Assessment of Power Swings in Hydropower Plants through High-Order Modelling and Eigenanalysis

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***Abstract* – Power plants are subject to introduce disturbances in the power grid, resulting from interactions with the dynamical behavior of the energy source subsystem. In the case of hydropower plants when used to compensate for variations of power generation and consumption, instabilities or undesirable disturbances may arise. They may be caused by phenomena such as part load vortex rope pulsations in the draft tube of Francis turbines. This may affect the dynamical behavior of the power plant and lead to troublesome interactions with the grid. This paper presents a case study of an existing hydropower plant that illustrates the effects of pressure pulsations due to vortex rope precession on the draft tube of Francis turbines. It also showcases possible solutions to the mitigation of the effects of this disturbing hydraulic phenomenon over the operation of the generators and electrical system. The investigated system is a 1 GW hydropower plant (4 × 250 MW units). The assessment of the power swings is performed through modal analysis combined with frequency-domain and time-domain simulations, which are then compared with on-site measurements.**

***Index Terms* – Eigenvalues and eigenfunctions, Modal analysis, Power system stability, Power system dynamics, Generators, Hydroelectric power generation.**

I. Introduction

POWER system stability has been a topic of major concern for almost one hundred years [1], [2]. Ever since then, a number of forms of instability have come forth as a consequence of the growth of power networks. They can be conveniently divided into several classes according to distinct criteria, as presented in [3], [4]. Besides the stability issues described in [4], other types of disturbances may be introduced in the power grid. They may be originated by the intrinsic dynamical characteristics of the primary source of energy of power plants, such as wind and solar PV, that do not have guaranteed availability and have high volatility. Therefore, in order to keep the stability of the power grid, it is opportune to take profit of the intrinsic flexibility of hydropower plants, which are capable of withstanding rapid set-point variations of active and reactive power [5].

Nevertheless, operating hydropower plants to counterbalance constant variations of electricity generation and consumption leads to off-design operation. This may provoke instabilities or adverse oscillations, as pulsations originated in the hydraulic system may propagate in the electrical system, deteriorating the dynamical behavior of the power plant. This may also lead to troublesome interactions with the grid, whose prediction depends on an adequate modeling of both electrical and hydraulic elements. Results obtained with a comprehensive representation give more precise information about the stability of the system, from hydraulic and electrical viewpoints [6], [7].

At part load operation of Francis turbines, when the flow rate ranges from 50% to 85% of the flow rate at the best efficiency point, there exists a swirly pattern in the water flow leaving the turbine due to a circumferential component on its velocity. As a result, a helical vortex rope is induced in the draft tube, which has a precession frequency most commonly between 0.2 and 0.4 times the turbine rotational speed [8], [9]. This characterizes periodic pressure pulsations in the draft tube that act as an excitation source for the whole system. Chances exist that the system response to this excitation is amplified, possibly resulting in intense pressure surges and mechanical power fluctuations. Such fluctuations in their turn may interact with the power network, eventually resulting in considerable electrical power swings [10], [11].

Indeed, the frequency of part load vortex rope pulsations may resonate with inter-area (0.1-0.7 Hz), intra-plant (or inter-machines) and local modes frequency (0.7-2.0 Hz) [5], [12]. Depending on the stiffness of the grid, the electrical power swings resulting from this resonance may be detrimental to the dynamical behavior of the generating unit and to the stability of the local network. This is particularly the case for power plants operating in islanded networks.

This paper presents a case study that illustrates the effects of pressure pulsations due to vortex rope precession on the draft tube of Francis turbines. It also showcases possible solutions to the mitigation of the effects of this disturbing hydraulic phenomenon over the operation of the generators and electrical system. The investigated system is an existing 1 GW hydropower plant which was previously analyzed in [13]. This work proposed a methodology for the assessment of part load resonance risk based on time-domain simulation. Here, eigenanalysis (modal analysis) is the main approach, which is then combined with frequency-domain and time-domain simulations. Also, comparisons are performed with on-site measurements. Furthermore, the electromechanical modes are identified not only for one single generator, but with all four generators connected to the grid. Another important aspect is that the grid is not considered here as an infinite bus. This allows for a more precise representation of the dynamical behavior of the electrical system [5].

II. Description of the Hydropower Plant

A. Characteristics of the Hydro Scheme

The hydropower plant studied in this paper is composed of

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4 × 250 MW generating units. The layout of the power plant is presented in Fig. 1.
The electrical subsystem is constituted of 4 × 281.5 MVA synchronous generators connected to four corresponding 500 kV/18 kV *Yd5* step-up transformers. The ABB UNITROL® excitation system is applied as voltage regulator and power system stabilizers of type PSS2B are available. The four units are connected to the power grid through two parallel transmission lines, represented by *RL* elements.

The hydraulic installation comprises an 1485 meter long gallery, a surge tank with variable cross section, an 1396.5 meter long penstock and a manifold feeding 4 × 250 MW Francis turbines. Table I gives the parameters of the system. A specific constant value is calibrated for the draft tube wave speed during vortex rope in order to match simulations and calculations to on-site measurements ($a = 56$ m/s). In practice, the determination of the wave speed is an intricate task that requires specific knowledge which is out of the scope of the present work [14]. In order to facilitate the identification of the hydraulic eigenmodes, the turbine governor is kept off-line in the model.

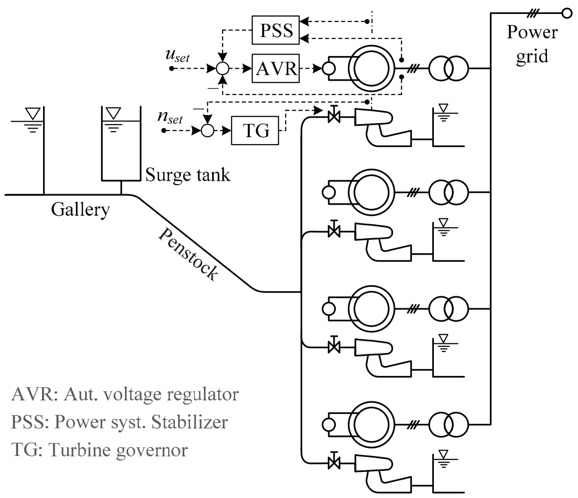

Fig. 1. Layout of the hydroelectric power plant [13].

B. Issues Encountered During Operation

During the operation at part load condition (0.43 p.u. of active power), undesirable electrical power swings were detected with a frequency $f = 1.34$ Hz. It was assumed that they were related to interactions of the electrical system with possible draft tube vortex rope pulsations.

For this turbine (which has $n_N = 333.3$ rpm), the frequency range of part load vortex rope precession is 1.11 to 2.22 Hz. This matches the typical range of intra-plant and local modes (0.7 to 2.0 Hz). Therefore, such interaction is likely to occur. It depends, however, on the operating point of the generators and on the topology of the electrical system, since these two factors influence the frequency and damping of the electromechanical modes. The power swing encountered during the commissioning was solved by using standard measures, see V. The present study, conducted after the measures implementation, aims at a better understanding of the root cause of the power swing and defining ways to anticipate such undesirable oscillations.

TABLE I
SYSTEM PARAMETERS

Synchronous generators	
Rated power	$S_N = 281.5$ MVA
Rated voltage	$U_N = 18$ kV
Frequency	$f_N = 50$ Hz
Number of pairs of poles	$P_p = 9$
Inertia of the generator rotor	$J_g = 2.25 \cdot 10^6$ kgm^2
Step-up transformers *Yd5*	
Rated power	$S_N = 281.5$ MVA
Rated primary voltage	$U_{1N} = 500$ kV
Rated secondary voltage	$U_{2N} = 18$ kV
Transmission lines	
Resistance	$R_{line} = 2.8\ \Omega$
Inductance	$L_{line} = 120$ mH
Power grid	
Rated voltage	$U_N = 500$ kV
Short-circuit power	$S_{sc} = 1750$ MVA
Gallery	
Length	$L = 1485$ m
Diameter	$D = 9.2$ m
Wave speed	$a = 1200$ m/s
Surge tank	
Mid tank cross section	$A_{ST} = 133$ m^2
Penstock	
Length	$L = 1396.5$ m
Diameter	$D = 8.8 / 7.15$ m
Wave speed	$a = 1200$ m/s
Draft tubes	
Length	$L = 25$ m
Wave speed (with cavitation)	$a = 56$ m/s
Francis turbines	
Rated power	$P_N = 250$ MW
Rotational speed	$n_N = 333.3$ rpm
Rated head	$H_N = 352$ m
Specific speed	$\nu = 0.22$
Inertia	$J_t = 1.71 \cdot 10^5$ kgm^2

III. THEORETICAL APPROACH

A. Methodology of the Study

In order to investigate the problems detected on-site, a comprehensive, precise model of the power plant is necessary. Therefore, a high-order representation of the system must be established, comprising models for all the hydraulic, electrical and regulation elements. These models are briefly described in the following paragraphs of the present section.

Then, a comparison is performed between on-site measurements (taken during commissioning phase), time-domain simulations and eigenvalues calculation, in order to confirm the value of the short-circuit power of the grid. This validates the complete model and is presented in section IV.

Finally, the analysis of the system is carried out under part load condition. The assessment is divided into three steps: (i) Eigenanalysis of the electrical system, in order to identify electromechanical modes likely to interact with pulsations originated in the hydraulic circuit; (ii) Eigenanalysis of the hydraulic system combined with forced response (harmonic response), in order to detect possibly lowly damped eigenmodes of the draft tube; (iii) Eigenanalysis of the complete hydroelectric installation combined with time-domain simulations, which confirms the interaction between hydraulic and electrical elements. All simulations are performed using the SIMSEN software package [15], which has complete hydroelectric simulation capabilities.

B. Models of Electrical Components

In SIMSEN, all electrical elements are described based on *a,b,c* phase variables, taking into account transient behavior on the stator side. Therefore, six differential equations are used to describe the transient behavior of the synchronous machine: three for the stator phases, one for the field winding and two others for the damper windings.

The three-phase power transformer model takes into account the resistance and the leakage inductance of the windings, as well as the magnetizing inductance. For the *Yd5* vector group, the model is described by a set of seven differential equations: one for each phase winding, plus one for the circulating current in the delta-connected windings.

In this study, transmission lines are represented by series *RL* elements, each described by three differential equations.

The small-signal models used for the eigenvalues calculations are analytically derived from the time-domain models. They are also described based on an *a,b,c* phase variables. The derivation of such small-signal models require a different approach which was presented in details in [16].

The exhaustive description of time-domain and small-signal models for electrical elements present in SIMSEN can be found, respectively, in [17] and [5].

C. Model of the Automatic Voltage Regulators

The voltage regulators installed in the power plant studied in this paper is the ABB UNITROL®. Its transfer function is depicted in Fig. 2, where u_c is the set-point, u_m is the terminal voltage of the generator and u_{reg} is the output.

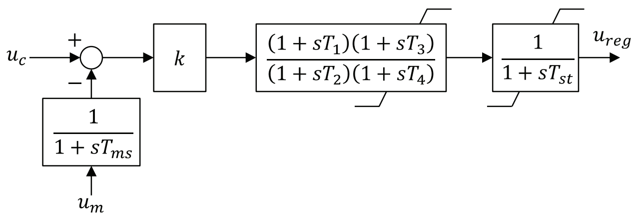

Fig. 2. Transfer function of the ABB UNITROL® voltage regulator.

D. Model of the Power System Stabilizers

The power system stabilizers of the power plant are of type IEEE PSS2B. This is a two-input PSS (speed deviation and electrical active power), whose output signal is calculated based on the estimated integral of the accelerating power. Further information about this PSS can be found in [18], [19].

E. Models of Hydraulic Components

The hydraulic components are modelled with an equivalent electric circuit, where pressure is analogue to voltage and discharge is analogue to current. That is possible because the momentum and continuity equations, which describe the motion of water, can be simplified to an equivalent form of a transmission line model [6], [7].

In the case of the model of a pipe, one can assume uniform pressure and velocity distributions in its cross section and neglect convective terms [20]. The one-dimensional momentum and continuity balance for an elementary pipe filled with water of length *dx*, cross section *A*, and wave speed *a*, can be thus represented as a T-shaped equivalent *RLC* circuit [21]. In such representation, the hydraulic resistance *R*, inductance *L*, and capacitance *C* correspond respectively to energy losses, inertia and storage effects. These parameters are given by:

$$R = \frac{\lambda |Q| dx}{2gDA^2} \qquad L = \frac{dx}{gA} \qquad C = \frac{gAdx}{a^2} \tag{1}$$

Where λ is the friction factor, *Q* is the discharge, *dx* is the length of the pipe segment, *g* is the gravitational acceleration, *D* is the diameter of the pipe, *A* is the cross section and *a* is the wave speed on the pipe.

The model of a pipe of total length *L* is made of a series of *N* elements of length $dx = L/N$, each one represented by the T-shaped equivalent *RLC* circuit. For the model of the hydropower plant in question, the penstock is discretized in 116 elements, whereas the draft tube is represented by two pipe elements. The discretization of the whole piping system respects the CFL condition, which states that the Courant number, defined as $C_r = a \cdot dt/dx$, must be $C_r \leq 1$ [20].

Francis turbines are modelled by a pressure source converting hydraulic energy into mechanical work, in series with an inductance L_t related to the inertia effects of the water and a resistance R_t, which ensures zero discharge when the guide vanes are fully closed. The pressure source $H_t = H_t(Q_t, \omega, y)$ depends on the turbine characteristic curves, which are nonlinear functions of the turbine discharge Q_t, the rotational speed ω and the guide vane opening *y* [7].

The model used in section V takes into account the turbine characteristic curves. Therefore, this high order model takes into account effects of water hammer, mass oscillation, and transient behavior of the turbine in the four quadrants, linked to the corresponding rotating inertia.

Further details concerning time-domain and small-signal models for hydraulic elements present in SIMSEN can be found, respectively, in [7] and [8].

IV. Practical Validation of the Model

In order to validate the parameters of the electrical model, time-domain simulations are compared to on-site measurements, which were performed in order to validate the tuning of both voltage regulator and PSS. During these tests, only one machine was synchronized to the grid. The disturbance applied to the system is a ±2% step on the set-point of the voltage regulator. The reaction of the system is observed on the active (P_{el}) and reactive (Q) powers, as well as on the excitation voltage (u_f). The comparison between measurements and simulation is presented in Fig. 3.

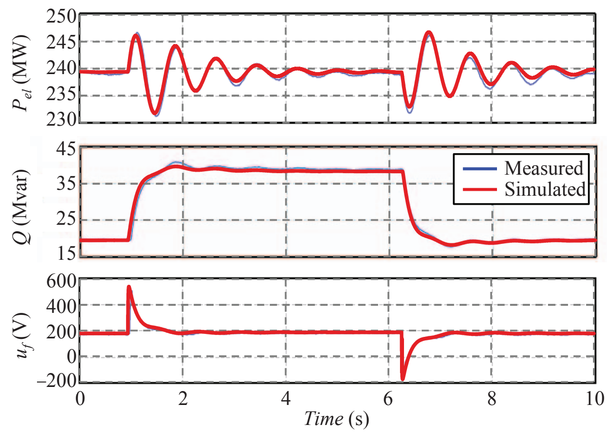

Fig. 3. On-site measurements versus time-domain simulation.

The very good agreement that can be observed in this comparison validates the model, including the short-circuit power of the grid (S_{sc} = 1750 MVA). It is easy to observe the action of the local mode oscillations, which are clearly dominant in the dynamical behavior of the active power. Due to the relatively low short-circuit power of the grid, these oscillations take some time to be damped.

In order to verify the characteristics of the local mode, the calculation of eigenvalues for this system is carried out, considering the same operating point of measurements: $u = 0.97$ p.u., $p_{el} = 0.85$ p.u. and $q = 0.02$ p.u.

The calculation yields thirty-one eigenvalues of the form $\lambda_i = \sigma_i \pm j\omega_i$, that describe the small-signal response of the system. This operating point is indeed stable (all $\sigma_i < 0$) and most of the eigenvalues are very well damped. Amongst these eigenvalues, the dominant one is the local mode:

$$\lambda_{lm} = -0.90 \pm j7.96$$

The related damping time constant $\tau = |\sigma|^{-1}$ is 1.11 seconds and the oscillation frequency $f = \omega/2\pi$ is 1.27 Hz.

The active power behavior after the first disturbance is presented again in Fig. 4. It is possible to graphically extract from it the oscillating frequency f, the attenuation σ and the damping time constant τ. After the first peak, once the influence of rapidly damped modes is past, the global transient behavior is satisfactorily described by the frequency f, the attenuation σ and the damping time constant τ extracted from this graph. The good match with the eigenvalues result shows that a very good agreement exists between measurements, time-domain and small-signal models (the highest deviation value is 3.6%). Therefore, both models can be considered validated.

V. Stability Assessment under Part Load Operation

A. Modal Analysis of the Electrical Installation

In order to perform an analysis focused on the understanding of the interactions between the vortex rope phenomenon and the electrical system, the four units are considered at the operating point for which the issues described in section II took place. That is: $u = 0.97$ p.u., $p_{el} = 0.426$ p.u. and $q = 0.345$ p.u.

The order of the electrical system (with PSS) is 124. The eigenanalysis performed here, however, aims at revealing the electromechanical modes of the system, since they are those likely to interact with vortex rope pulsations. Furthermore, the high value of inertia of the turbine-generator set acts as a low-pass filter in such a way that phenomena over the range of a few hertz on the hydraulic side do not affect the electrical one, and vice-versa [16]. Thus, only results related electromechanical modes are presented. Table II summarizes the results, indicating the eigenvalues (λ) and the corresponding damping ratios ($\zeta = \sigma/\omega_o$) and frequencies (f).

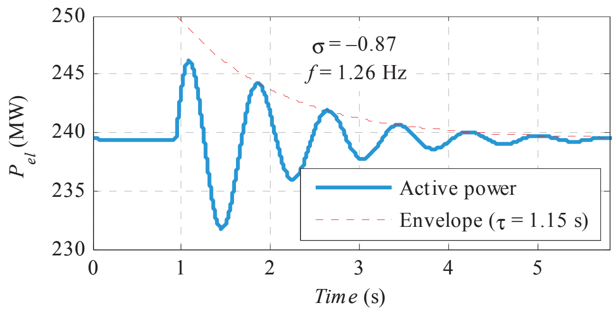

Fig. 4. Frequency and damping time constant from time-domain results.

TABLE II
ELECTROMECHANICAL MODES OF THE ELECTRICAL INSTALLATION

Eigenmode	Without PSS			With PSS		
	Λ	ζ (%)	f (Hz)	λ	ζ (%)	f (Hz)
Local	$-0.4 \pm j5.35$	7.5	0.85	$-1.6 \pm j4.67$	32.4	0.74
Intra-plant 1	$-1.9 \pm j8.41$	22.0	1.34	$-7.3 \pm j8.57$	64.8	1.36
Intra-plant 2	$-1.9 \pm j8.41$	22.0	1.34	$-7.3 \pm j8.57$	64.8	1.36
Intra-plant 3	$-1.9 \pm j8.41$	22.0	1.34	$-7.3 \pm j8.57$	64.8	1.36

Besides the local mode, one can observe the presence of three intra-plant (inter-machine) modes. Whereas the local mode represents the oscillations of the whole power plant against the grid (the rotors of all machines oscillate with the same frequency and same phase), the intra-plant modes represent oscillations that happen mainly inside the power plant, with the machines swinging against the other.

Although all three have the same value, the dynamical behavior of the generators is different for each one. That is, they oscillate with the same frequency for all intra-plant modes, but the phase shift between the machines is different for each one of these modes.

The strong influence of the PSS on damping the electromechanical modes is clear. It must be noticed, however, that the frequencies of the intra-plant modes, with and without PSS, are very close to the frequency of the power swings recorded on-site (1.34 Hz). This means that interactions may occur between the electrical and the hydraulic systems around this value of frequency. Therefore, a small-signal stability analysis of the hydraulic circuit must be performed in order to identify the eigenmodes that are most likely to interact with the generators and the grid.

B. Modal Analysis and Forced-Response of the Hydraulic Installation

In order to perform the modal analysis of the hydraulic installation, the same operating point must be considered. From the turbine point of view, it corresponds to a mechanical power $p_{turb} = 0.48$ p.u. (120 MW).

The order of the complete hydraulic system is 381. Nonetheless, the hydraulic eigenmodes most likely to interact with the electro-mechanical ones are the low frequency elastic modes from the penstock and draft tube. Therefore, only these eigenmodes are given in Table III.

TABLE III
MAIN EIGENMODES OF THE HYDRAULIC INSTALLATION

Eigenmode	λ	ζ (%)	f (Hz)
Penstock 1st elastic mode	$-1.02 \pm j2.16$	42.7	0.34
Penstock 2nd elastic mode	$-0.68 \pm j4.41$	15.2	0.70
Penstock 3rd elastic mode	$-0.52 \pm j6.49$	8.00	1.03
Draft tube 1st elastic common mode	$-0.02 \pm j3.50$	0.57	0.56
Draft tube 1st elastic inter-machines mode	$-0.07 \pm j3.50$	2.00	0.56
Draft tube 2nd elastic common mode	$-0.005 \pm j8.38$	0.06	1.33
Draft tube 2nd elastic inter-machines mode	$-0.01 \pm j8.39$	0.12	1.34

As it can be seen, the penstock elastic modes are better damped than draft tube ones. Moreover, their frequencies do not coincide with the electro-mechanical ones. Thus, the penstock modes are not likely to affect the behavior of the electrical system at this specific operating point.

On the other hand, the draft tube modes are indeed very relevant in this case. Both the 1st and 2nd modes appear four times in the results and they are poorly damped. For each of them, in one occurrence the damping ratio is weaker. These less damped modes have the same phases in all four units. Oscillations due to such modes add up and their influence goes upstream through the penstock, surge tank and gallery. Due to their common nature for all four units, these modes are called here *draft tube 1st and 2nd elastic common modes*.

The other three occurrences of each of the two draft tube elastic modes have a distinct characteristic. They indicate inter-machine oscillations, since their actions in the four machines happen in phase opposition. The superposed contributions of the four machines cancel each other in the penstock. Thus, they do not have any upstream influence. In view of the phase opposition of these modes, they are called here *draft tube 1st and 2nd elastic inter-machine modes*. The mode shapes of the 2nd common and inter-machine modes are depicted in Fig. 5.

The influence of these two modes on the whole hydraulic system can be confirmed through a forced response analysis, which is performed using the linearized matrices of the model. Fig. 6 depicts the forced response of rotor speed, draft tube and turbine inlet pressures to an excitation source located in the draft tube of generating unit 1. It is easy to observe that the strongest resonances happen at the frequencies of the draft tube elastic modes, and their influences are extended to the rotational speed. Therefore, oscillations at such frequencies are likely to spread to the electrical system through the mechanical torque behavior.

It is important to stress that the frequency of the 2nd draft tube elastic modes lies in the range in which vortex rope pulsations are likely to occur (1.1 to 2.2 Hz in this case). Hence, a risk of resonance in the draft tube exists. Moreover, it is very close to the frequency of the electrical intra-plant eigenmodes and it corresponds to the frequency of the power swing recorded on-site. This indicates that significant

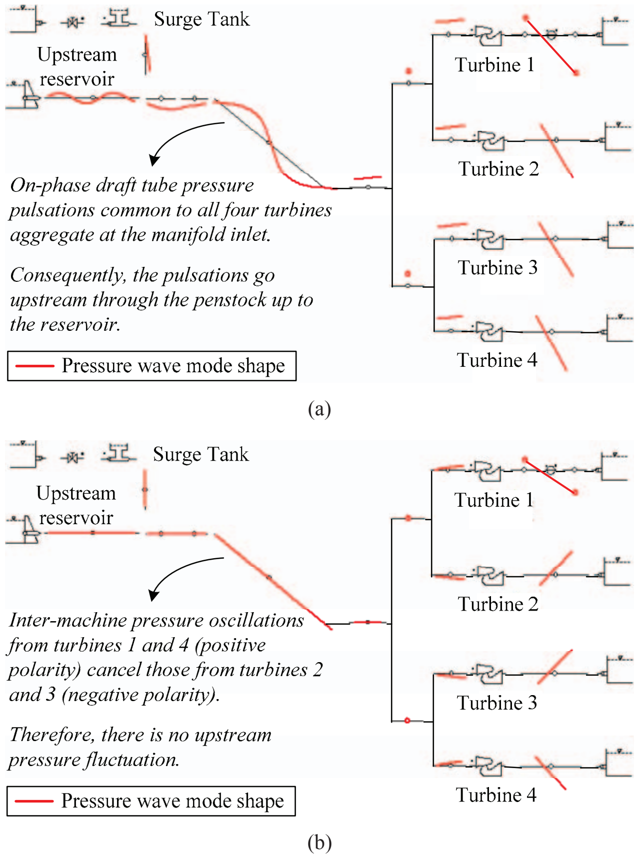

Fig. 5. Draft tube 2nd elastic common (a) and inter-machines (b) modes.

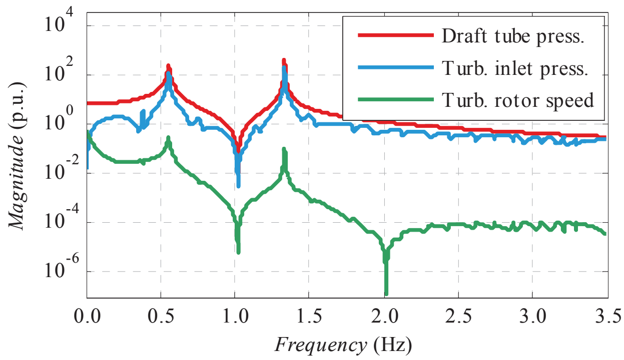

Fig. 6. Forced response of the hydraulic system.

interactions may happen between hydraulic, mechanical and electrical subsystem at least around this frequency value.

C. Stability Assessment of the Complete Hydroelectric Site

The complete hydroelectric model is obtained by combining the electrical and hydraulic models studied previously. The interface between these two systems lies in the mechanical coupling between turbine and generator inertias, through coupling shaft. Thus, interactions between both subsystems are related to interactions between the mechanical and the electromagnetic torques.

The order of the complete system (with PSS) is 493. Table IV summarizes the most relevant eigenvalues.

It can be observed that the connection of both subsystems causes slight changes in the eigenvalues. The influence of the

TABLE IV
MOST RELEVANT MODES OF COMPLETE HYDROELECTRIC SITE

Eigen mode	Without PSS			With PSS		
	λ	ζ (%)	f (Hz)	λ	ζ (%)	f (Hz)
A	$-0.98 \pm j2.09$	42.5	0.33	$-0.97 \pm j2.10$	41.9	0.33
B	$-0.63 \pm j4.39$	14.2	0.70	$-0.66 \pm j4.33$	15.1	0.69
C	$-0.54 \pm j6.50$	8.28	1.03	$-0.54 \pm j6.48$	8.30	1.03
D	$-0.02 \pm j3.50$	0.57	0.56	$-0.02 \pm j3.50$	0.57	0.56
E	$-0.07 \pm j3.50$	2.00	0.56	$-0.07 \pm j3.50$	2.00	0.56
F	$-0.005 \pm j8.38$	0.06	1.33	$-0.005 \pm j8.38$	0.06	1.33
G	$-0.01 \pm j8.39$	0.12	1.34	$-0.01 \pm j8.39$	0.12	1.34
H	$-0.43 \pm j5.44$	7.88	0.87	$-1.59 \pm j4.83$	31.3	0.77
I	$-1.92 \pm j8.53$	22.0	1.36	$-7.20 \pm j8.34$	65.3	1.33

Eigenmodes legend:
A Penstock 1st elastic mode
B Penstock 2nd elastic mode
C Penstock 3rd elastic mode
D Draft tube 1st elastic common mode
E Draft tube 1st elastic inter-machine modes 1, 2 and 3
F Draft tube 2nd elastic common mode
G Draft tube 2nd elastic inter-machine modes 1, 2 and 3
H Electromechanical local mode
I Electromechanical intra-plant modes 1, 2 and 3

PSS on the hydraulic modes is not significant, and the draft tube modes remain very poorly damped. Moreover, the draft tube 2nd modes and the electromechanical intra-plant modes do not undergo any frequency shift, and their frequencies still match. Hence, the risk of resonance is confirmed.

Although the intra-plant modes are strongly damped, resonances in the draft tube are amplified by turbine head and mechanical torque fluctuations. Consequently, the electromagnetic torque will also pulsate, resulting in significant electrical power swing. As vortex rope precession may act as a sustained excitation source, the resulting power swing is also persistent. Still, the high damping ratio is important to limit the amplitude of these oscillations.

Once the risk of resonance is identified and the frequency is known, a targeted time-domain simulation is performed in order to assess the magnitude of the power swing in case vortex precession frequency would match the system natural frequency. Therefore, a pressure excitation source placed in the draft tube of unit 1 with frequency equal to 1.33 Hz and amplitude equal to 2% (4% peak-to-peak) of the rated turbine head (H_N). This is a plausible relative amplitude of vortex rope pulsation. Fig. 7 shows that the peak-to-peak electrical power fluctuations reach 62% of the rated power of the generator (with PSS off-line). When the PSS is active, the peak-to-peak value drops to 31%. Such oscillations can be harmful not only to the power plant, but also to the local grid, and would not be admissible. Even if the on-site situation was not as bad as the hypothetical one depicted above, the latter clearly shows that this power plant is very sensitive to part load vortex rope phenomena, which must be avoided or attenuated by all means.

Some measures permit to mitigate vortex rope precession: compressed air injection in the draft tube modifies the local wave speed, which results in changing the characteristics of the draft tube eigenmodes. Fins installed in the draft tube cone introduce flow modifications that shift the precession frequency of the vortex rope. Another possibility is to

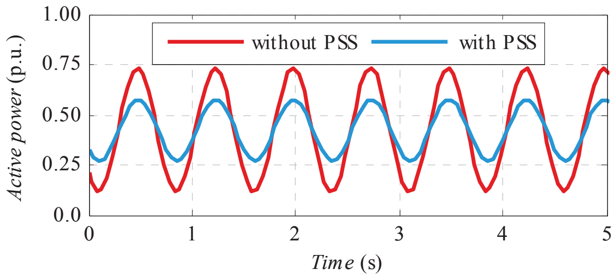

Fig. 7. Turbine net head, mechanical torque (a) and active power (b) response to vortex rope pulsation.

especially tune the PSS parameters or to condition the PSS power input signal with a filtered draft tube pressure signal in order to avoid the disturbances [5], [22]. After suitable measures were implemented on-site, it was possible to reduce the peak-to-peak amplitude of power swings to 1.1% of the rated power. These measures were thus satisfactory, as a tolerable level of power swing was obtained.

Finally, further investigations (considering the original design of the site, without the corrective measures) revealed that the 1.1% peak-to-peak power swing recorded on-site could be caused by a very weak excitation on the draft tube. With active PSS, the amplitude of the excitation necessary to reproduce the phenomenon was equal to 0.021% of the rated turbine head. This shows that very few energy is necessary to cause electrical power swings in this hydropower plant. This means that the broadband excitation spectrum in the draft tube would be sufficient to cause undesired behavior with significant power fluctuation under part load operation.

Fig. 8 presents the electrical power fluctuations caused by this attenuated excitation source. Unit 1 (location of the excitation) oscillates with a peak-to-peak amplitude equal to 1.11% of the rated power. The other three units have a peak-to-peak oscillation equal to 0.4%. Moreover, they oscillate in phase opposition with respect to unit 1, which means that an electromechanical intra-plant mode is excited.

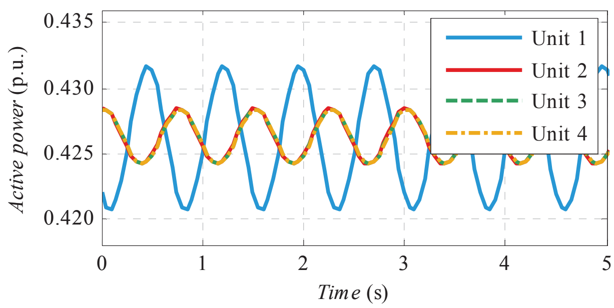

Fig. 8. Electrical power swing due to attenuated vortex rope pulsations.

VI. CONCLUSIONS

This paper presented a case study illustrating the effects of pressure pulsations due to part load vortex rope precession on the draft tube of Francis turbines. Such disturbance is likely to introduce severe power oscillations that can be extremely detrimental to the overall dynamical behavior of the power plant as well as the power grid.

Comparisons between on-site measurements, time-domain simulation and eigenanalysis results confirmed a very good agreement between the high-order model and the actual installation.

Eigenanalyses were performed to the electrical, hydraulic and complete hydroelectric system. They have proven to be a very useful tool, since they permitted to assess the global stability around the operating point for the part load condition. Also, they clearly indicated all the risks of resonance between the draft tube and the electromechanical eigenmodes. Moreover, these analyses revealed that the draft tube modes for a multi-machine system may present two types of behavior: they may characterize either oscillations in all the machines with the same phase (common mode), or oscillations in phase opposition (inter-machine modes.

Finally, having identified the frequency at which resonance could occur, it is possible to perform targeted, time-domain simulations to quantify the consequences of vortex rope precession, which were proven to be very detrimental. Moreover, the combination of eigenanalysis and time domain simulation is helpful tool for evaluating the pertinence of some possible counter measures.

Although this study was dedicated to one specific power plant, it is important to note that this is a risk common to any Francis turbine hydropower plant, as it may be repeatedly subjected to part load operation. Furthermore, the methodology and models applied in this paper are a general approach, applicable to any hydropower plant independently of its topology.

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IX. Biographies

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