclusion can be drawn for all operating points investigated and marked with black spot in Figure 3.

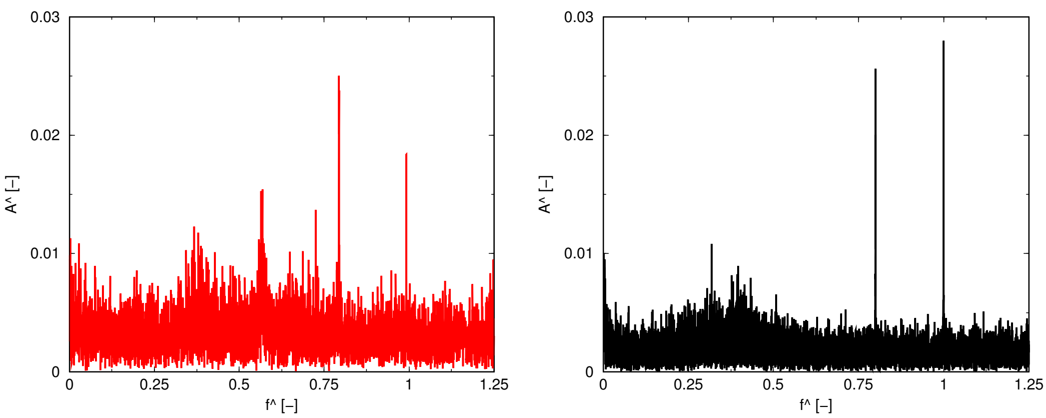

Fig. 7. Fourier spectra of the unsteady pressure transducer P3 installed near to the throat on the cone of Francis turbine prototype operating at 77% partial discharge: without air admission (left) and with air admission (right).

6. Conclusions

The experimental investigations into a Francis turbine with low specific speed are performed in order to evaluate the energetic and cavitational performances. The Francis turbine presents good energetic and cavitational performances when operates at nominal constant head and variable discharge. Moreover, the experimental investigations of the unsteady pressure on draft tube cone has no identified pressure fluctuations associated to the vortex rope when Francis turbine operates at nominal head and variable discharge.

7. Acknowledgements

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