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Dynamic effects in reversible hydro systems towards safety solutions

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Abstract
The purpose of this paper is to establish general strategies to evaluate the dynamic effects occurring in reversible hydro systems (i.e. turbine/pumping) with long penstocks resulting from the regular operation of the hydro equipments and, most significantly, from accidental events. It is of great importance these particular aspects are considered in the early stages of a design, in order to ensure the best technical, economical and safety operation for each developed solution. This work presents two complementary approaches to the study of dynamic effects associated to reversible hydro systems based on a parametric analysis and a simulation-based procedure, as well as in the definition of design and operation rules to guarantee a safe solution.
The first approach establishes the dynamic behavior of the system by means of a parametric analysis of the hydraulic and the hydro mechanical aspects associated to system operation. Based on this methodology, it is possible to estimate the maximum upsurge, the flow variation under turbogenerators runaway conditions and a valve manoeuvre.
The second approach consists in the implementation of a numerical model that simulates accurately enough, the interaction between different components of the system during transient flow regimes associated to the hydropower load rejection, pumps shutdown, actuation of upsurge protection devices and wave propagations along the all system. This methodology gathers the necessary tools for the computational transient analysis of a complex reversible system.

Keywords: Dynamic effects, Reversible hydro, Safety solutions, Hydraulic transients.

1. Introduction

In the operation of hydropower systems it is inevitable the occurrence of variations in the flow, being true either in routine manoeuvres, either in accidental or exceptional unforeseen events. This transition between two steady states brings in a disturbance in the all system in the form of pressure waves that propagate along the pipe system (i.e. penstock /pumping system). To ensure the safety and reliability throughout the system life, it is very important that these dynamic effects and the associated risk factor are considered in the early stages of each design and the overpressures are accurately estimated. A detailed analysis for each operating situation is vital for understanding the dynamic behavior of hydro-mechanical equipment and their interaction with the flow and hydraulic circuit.
This study is based on a parametric characterization of the different components of a hydro system as a support for a CFD modeling.

2. Parametric analysis

In a pressurized system, any disturbance causing a variation of flow will propagate along the hydraulic circuit as a pressure wave. The velocity of the propagation is known as the elastic wave celerity and its magnitude depends on the properties of both the fluid and pipe walls accordingly to Eq. (1).

$$c = \frac{\sqrt{\frac{\varepsilon}{\rho}}}{\sqrt{1+\frac{\varepsilon}{E}\frac{D}{e}}} \tag{1}$$

where ε = volumetric elasticity of the fluid; ρ =density of the fluid; E = Young's modulus of elasticity of the pipe wall; e = thickness of the pipe's wall.

The fundamental theories tell us that the maximum pressure variation without head losses caused by a transient is given by the Frisel-Joukowky formula:

$$\Delta H = \frac{c}{g}U_0 \tag{2}$$

in which U_O is the average flow velocity inside the pipe for the initial steady state flow. This formula is only valid for fast and instantaneous manoeuvres, in which the closure time does not exceed $T_E = 2L/c$.

Extreme pressures may be avoided by adopting a closure time T_C much larger than T_E for the actuation of the flow control devices. For this case, the upsurge variation can be estimated utilizing the Michaud formula:

$$\Delta H = \frac{2L}{gt_f}U_0 \tag{3}$$

For the evaluation of the overspeed effect in a turbogenerator, the author [1] presents the relative overpressure values as shown in Figure 1 as a function of the turbine specific speed, N_s, and the relations T_W/T_m and T_C/T_E.

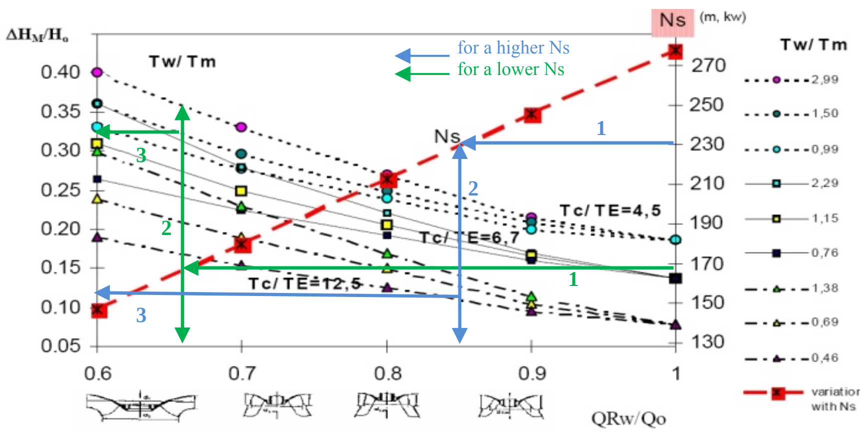

Figure 1. Relative overpressure induced by the turbine overspeed and guide vane closure [1]

The start-up time of rotating masses, T_m *(s)*, is defined based on Eq. (4), where n_0 (rpm) is the nominal runner speed, P_0 the reference power (kW) and $WD^2=4gI$ (N.m^2):

$$T_m = \frac{WD^2 n_0^2}{3575 P_0} \times 10^{-3} \tag{4}$$

and T_W is the hydraulic inertia time constant, defined by the following equation:

$$T_W = \frac{LU_0}{gH_0} \tag{5}$$

where L = pipe length (m); U_0 = initial steady flow velocity (m/s); H_0 = reference net head (m) [2].

3. Simulation-based modelling

3.1 Method of characteristics

The quantitative analysis of unsteady flow through long conveyance systems is based on the fundamental hydrodynamic principles described by the dynamic (Eq. (6)) and continuity (Eq. (7)) equations:

$$\frac{\partial V}{\partial t} + g\frac{\partial H}{\partial x} + \frac{fV|V|}{2D} = 0 \tag{6}$$

$$\frac{\partial H}{\partial t} + V\frac{\partial H}{\partial x} + \frac{c^2}{g}\frac{\partial V}{\partial x} = 0 \tag{7}$$

The previous equations can be used given the following conditions:

1. The fluid in the pipe is homogeneous and mono-phase;
2. The flow is mainly one-dimensional and the velocity profile is considered uniform through the cross-section of the pipe;
3. The pipe axis remains static;
4. The fluid and the pipe walls have physical linear elastic properties.

The method of characteristics was firstly introduced by Streeter and Wylie and since then has been widely used in the analysis of water hammer in pipes where the flow has mainly unidirectional properties [3]. Eqs. (8) and (9) cannot be solved analytically, so a more sophisticated numerical approach is necessary. In order to transform these equations, they can be solved utilizing finite differences, in particular the method of characteristics for the characteristic lines C^+ and C^- :

$$C^+ : \frac{g}{c}\frac{\partial H}{\partial t} + \frac{\partial V}{\partial t} + \frac{fV|V|}{2D} \tag{8}$$

$$C^- : -\frac{g}{c}\frac{\partial H}{\partial t} + \frac{\partial V}{\partial t} + \frac{fV|V|}{2D} \tag{9}$$

These equations are valid in the domain of the characteristic lines presented in Figure 2 and given by:

- Positive characteristic line C^+: $dx = c \cdot dt$;
- Negative characteristic line C^-: $dx = -c \cdot dt$; ;

in which c = celerity of the elastic waves calculated as a functions of the fluid and pipe properties.

Depending on the chosen calculation time step, each pipe will be divided in a finite number of stretches bounded by the calculation sections building a rectangular mesh.

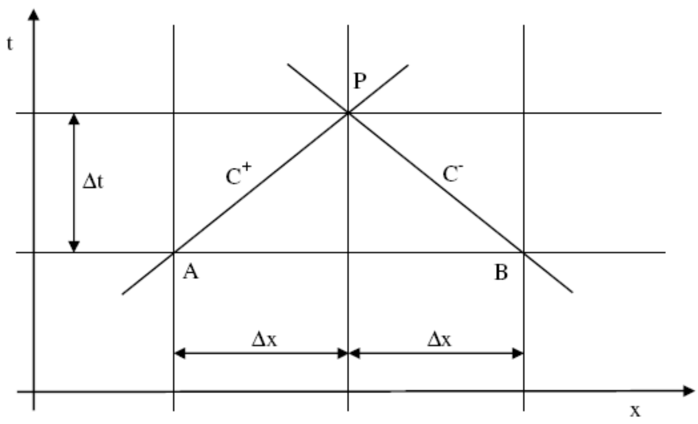

Figure 2. Method of characteristics calculation grid

The positive and negative characteristic equations can be written as follows:

$$Q_P = C_P - C_a H_P \tag{10}$$

$$Q_P = C_n + C_a H_P \tag{11}$$

in which:

$$C_P = Q_A \frac{gA}{c} H_A - \frac{f\Delta t}{c} Q_A \left|Q_A\right| \tag{12}$$

$$C_n = Q_B \frac{gA}{c} H_B - \frac{f\Delta t}{c} Q_B \left|Q_B\right| \tag{13}$$

and:

$$C_a = \frac{gA}{c} \tag{14}$$

Eq. (10) and Eq. (11) are only valid along the positive and negative characteristic lines, respectively. The values of C_P and C_n are known for each time step, although for the interior points of the mesh we have the unknowns Q_P and H_P. These values can be determined by solving simultaneously Eq. (10) and Eq. (11):

$$Q_P = 0.5\left(C_P + C_n\right) \tag{15}$$

At the ends of each pipe it is only possible to define one characteristic line, so in order solve the system another equation is needed. This additional information may be obtained through the introduction of a boundary condition. Depending on the simulated system this boundary may be another pipe (with different characteristics), a reservoir, a turbine, a pump, a valve, a protection device or any other component capable of being analytically described.

3.2 Constant-head reservoir

Considering negligible the head loss at the entrance of the reservoir, we can write the boundary condition in a easy way:

$$H_P = H_{\mathrm{Res}} \tag{16}$$

Depending if the reservoir is at upstream or downstream end of the pipe, Eq. (16) should be substituted on the negative or positive characteristic equation in order to obtain Q_P.

3.3 Turbine

The functioning of a turbogenerator can be characterized by a specific valve-type with adapted characteristics associated to customized closure maneuvers. Even though this solution may roughly simulate the vast majority of systems and situations, it does not consider the specific parameters of the turbine and guide vane devices, nor does it consider the overspeed of the turbine wheel and its complex interaction with the conveyance system. On this matter an innovative formulation was proposed for the simulation of the dynamic behavior of a turbogenerator based on a set of significant parameters that characterize a turbogenerator, allowing the evaluation of overspeed induced extreme upsurges and its propagation along the hydraulic circuit [4-9]. This methodology simulates the turbine as a hydraulic resistive element, where the head lost by the flow is characterized by the basic formula of a hydraulic orifice equipped with dynamic discharge and rotational speed coefficients:

$$q_P = C_g C_S \sqrt{h_P} \tag{17}$$

in which q_P and h_P are respectively the relative flow through the orifice and available head at a certain time step. The factor C_g is the gate opening coefficient that defines the maximum discharge for a given head and rotational speed, as a function of the gate opening. The factor C_S accounts for the runner's rotational speed, adjusting the flow in each time step accordingly with the following formulation:

$$C_S = 1 + \frac{\alpha_R - 1}{\beta_R - 1}\left(\frac{n}{\sqrt{h}} - 1\right) \tag{18}$$

where n stands for the dimensionless value of the rotational speed. The parameters $\alpha_R = Q_E / Q_o$ and $\beta_R = N_E / N_o$ are established for each turbomachine and represent the relation between the runaway situation and rated conditions, respectively for the flow and rotational speed. These values may be obtained from the manufacturers of the turbo equipments.

The variation of the rotational speed for each time step depends on the inertia of the rotating masses, I, and the established equilibrium between the hidraulic torque, T_H, and the magnetic torque, T_M. This relation may be expressed through the rotating mass equation:

$$\frac{d\omega}{dt} = \frac{60}{2\pi}\left(T_H - T_M\right) \tag{19}$$

in which ω is the angular speed (rpm). From Eq. (19), we conclude that for $T_M = 0$, the acceleration of the runner depends only on the hydraulic torque (T_H) and the inertia of the rotating masses, I.

The hydraulic torque actuating in the rated operating conditions is given by the following equation:

$$B_{H,0} = \left(\frac{60}{2\pi}\right)\frac{\gamma \eta_0 Q_0 H_0}{N_0} \tag{20}$$

in wich η_0 = rated efficiency; Q_0 = rated flow; H_0 = rated net head; N_0 = rated runner speed.

Assuming a linear variation of the discharge with the rotating speed under runaway conditions, the hydraulic torque at each timestep may obtained by multiplying the initial torque by a corrective factor, b, given by the following equation where $e = \dfrac{\eta}{\eta_0}$:

$$b = \frac{B_H}{B_{H,0}} = h^{3/2} C_g \frac{e}{n}\left[1 - \frac{\frac{n}{\sqrt{h}} - 1}{\beta_R - 1}\right] \tag{21}$$

The evaluation of the efficiency is a complex matter that depends on a vast set of parameters. This value may be considered to vary accordingly to the following aproximate equations [4] and [9].

$$\begin{cases} \eta_0 \dfrac{N}{N_0} & \text{for N<N}_0 \\ C_g\left(\dfrac{N_E}{N_E - N_0} - \dfrac{N}{N_E - N_0}\right)\eta_0 & \text{for N>N}_0 \end{cases} \tag{22}$$

The formulation presented by the referenced authors, based on the turbine parameters, is of extreme interest, as it can be used as a dynamic boundary condition in the computational evaluation of the extreme upsurges occuring in a hydropower system.

3.4 Pumping system

The pump operation may be accurately simulated using the method of characteristics incorporating specific boundary conditions. Pump manufacturers generally provide the characteristic curves for their pumps as a function of the wheel diameter. These curves relate flow, manometric head, efficiency and power for a specific rotational speed and can be approximately represented in the first quadrant of operation (H,Q) by a second degree polynomial (Eq. (23)) [10]. Should this methodology be applied, with a check valve installed at the downstream pump section to avoid reverse flow occurrence.

$$H_0 = AN^2 + BNQ - CQ^2 \tag{23}$$

The values A, B and C are constants and can be estimated from three known pairs of values (Q, H) of the pump characteristic curve. If correct data is available, a higher degree polynomial may be considered to enable a multi operating zone simulation. Although pump characteristic curves in the pumping zone are usually easy to obtain, the same is not true for the remaining zones of operation.

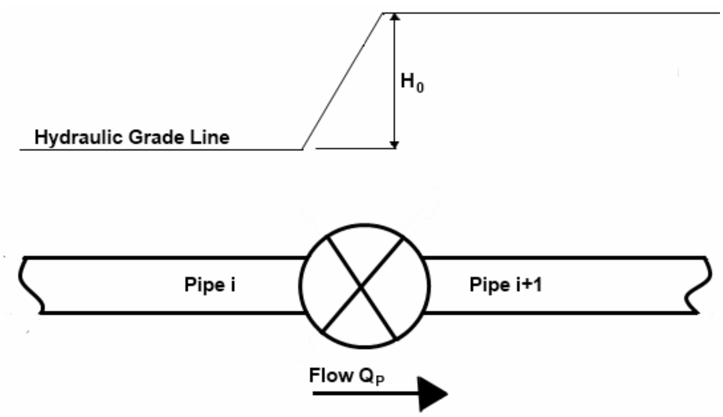

Figure 3. Schematic drawing of a pump

Referring to Figure 3 and considering the continuity equation, we can write $Q_{P,i} = Q_{P,i+1}$. If the losses at the junctions are neglected we have from Eq. (24) the relation between head and discharge, in which *N* is the rotational speed:

$$H_{P,i+1} - H_{P,i} = AN^2 + BNQ - CQ^2 \tag{24}$$

when $A_i = A_{i+1}$, substituting Eqs. (10) and (11) into Eq. (24) we obtain the following polynomial:

$$CC_aQ_P^2 + (BNC_a - 2)Q_P + (AC_aN^2 + C_n + C_P) = 0 \tag{25}$$

solving for Q_P yields:

$$Q_P = \frac{2 - BNC_a - \sqrt{(BNC_a - 2)^2 - 4CC_a\left(AC_aN^2 + C_P + C_n\right)}}{2CC_a} \tag{26}$$

we obtain the flow Q_P and heads $H_{P,i}$ and $H_{P,i+1}$ for each time step. The total or partial shutdown of the pump as the cause of severe transient regimes may also be simulated. In this case the variation of the runner speed is given by Eq. (19) in which the hydraulic torque can be calculated based on efficiency data provided by the manufacturer.

3.5 Valve

Valves are installed in hydro systems to regulate the flow (or the pressure) by opening, closing or partially obstructing the pipe section (Figure 4). They can also be utilized to isolate system components in order to enable maintenance procedures or even their replacement.

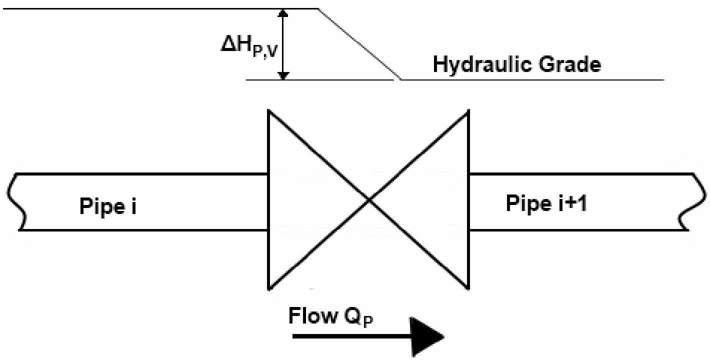

Figure 4. Schematic of a generic valve

The boundary condition for a valve can be written based on steady state head loss equation:

$$\Delta H_{P,V} = K_V \frac{Q_P^2}{2Ag} \tag{27}$$

in which K_V is a head loss coefficient obtained experimentally that depends on the Reynolds number, the type of valve and its opening at each time step. The flow through the valve, Q_P, remains constant and the head loss can be written as the difference between the head at upstream and downstream valve section:

$$\Delta H_{P,V} = H_{P,i} - H_{P,i+1} \tag{28}$$

Substituting Eq. (10) or (11) in Eq. (27) we obtain the second degree polynomial equation:

$$\frac{K_VC_a}{2gA^2}Q_P^2 + 2Q_P - (C_P + C_n) = 0 \tag{29}$$

Solving for Q_P we obtain the flow through the valve for each instant of simulation and subsequently the upstream and downstream head:

$$Q_P = \frac{-2 + \sqrt{4 + 2\frac{K_V C_a}{gA^2}\left(C_P + C_n\right)}}{\frac{K_V C_a}{gA^2}} \tag{30}$$

For the situation where the valve is fully open, the head loss may be almost neglected and the flow can be determined by solving Eq. (15) or a fixed value should be considered. For the determination of the parameter K_V, a set of empirical results for each type of valve is presented in [11].

3.6 Air vessel

An air vessel is a container filled with air at its top and water in its lower part (Figure 5). This vessel is connected to the main pipe via an orifice designed to increase entrance head loss. Generally, to prevent the occurrence of lower pressures inside the pipe, the outflow from the container should be without restriction, while the inflow may be constrained to increase the damping effect and reduce the necessary volume. This may be obtained by considering an orifice designed so it has a differential effect, producing more head loss for the inflow to the vessel than for the equivalent outflow.

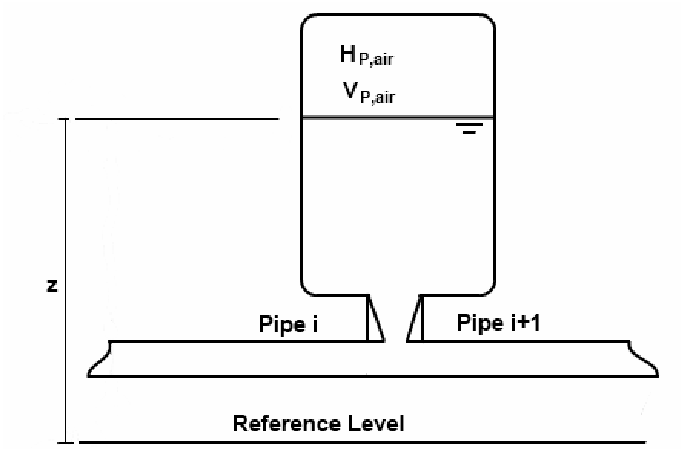

Figure 5. Air vessel schematic

If head losses at the junction between the vessel and the pipe are neglected, we may consider the approximation $H_{P,i} = H_{P,i+1} = H_P$. From the continuity principle we can write Eq. (31) in which $Q_{P,orifice}$ is the flow through the orifice.

$$Q_{P,i} = Q_{P,i+1} + Q_{P,orifice} \tag{31}$$

Substituting the positive and negative characteristic equations (Eqs. (10) and (11)) in Eq. (31) we relate the flow through the orifice with the head at the adjacent pipe section, H_P:

$$Q_{P,orifice} = \left(C_P - C_n\right) - \left(C_{a,i} + C_{a,i+1}\right) H_P \tag{32}$$

The relation between the flow $Q_{P,orifice}$ and the head loss may be expressed by the following general equation, where C_C is the contraction coefficient and A is the section of the orifice:

$$Q_{P,orifice} = C_C A\sqrt{2g\Delta H_P} \Leftrightarrow \Delta H_P = \frac{1}{2g(C_C A)^2} Q_P \left|Q_P\right| \tag{33}$$

Considering that the entrapped air volume follows a polytropic process during its contraction and expansion, we can write the following relation:

$$H_{P,air} \cdot V_{P,air}^{n} = C \tag{34}$$

in which $H_{P,air}$ and $V_{P,air}$ are respectively the absolute pressure head and volume of the enclosed air at the end of the current time step; n is the polytropic index and C is a constant value that may easily be obtained from the initial conditions. The values n = 1.0 and n = 1.4 corresponds to an isothermal and adiabatic expansion and contraction of the air, respectively. The process is approximately adiabatic for small vessels and rapid variations of volume, and it is almost isothermal for large volumes and slow contraction or expansion. Some authors recommend for design calculations an average value of m = 1.2 based on the fact that transients are generally rapid in the beginning and slow towards the end [12, 13]. The pressure head of the entrapped air can be written as a function of the head, $H_{P,i}$, the barometric pressure, H_b, the elevation of the free surface, z_P, and the head loss at the orifice, $\Delta H_{P,orifice}$:

$$H_{P,air} = H_P + H_b - Z_P - \Delta H_{P,orifice} \tag{35}$$

The volume of air, $V_{P,air}$, at the end of a certain time step can be obtained through the geometric relation, in which A_C represents the section of the air vessel and z is the elevation of the free surface at the beginning of the considered time step:

$$V_{P,air} = V_{air} - A_C\left(Z_P - Z\right) \tag{36}$$

The free surface at the end of a time step can be estimated based on the average flow through the orifice during that same time step:

$$Z_P = Z + 0.5\left(Q_{P,orifice} + Q_{orifice}\right)\frac{\Delta t}{AC} \tag{37}$$

Replacing Eqs. (32), (33) and (35) to (37) in Eq. (34) we finally obtain one equation with one unknown, $Q_{P,orifice}$. This equation can easily be solved by an iterative method using the values of the previous time step as a good first approximation.

4. Comparison between simulations and tests

The described methodology was utilized in the analysis of the hydro loop represented in Figure 6. The circuit is composed by a pressurized tank at the upstream section, a pipe with D_{int} = 0.043 m and a length of 200 m, a downstream valve and a pump responsible for setting a circulated constant steady flow.

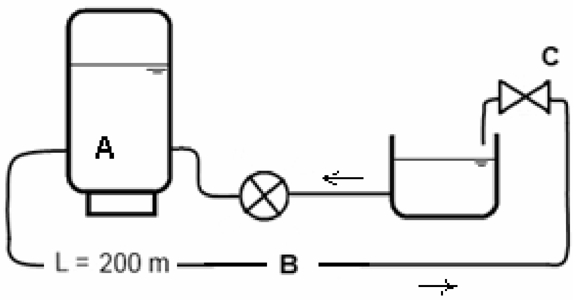

Figure 6. Experimental set-up

Figure 7 represents the maximum upsurge recorded respectively in sections C (downstream end) and B (middle pipe section) for an initial flow of 2.46 l/s and a head of 13.5 m in the air vessel. Even though the experimental recordings of pressure head across time follow approximately the general shape of the simulation generated data, the two series differ significantly in some instants. This is originated by different factors: the rheological pipe material behavior of the induced stress in viscoelastic performance and wave speed variation and the total head losses of the whole system. In real systems the unpredicted ageing of the pipes can also be a cause of a poorly estimated head loss.

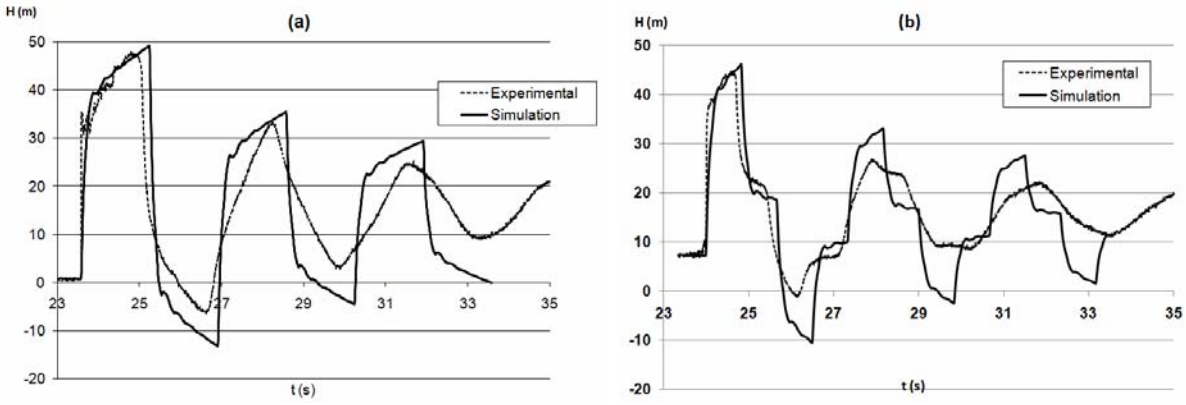

Figure 7. Pressure variation: in C pipe section (downstream end) (a); and in B (the middle pipe section) (b)

For a successful computational modeling it relies on its correct calibration and validation based on lab and real life measurements, not always attained. In case of doubts is preferable to be able to simulate the safe solutions with a safety factor.

5. Simulations

5.1 Case study

For the purpose of demonstration we consider a hypothetic reversible hydropower installation (Figure 8). The system is equipped with a long pressurized conveyance system with a length of 4000 m. The considered elastic wave celerity is 1000 ms^{-1}. The water level above the datum is 140 m for reservoir A and 240 m for reservoir B, resulting in a gross head of 100 m. The installed equipment is a pump with a nominal rotating speed is 1500 rpm that is able to generate power when reversed (PAT). When working as a turbine, the air vessel is out of service and the valve is operating as a guide vane, and the specific rotating speed is N_s=100 rpm (m, kW). When working in the normal mode, as a pumping system, the air vessel is considered as the protection device for this pumping system.

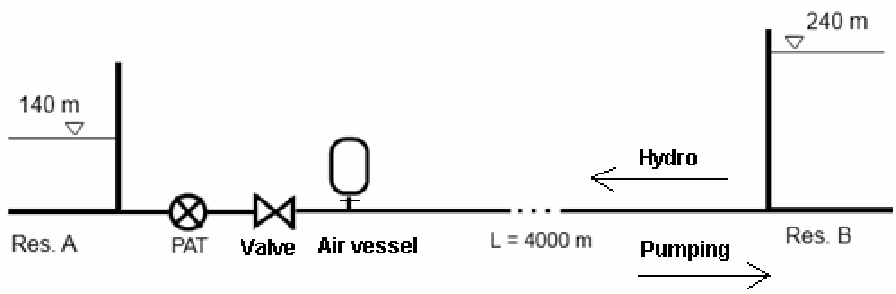

Figure 8. Scheme of the case study for a reversible hydro system

5.2 Hydro system

The reduction of the flow in a control device located downstream of a long penstock induces in the system transient regime condition able to produce severe upsurges caused by the inertia forces actuating the water column. The developed analysis should consider the system functioning for extreme events, where it is simultaneously necessary to quickly reduce the flow and guarantee the safety of the entire hydropower system. Thus, utilizing the previously described methodology, a set of simulations is carried out in order to better understand the involved dynamic effects. We obtain, for a full load rejection and the downstream valve closure in a time $t_c = 30\ s$, the relative pressure head in the power house presented in Figure 9 and the variation of rotational speed and flow presented in Figure 10 (a) and (b) respectively.

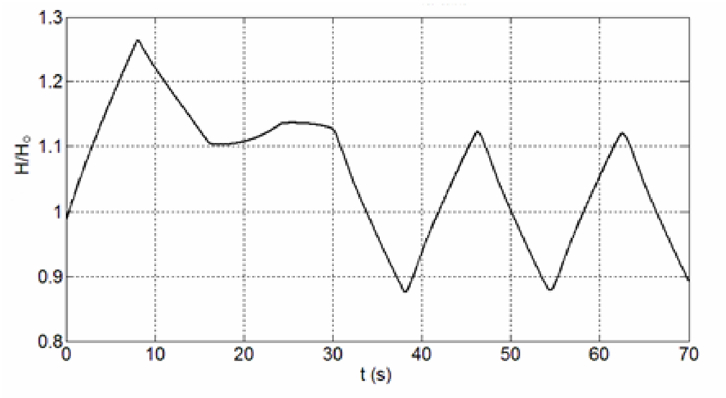

Figure 9. Relative head at the valve section

Since the PAT during a sudden load rejection is actuated by the hydraulic torque, the runner will rapidly accelerate until the runaway speed, which is attained in few seconds as visible in Figure 10 (a). The chosen equipment is characterized by a low specific speed, N_S. According to Figure 1, this corresponds to a quasi linear flow reduction (Figure 10 (b)) causing a transient regime responsible for the first relative pressure head peak visible in Figure 9.

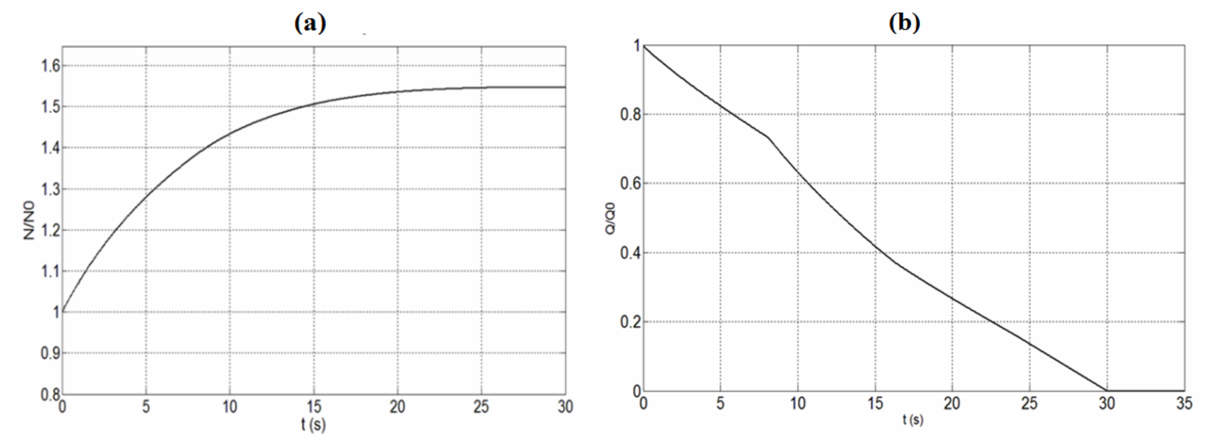

Figure 10. Runaway condition: relative runner rotating speed (a) and flow variation (b)

Figure 11 presents the variation of the relative hydraulic torque and efficiency along time for the runaway situation. These parameters decrease rapidly as the PAT begins to deviate from its rated running conditions.

In this particular of this case study it is important to mention that the most severe pressure head happens in the early seconds, which is caused by the flow reduction resulting from the rapid acceleration of the runner, as usually is neglected. Depending on the characteristics of each hydropower system, this CFD allows investigating the dynamic behavior of the installed turbomachine and its interaction with the rest of the system's components.

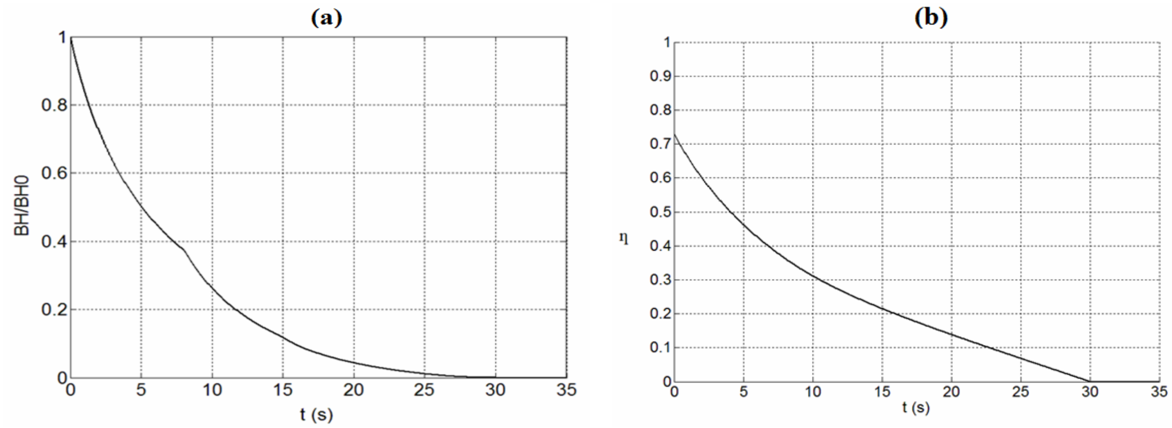

Figure 11. Runaway condition: hydraulic torque variation (a) and efficiency (b)

5.3 Pump system

The pump shutdown is a typical maneuver which originates low pressures that in extreme situation may attain the vapor pressure in some pipe sections. This causes the vapor pockets opening and closing, originating severe head oscillations across the system. In this operating scheme it is advisable to consider a protection device that helps maintain the pressure in all pipe sections within recommendable limits. In the developed methodology it was presented the basic formulation and some guidelines for its successful modeling. Considering the presented case study, Figure 12 (a) and (b) show respectively the relative pressure head variation at the valve section and the pressure envelopes with and without considering the protection device for closure time of the valve $t_c = 20\ s$.

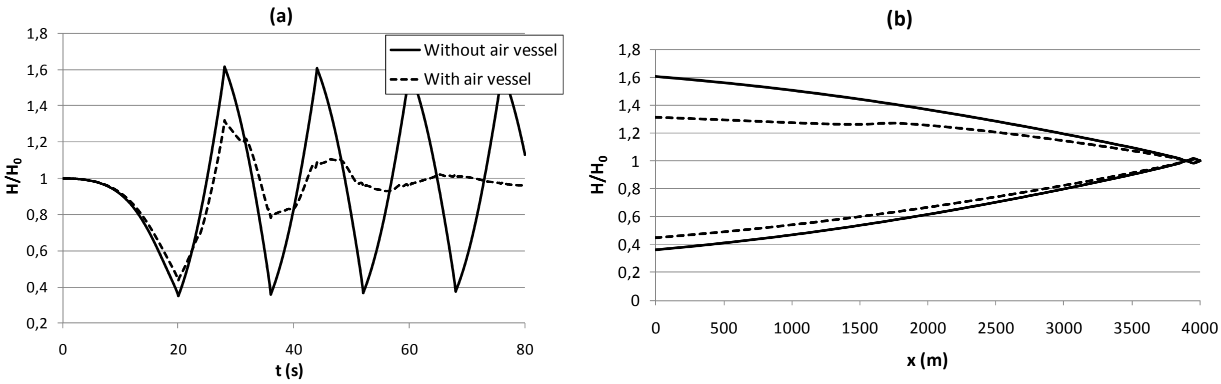

Figure 12. Head downstream of the valve (a) and head envelops (b) with and without protection device

It is immediately noticeably that the presence of the air vessel results in lower extreme values for all pipe sections and that the pressure wave propagation fades more rapidly causing less strain. For extreme low pressures, the relief comes from the negative flow through the differential orifice in the connection of the air vessel to the main pipe. In the occurrence of high pressures, the equivalent effect is obtained via the positive flow through the orifice. The transfer of the fluid from and to the protection device is accompanied by a variation of the water and air volume inside the vessel and the pressure vary accordingly. In Figure 13 (a) and (b) we can see respectively the flow through the orifice and the air pressure inside the vessel for each simulation instant. Figure 14 (a) and (b) shows the variation of the free surface level and the volume of air inside the vessel.

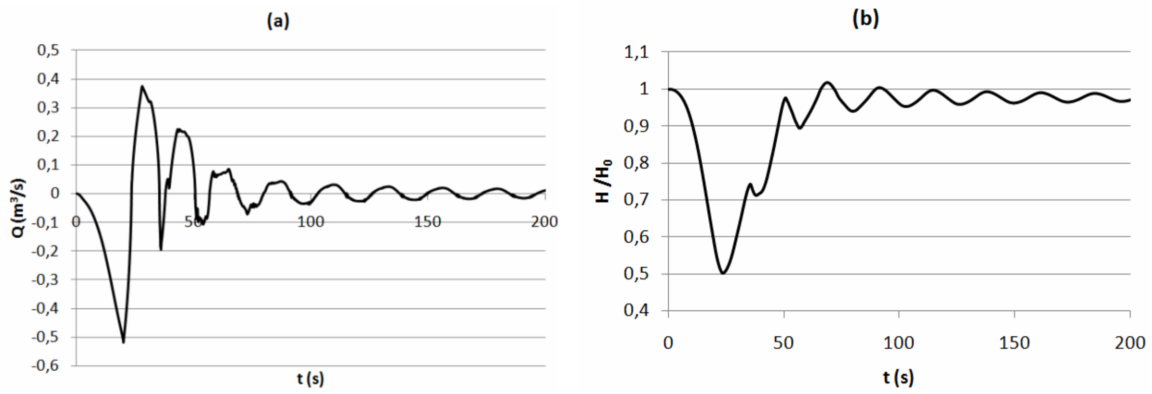

Figure 13. Air vessel: variation of the flow through the vessel orifice in the connection to the main pipe (a) and air pressure head (b)

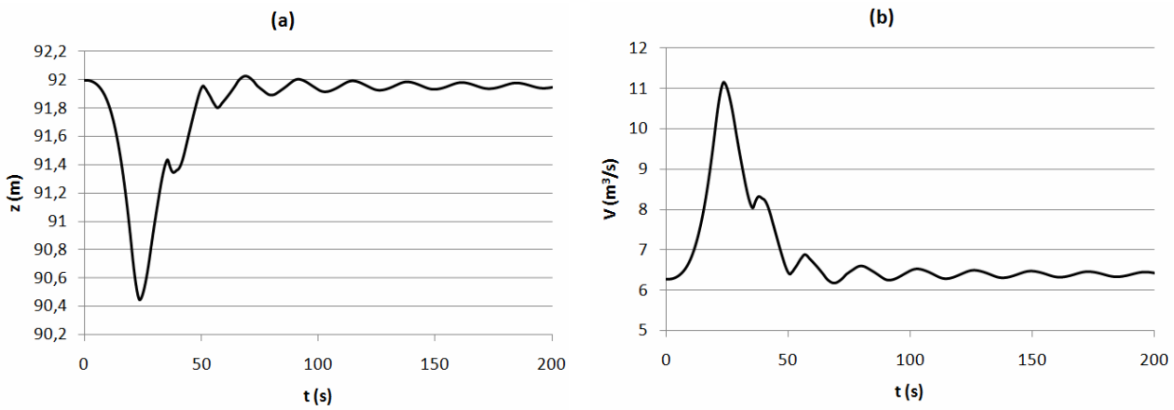

Figure 14. Air vessel: variation of the free surface level (a) and air volume (b)

6. Conclusions

The computational simulation based on hydraulic principles constitutes a powerful tool in the comprehension of the complex dynamic effects occurring in each system characteristics. This formulation based on modular linkable components proved to be flexible enough to allow the easy setup of a vast set of different models that are able to simulate the majority of situations. When compared to other less advanced approaches, this methodology has the ability of predicting the runner overspeed of turbogenerators and its effect on the flow along the pipe system. This analysis enables the engineers and designers to economically test different scenarios and solutions, quickly evaluate the pressures involved and make decisions regarding the necessity or not to consider special operating rules and protection devices.

From the conducted simulations, we can conclude for certain system parameters, the overspeed effect associated to complete load rejection in a power plant may be one of the most important security constraints, inducing significant overpressures that must be within the resistance tolerance of the pipe and accessories. In the pumping system the pumps shutdown are the major critical manoeuvre and the consideration of protection devices (such as an air vessel) leads to lower head envelopes and promoting the rapid dampening of the upsurges and constituting an economic and effective pressure control solution.

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