Risk of penstock fatigue in pumped-storage power plants operating with variable speed in pumping mode

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A B S T R A C T

The upgrade of a pumped-storage power plant (PSPP) to allow variable speed operation offer several advantages in pumping and generating modes. However, in pumping mode at part load, both pressure and torque pulsations develop in the pump turbine runner. This paper evaluates the risk of fatigue damage in the penstock of a variable-speed PSPP due to the propagation of the pressure pulsations developing in the pump turbine runner at partial load in pumping mode. For that purpose, a simulation model of a variable-speed PSPP has been developed. The pressure and torque pulsations are generated each from a different set of sinusoidal functions calibrated from the results of a Computational Fluid Dynamic model, which was in turn validated from experimental data. A Monte Carlo simulation has been performed considering different temporal gaps between the sinusoidal functions reproducing the pressure pulsations in one and another pump turbine. The number of stress cycles that may cause fatigue damage in the penstock has been obtained from the results of the simulations and the fatigue curves defined in the Eurocode, and then transformed into the maximum number of hours per year the PSPP can operate at partial load in pumping mode to avoid fatigue damages.

Keywords:
Failure analysis
Power plant
Fatigue life
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Simulation test

1. Introduction

Variable speed operation offer a lot of advantages for a pump-turbine operating both in generating and pumping mode, i.e. the possibility of an almost instantaneous active and reactive power control, higher efficiency or wider operating range [1].

Nevertheless, at part load in pumping mode, reversible pump-turbine machines suffer from unstable behaviour related to a strong fluid-dynamical interaction between rotor and stator parts that causes the development of unsteady phenomena [2]. The pressure and torque pulsations associated with these unsteady phenomena might propagate both along the power plant conduits and to the electrical grid [3]. The torque pulsations might in turn have a certain influence on the power quality and, as a consequence, affect the design and cost of the converter filters. The

propagation of pressure pulsations might cause fatigue in the penstock and, as a consequence, reduce the power plant lifetime or require a higher investment [4]. Material fatigue is a phenomenon of extreme importance recognised as one of the major causes of destruction of materials. In some cases, long structures such as penstock in hydropower plants can suffer serious damages caused by a sudden rupture due to fatigue effects [5].

Fatigue damages in hydropower plant conduits have been barely addressed in the scientific literature. In Ref. [6] the authors describe a detailed integrity assessment of a riveted penstock in a hydroelectric power plant, and conclude that the possibility of a fast fracture event in this particular penstock is remote. In Ref. [7] authors evaluated the risk of fatigue in the penstock of a specific hydropower plant due to the tracking of the secondary load-frequency control set-point signals, and concluded that the secondary load-frequency control did not cause significant fatigue damages with a proper set of turbine governor parameters. However, fast hydraulic transient phenomena such as the pressure pulsations that develop at part load in pumping mode were not considered in the above-mentioned studies. Ref. [8] is to the author's knowledge the only work where the impact of said pressure

Nomenclature

a	wave speed (m/s)	P	pressure (m.w.c.)
A_H, B_H, C_H	pump curve coefficients	p	water pressure
$A_{H,p}$	pressure amplitude for p peak (m)	P_p	higher amplitude pressure peaks assumed
B	impeller or guide vane or return channel width	Q	pump flow (m^3/s)
D	pipe diameter (m)	Q	flow (m^3/s)
D_p	pump turbine components diameter	Q_d	designed pump flow (m^3/s)
e	conduit thickness (mm)	Q_{fp1}	flow in feeding pipe 1 in steady state (m^3/s)
f	Darcy-Weisbach friction factor	Q_{fp2}	flow in feeding pipe 2 in steady state (m^3/s)
f_{fp1}	feeding pipe 1 Darcy-Weisbach friction factor	Q_{lab}	pump flow in laboratory scale ($0.0729 m^3/s$)
f_{fp2}	feeding pipe 2 Darcy-Weisbach friction factor	Q_p	pump flow in the PSHP
$f_{H,p}$	pressure frequency for p peak	$q_{p,i}$	flow at the i th Γ element of the penstock (p.u.)
f_p	penstock Darcy-Weisbach friction factor	Q_{pe}	flow in penstock in steady state (m^3/s)
g	gravity acceleration (m^2/s)	$r/2$	continuous head losses coefficient in pipes (p.u.)
H	pump head (m)	S	pipe section (m^2)
H_b	pressure in the union between feedings pipe and penstock (m.w.c.)	S_{fp}	water speed in the feedings pipes (m/s)
H_d	designed pump head (m)	S_{fp1}	feeding pipe 1 section (m^2)
H_{dp}	pressure downstream the pump (m)	S_{fp2}	feeding pipe 2 section (m^2)
$H_{fp1,N}$	pressure in the bifurcation previous point in feeding pipe 1 (m.w.c.)	S_p	penstock section (m^2)
$H_{fp1,N+1}$	pressure at bifurcation measured in feeding pipe 1 (m.w.c.)	s_p	water speed in the penstock (m/s)
$H_{fp2,N}$	pressure in the bifurcation previous point in feeding pipe 2 (m.w.c.)	S_t	Strouhal number
$H_{fp2,N+1}$	pressure at bifurcation measured in feeding pipe 2 (m.w.c.)	t	time (s)
h_i	head at the i th Γ element of the penstock (p.u.)	t_0	starting time of speed reduction in pump 1
H_{i+1}	pressure at the points to the right of P_i (m)	T_e	L_p/a , water elastic time (s)
H_{i-1}	pressure at the points to the left of P_i (m)	T_w	penstock water starting time (s)
H_{lab}	pump head in laboratory scale (6.97 m)	V	mean flow velocity in conduits (m/s)
H_{lab}^*	pressure pulsation in laboratory scale (m)	$V_{fp1,N}$	water speed in the bifurcation previous point in feeding pipe 1 (m/s)
H_{lp}	pressure at the pump entrance from lower reservoir (m)	$V_{fp1,N+1}$	water speed at bifurcation measured in feeding pipe 1 (m/s)
H_{lr}	constant level in the lower reservoir (m)	$V_{fp2,N}$	water speed in the bifurcation previous point in feeding pipe 2 (m/s)
H_p	pump head (m)	$V_{fp2,N+1}$	water speed at bifurcation measured in feeding pipe 2 (m/s)
$H_{p,1}$	pressure at bifurcation measured in penstock (m.w.c.)	V_{i+1}	water speed at the points to the right of P_i (m/s)
$H_{p,2}$	pressure at after bifurcation point in penstock (m.w.c.)	V_{i-1}	water speed at the points to the left of P_i (m/s)
$H_{p,plant}$	pump head up-scaled (m)	$V_{p,1}$	water speed at bifurcation measured in penstock (m/s)
H_p'	pump head including pressure pulsations (m)	$V_{p,2}$	water speed at after bifurcation point in penstock (m/s)
H_{pi}	pressure at the i th sample point along of the pipe	V_{pi}	water speed at the i th sample point along of the pipe
H_{plant}^*	pressure pulsation up-scaled (m)	x	axial distance along the streamline
H_{up}	pressure at the pump entrance from upper reservoir (m)	z	elevation at any point of the streamline
H_{ur}	constant level in the upper reservoir (m)	α	hydraulic similarity factor
L_{fp}	feeding pipes length (m)	β	hydraulic similarity factor
L_p	penstock length (m)	Δt	time interval in MoC (s)
n	rotational speed (r.p.m.)	Δt_0	time gap between both pumps (s)
$n_{b,r}$	impeller number of blades	Δx	length in pipe elements which it has been divided in MoC
$n_{b,rc}$	return Vane number of blades	λ	Lagrange multiplier
$n_{b,wg}$	wicket Guide number of blades	λ_p	guide vanes' azimuthally position
N_g	probability factor in Monte Carlo simulation	ρ	Water density
N_k	number of times a stress peak is repeated	σ_{cp}	tangential stress [MPa]
n_t	number of segments in which the penstock is divided	$\sigma_{cp,k}$	tangential stress k obtained by RFCM [MPa]
p	peak number in pulsation generators	$\sigma_{eq,g}$	tangential stress obtained for each time gap g considered [MPa]
		σ_{eq}'	equivalent tangential stress for each gross head in Monte Carlo simulation [MPa]

pulsations on the risk of fatigue damage in the penstock of a pumped-storage power plant (PSPP) has been preliminarily evaluated. For that purpose, the authors developed a dynamic simulation model of a variable-speed PSPP operating in pumping mode in

unstable operating conditions. A set of sinusoidal functions were proposed in the paper, and integrated in the dynamic simulation model, in order to synthetically generate both pressure and torque pulsations that reproduce the operation of the hydraulic machine

in its instability region. The proposed sinusoidal functions were validated against the results of CFD (Computational Fluid Dynamics) simulations carried out on a reversible pump-turbine model installed in the Department of Industrial Engineering of the University of Padova (DIE-UP).

The objective of this paper is to evaluate the risk of fatigue damage in the penstock of different variable-speed PSPPs due to the propagation of the pressure pulsations developing in the pump-turbine in unstable operating conditions in pumping mode. This objective is of special interest for existing PSPPs which, maintaining the existing pump turbine runner, will be upgraded to allow variable speed operation, as is the case in some PSPPs recently analysed in the EU FP7 eStorage project [9]. According to the deliverable D4.1 of the said project, a number of pump-turbine units amounting to 34.9 GW will need to be refurbished by 2020, some of which might be upgraded to allow variable speed operation.

The main reason for upgrading a PSPP to operate with variable speed is that the operation with variable speed allows the PSPPs to provide frequency restoration reserves in pumping mode, and therefore [10] to significantly increase its revenue. In a significant number of European regional power systems (France, Spain and Belgium, among others), frequency restoration reserves are remunerated for two different concepts, reserve availability and delivery [11]. In general, the more reserve capacity, the higher the revenue.

This paper is an extension of the work presented in Ref. [8]. The results of the fatigue analysis presented in that paper indicated that there is no risk of fatigue damage in the penstock of the PSPP analysed therein, as long as the pump-turbines do not operate at part load in pumping mode for a long time. However, authors recognised that evaluating the effect of pressure pulsations in pumped-storage power plants with higher heads than the one analysed in that paper is necessary since unstable operating conditions might have a greater impact. The methodology used in Ref. [8] to analyse the risk of fatigue damage in the penstock of a single variable-speed PSPP is applied here in a number of variable-speed PSPPs with a similar configuration but different design parameters (gross head and life time). A typical PSPP scheme has been assumed, with a single penstock feeding two identical pump-turbines. As an additional contribution to the work developed in Ref. [8], the maximum number of hours per year the PSPP can operate at part load in pumping mode to prevent fatigue damages in the penstock has been obtained from the results of the fatigue analysis, as a function of the PSPP expected lifetime and design gross head. These values have been compared to: a) the number of hours per year in which several PSPPs operating in the Spanish power system provide secondary regulation in generating mode and b) the number of hours per year in which several theoretical PSPP whose optimum operation has been simulated would operate at partial load in pumping mode [12]; c) the number of hours per year in which two existing variable speed PSPPs provide regulation services both in generating and pumping mode.

The methodology used in Ref. [8] is elaborated in this paper, since for extension reasons it was not fully described in the Hyperbole Conference. The risk of fatigue damage in the penstock is evaluated by means of a Monte Carlo simulation-based fatigue analysis. A dynamic simulation model of the PSPP is developed for this purpose. The pressure and flow transients in the PSPP conduits are modeled by the well-known Method of Characteristics [13]. The pump-turbine operation curves at steady-state (head-flow-rotation speed and torque-flow-rotation speed) and the sinusoidal functions that reproduce the pressure and torque pulsations developing in the pump-turbine at part load in pumping mode were up-scaled by hydraulic similarity, from those calibrated in Ref. [8] against the results of several CFD simulations that were carried out on a reversible pump-turbine model installed in the DIE-UP, in a similar

way to [14]. A dynamic simulation model of the DIE-UP test rig has been developed for calibration purposes.

The paper is organised as follows. In section 2, the validation of the CFD model and the CFD simulations carried out on the pump-turbine model of the DIE-UP are summarised. In section 3, the DIE-UP test rig and its simulation model are presented, and the calibration of the pressure pulsations generators is reported. In section 4, the scheme of the analysed PSPP is described and the PSPP simulation model is explained in detail. The fatigue analysis is carried out in section 5. Finally, section 6 contains the conclusions reached in this study.

2. CFD simulations

Numerical analyses were carried out on the first stage of a reversible pump-turbine installed in the Laboratory of Hydraulic Machines of the Department of Mechanical Engineering of the University of Padova (Fig. 1). The pump-turbine consists of a seven-blade 3D impeller coupled with a vaned diffuser and a vaned return channel, reproducing the scaled down first stage of a two-stage reversible pump-turbine. The main geometrical characteristics of the reversible pump-turbine are reported in Table 1.

The peculiarity of this reversible pump-turbine is the possibility of modifying not only the opening angle of the wicket guide vanes but also the relative azimuthal position of the wicket gate vanes with respect to the return channel blades. In particular, being fixed the wicket gate angle α_{3b} , the whole wicket gate cascade can rotate, as a rigid system, of an angle λ_p around the pump-turbine axis, modifying in this way the azimuthal position of the guide vanes with respect to the return channel vanes. Fig. 2 reports the so called “face-to-face” configuration in which the return channel vanes are aligned to the wicket gate vanes ($\lambda_p = 0^\circ$). The rotation of the whole system allows to modify this azimuthal position up to the maximum misalignment of $\pm 8^\circ$ from the face-to-face configuration. In the considered analyses, the wicket gate vanes were $+8^\circ$ out of alignment with an opening angle of 18° , corresponding to a radial gap between their leading edge (in pumping mode) and the impeller tip of 10.5 mm (5.25% of the impeller radius).

The entire pump-turbine (inlet duct, impeller, wicket gate vanes, return channel and leakage system) was discretised by means of structural meshes, properly sized and designed to capture unsteady phenomena developing in the unstable operating area of

Fig. 1. Three-dimensional scheme of the pump-turbine.

Table 1

Main geometrical and performance characteristics of the pump-turbine.

Impeller data				
D_{p2} (mm)	B_2 (mm)	$n_{b,r}$	β_{2b} (°)	ϕ_{Des}
400	40	7	26.5	0.125
Wicket Guide data				
D_{p3} (mm)	B_3 (mm)	$n_{b,wg}$	α_{3b} (°)	λ_p (°)
410	40	22	10÷30	−8÷8
Return channel vanes data				
D_{p4} (mm)	B_4 (mm)	$n_{b,rc}$	α_{4b} (°)	
516	40	11	30	

the pump-turbine with a resulting computation domain of more than 13 million of cells (Fig. 3). More details on the mesh can be found in Ref. [15].

As regards the turbulence model, a Detached Eddy Simulation (DES) model was adopted due to the large flow separations expected at part loads and particularly in the unstable operating area. A fully turbulent boundary layer was assumed on blades and end walls with transient sliding interfaces between stator and rotor blocks. Experimental values of the total pressure were assumed at the inlet boundary condition, whereas at the outlet average static pressure values were combined with an opening condition in the hypothesis of a perturbed flow field.

2.1. Validation of the numerical model

To validate the numerical model, the numerical results, obtained by steady simulations carried out at several operating conditions and at a constant rotation rate of 600 rpm, were compared with experimental data acquired, according to ISO standards, on the pump-turbine installed in the Laboratory of the DIE-UP [16]. For each operating point, the simulation convergence was achieved after more than 20 impeller revolutions and the numerical results were acquired for a time period equal to five more revolutions.

The comparison with the experimental results showed a good accuracy of the numerical model with a maximum percentage error of 3.84% in head and a small over-prediction (max +1.24%) in the unstable operating area (Fig. 4).

2.2. Numerical analysis of a power reduction scenario

The evolution of the flow field in a power reduction scenario was numerically analysed by means of a transient simulation with time-varying boundary condition. This scenario was reproduced by progressively reducing the impeller rotation rate and by

simultaneously varying the inlet total pressure and the outlet average static pressure on the basis of the corresponding experimental values at steady conditions.

The initialization was based on the results of the steady simulation at the design flow rate and the impeller rotation rate was assumed to linearly vary from 100% (600 rpm) to 88% (530 rpm), corresponding to a power reduction from full load to about 46% with a ramp rate of 1.5% (continuous line in Fig. 4). In such a scenario, the flow field of the pump-turbine experienced the progressive onset and development of unsteady phenomena, characterised by pressure and torque pulsations more and more intense. More details on the flow field evolution can be found in Ref. [17].

3. DIE-UP plant description

A dynamic simulation model of the DIE-UP test rig has been developed in order to reproduce the above described CFD simulation and calibrate a sinusoidal function aiming to reproduce the pressure pulsations that develop in the DIE-UP pump-turbine at part load in pumping mode. Due to the scope of the paper, authors have not considered including in the model the expression which relates the torque to the pumped flow.

The DIE-UP test rig consists of a suction pipe with around 10 m length, a reversible pump-turbine, and a short impulsion pipe connected to a tank with a spillway (see Fig. 5 and Fig. 6).

The simulation model of the DIE-UP test rig considers an elastic water column in the suction pipe by means of an approach similar to the one used in Ref. [18]. Due to its short length, a rigid water column is considered in the impulse pipe. The flow-head loss function has been obtained by linear regression from the results of the CFD simulations.

The variable speed pump-turbine has been modeled through its operation curve (1) Parameters in expression (1) corresponding to

Fig. 2. Face-to-face position of the wicket gate vanes in comparison with the return channel blade ($\lambda = 0^\circ$).

Fig. 3. Detail of the coarse mesh of the numerical model: the rotor-stator interaction area.

Fig. 4. Head-flow discharge curves at different rotation rates (600 rpm and 530 rpm): comparison between numerical and experimental results.

$n = 558$ rpm were obtained by linear regression from the results of the CFD simulation. The parameters corresponding to other rotation speeds were then obtained by hydraulic similarity from the previous ones, where α are the hydraulic similarity factors defined in (2).

$$H_p = \alpha^2 A_H + \alpha B_H Q_p + C_H Q_p^2 \quad (1)$$

$$\alpha = \frac{n}{558} \quad (2)$$

To include the effect of pressure pulsations in the model, a function that synthetically generate pressure H_{lab}^* (3) pulsations has been added [8].

$$H_{lab}^* = \sum_{p=1}^{p=P_p} A_{H,p}(n) \cdot \sin(2\pi \cdot f_{H,p}(n) \cdot t) \quad (3)$$

The amplitude ($A_{H,p}$) and frequency ($f_{H,p}$) in (3) has been obtained from the results of a spectral analysis carried out on the results of the CFD simulation. The results of the CFD simulation have been partitioned into 32 groups, each corresponding to a different range of rotation speeds. For each group, the pressure FFT has been obtained. The P_p higher amplitude peaks of the pressure

Fig. 5. DIE-UP test rig photography.

Fig. 6. DIE-UP test rig scheme.

FFT of each group, along with their corresponding frequencies, have been associated to the mean rotation speed of the group. The amplitudes and frequencies of pressure pulsations, corresponding to other intermediate rotation speeds have been obtained by linear interpolation.

The CFD simulation has been reproduced in the simulation model, considering different values of P_p . As can be seen in Fig. 7a and b, both the power spectrum of pressure pulsations and the results of the dynamic simulation model with $P_p = 6$ matched significantly with those obtained in the CFD simulation.

4. PSPP description and simulation model

A typical PSPP scheme (Fig. 8) has been considered which consists of a long penstock that branches into two short feeding pipes, each connected to a variable speed pump-turbine (identical to each other). Each pump-turbine is in turn connected to the lower reservoir through a short draft tube. Different gross heads have been considered in order to study its influence on the risk of fatigue damage in the penstock. The length and water speed of/in the penstock and feeding pipes are the same in all analysed cases (see Table 2). The rated flow of the pump-turbines for each considered gross head has been obtained by hydraulic similarity (4) from the rated flow and head of the DIE-UP pump-turbine.

$$Q_d = Q_{lab} \left(\frac{H_d}{H_{lab}} \right)^{3/2} \quad (4)$$

Due to the short length of the draft tubes, their dynamics and head losses have been neglected. Therefore, H_{lr} and H_{up} have been assumed equal to zero considering it the reference level (see Fig. 8).

The transient flow in closed conduits is well described by the equations for the conservation of mass and momentum (5) and (6) [13].

$$\rho a^2 \frac{\partial V}{\partial x} + \frac{\partial p}{\partial t} = 0 \quad (5)$$

$$\frac{\partial V}{\partial t} + \frac{1}{\rho} \frac{\partial p}{\partial x} + g \frac{dz}{dx} + \frac{f}{2D} V|V| = 0 \quad (6)$$

Several approaches have been used in the literature to solve (5) and (6). The well-known Method of Characteristics (MoC) is used in this paper to solve (5) and (6), in order to properly capture the propagation of the pressure pulsations developing in the pump-turbine along the PSPP penstock. From the authors' experience, the MoC outperforms the well-known Lumped Parameters approach (LPa) [20–22] for the above purpose (see Appendix).

The MoC approach transforms the water hammer partial

Fig. 7. Power spectrum of pressure pulsations corresponding to (a) the CFD simulation and (b) (1). Frequency values have been expressed in terms of the Strouhal number, S_t [19].

Fig. 8. PSPP scheme.

differential equations (5) and (6) into a system of ordinary differential equations, whose validity is constrained to a set of characteristic curves, by using a linear combination of transient flow equations. Introducing the Lagrange multiplier λ in (5) and (6) results expression (7).

$$\lambda \left\{ \frac{\partial V}{\partial t} + \frac{1}{\rho} \frac{\partial p}{\partial x} + g \frac{dz}{dx} + \frac{f}{2D} V|V| \right\} + \left\{ \rho a^2 \frac{\partial V}{\partial x} + \frac{\partial p}{\partial t} \right\} = 0 \quad (7)$$

Expression (7) can be rewritten as follows

$$\left(\lambda \frac{\partial V}{\partial t} + \rho a^2 \frac{\partial V}{\partial x} \right) + \left(\frac{\lambda}{\rho} \frac{\partial p}{\partial x} + \frac{\partial p}{\partial t} \right) + \lambda g \frac{dz}{dx} + \lambda \frac{f}{2D} V|V| = 0 \quad (8)$$

The first term of equation (8) can be expressed as a full derivative considering constraint (9) while the second term of equation (8) can be expressed as a full derivative considering constraint (10).

$$\lambda \frac{dx}{dt} = \rho a^2 \quad (9)$$

$$\frac{\lambda}{\rho} = \frac{dx}{dt} \quad (10)$$

So to satisfy both constraints (9) and (10) it is need that:

Table 2
PSPP parameters.

	Feeding pipes		Penstock	
Fixed parameters	L_{fp}	50	L_p	2000
	S_{fp}	5	S_p	5.5

$$\lambda = \pm \rho a \quad (11)$$

It is possible to use the piezometric head instead of the pressure by recalling that

$$p = \rho g (H - z) \quad (12)$$

Also note the definition of the wave speed expressed in (13)

$$a = \frac{dx}{dt} \quad (13)$$

Therefore, the system of equations (14) and (15) describes perturbations movements at speeds $\pm a$ along the positive and negative characteristic lines C^+ and C^- [23,24].

$$C^+ : \frac{dV}{dt} + \frac{g}{a} \frac{dH}{dt} + \frac{f}{2D} V|V| = 0 \quad (14)$$

$$C^- : \frac{dV}{dt} - \frac{g}{a} \frac{dH}{dt} + \frac{f}{2D} V|V| = 0 \quad (15)$$

Moreover, applying finite difference solution for the MoC expressions (13), (14), (15) along the positive and negative characteristic lines, shown in Fig. 9, reads as follows.

$$V_{Pi} = \frac{1}{2} \left[(V_{i-1} - V_{i+1}) + \frac{g}{a} (H_{i-1} + H_{i+1}) - \frac{f \Delta t}{2D} (V_{i-1}|V_{i-1}| + V_{i+1}|V_{i+1}|) \right] \quad (16)$$

$$H_{Pi} = \frac{1}{2} \left[\frac{a}{g} (V_{i-1} - V_{i+1}) + (H_{i-1} + H_{i+1}) - \frac{a f \Delta t}{g 2D} (V_{i-1}|V_{i-1}| - V_{i+1}|V_{i+1}|) \right] \quad (17)$$

$$\Delta x = a \Delta t \quad (18)$$

Fig. 9. Positive and negative characteristic lines and Characteristic grid [13].

For solving the governing equations the characteristic grid in the x-t plane has been considered as shown in Fig. 9.

Expressions (16) and (17) are only valid for the interior points of the solution space. The boundary conditions for (16) and (17) are given by the head in the upstream reservoir H_{ur} , which is assumed to be constant, and the one right upstream of the pump-turbine H_{up} , which is given by (24).

In the bifurcation (Fig. 10), flow continuity (19) is required, and neglecting losses, the head at this point must be identical for all branches (20) [25].

$$S_p V_{p,1} = S_{fp1} V_{fp1,N+1} + S_{fp2} V_{fp2,N+1} \quad (19)$$

$$H_{fp1,N+1} = H_{fp2,N+1} = H_{p,1} = H_b \quad (20)$$

The pump-turbine operation curves at steady-state (1) and the sinusoidal functions that reproduce the pressure pulsations developing in the pump-turbine at part load in pumping mode (3) have been up-scaled in (21) and (22) by hydraulic similarity [2], from those previously calibrated against the results of the CFD simulation (see section 3).

$$H_{p,plant} = \alpha^2 \beta^2 A_H + \alpha \beta^{-1} B_H Q_p + \beta^{-4} C_H Q_p^2 \quad (21)$$

$$H_{p,plant}^* = \beta^2 H_{lab}^*(t, n) \quad (22)$$

where β are the hydraulic similarity factors defined in (23)

$$\beta = \sqrt{\frac{H_b}{H_{lab}}} \quad (23)$$

Therefore, (24) represents pump-turbine operation curve at variable speed including pressure pulsations at part load.

$$H'_{p,plant} = H_{p,plant} + H_{p,plant}^* \quad (24)$$

The dominant frequencies of the pressure pulsations in the PSPP simulation model have been assumed to be the same as those used in the simulation model of the DIE-UP test rig, thus assuming that the rotation speed of the pump-turbine is the same. As demonstrated in Ref. [26], similar dimensionless amplitude and frequency of the pressure pulsations can be found both in model and prototype turbines, provided that their rotation speed is the same.

Fig. 10. Bifurcation scheme.

5. Fatigue analysis

Fatigue damages in conduits are mainly caused by variances in tangential stresses in pipe welds [27]. Tangential stress variances in conduits can be obtained as a function of pressure variances $\Delta\sigma_{cp} = f(\Delta P)$ [7]. Tangential stress in conduits is obtained by expression (25) [28].

$$\sigma_{cp} = \frac{PDg}{2e} \quad (25)$$

5.1. Monte Carlo simulation

In order to study if the propagation of pressure pulsations may cause damages in the conduits due to the fatigue phenomenon, a linear reduction from 600 to 550 r.p.m. in both pump turbines in pumping mode has been simulated, considering 25 different temporal random gaps (Δt_0) between the sinusoidal functions reproducing the pressure pulsations in one and another pump-turbine Δt_0 is obtained randomly between 0.00 and 3.50 s, having every simulation a duration of 20 s.

A set of tangential stresses, with different amplitudes, is obtained from each dynamic simulation in the critical point -bifurcation between the penstock and feeding pipes-. In this stress time series, large cycles are interspersed with small cycles. Therefore, it is necessary to use a counting method that breaks the stress history down into individual cycles which can be summed up into a stress range distribution. For high fatigue cycles, the most frequently methods are the Reservoir Counting Method and the Rain Flow Counting Method (RFCM) [29–31]. As RFCM is the most frequently used cycle counting method in the industry, it is used in this paper. For each simulation, the RFCM allows identifying individual stress cycles from a set of variable amplitude tangential stresses. An equivalent stress has been then obtained using (26) [5] from the twenty [32] higher amplitude stresses obtained by RFCM.

$$\sigma_{eq,g} = \sqrt{\frac{\sum_{k=1}^{20} (N_k \sigma_{cp,k}^2)}{\sum_{k=1}^{20} N_k}} \quad (26)$$

Where $N_k = 1$ due to every stress peak occurs only once.

Finally, it is necessary to obtain an equivalent stress for the set of Monte Carlo simulations. This equivalent stress is calculated with a similar expression (27) to the one used previously [5].

$$\sigma'_{eq} = \sqrt{\frac{\sum_{g=1}^{25} (N_g \sigma_{eq,g}^2)}{\sum_{g=1}^{25} N_g}} \quad (27)$$

Where $N_g = 1$ due to every temporal gap has been assigned the same probability.

5.2. Results

The number of stress cycles that may cause fatigue damage in the PSPP penstock are obtained for each equivalent stress σ'_{eq} by entering the fatigue curves (Fig. 11) defined in Ref. [32], considering the worst category of welding due to possible imperfections that may take place in some areas of the pipe, as suggested in Ref. [27].

The fatigue analysis was carried out considering different pumped heads. Table 3 summarises the maximum number of stress cycles calculated for each σ'_{eq} to avoid fatigue damages.

The maximum number of stress cycles shown in Table 3 has been expressed in Fig. 12, in terms of the maximum hours per year

Fig. 11. Fatigue resistance curves for normal stress variations [32].

the PSPP can operate at partial load in pumping mode, considering different power plant lifetimes. Note that it is widely accepted in the technical literature a lifetime of PSPP between 60 years (the reader is referred to [33] or [34], for instance) and 100 years (see for example [35] or [36]).

For this purpose, it has been considered that σ'_{eq} occurs every 20 s, which matches the simulation time. Therefore, in an hour in which the PSPP operates at partial load in pumping mode, there would be 180 stress cycles of σ'_{eq} MPa.

The maximum number of hours per year at partial load in pumping mode presented in Fig. 12 is compared next to the number of hours per year that i) real PSPPs that are currently operating in the Spanish electricity system, are operated in the secondary regulation service in generating mode, and ii) theoretical PSPPs whose optimal operation has been simulated, are operated at partial load in pumping mode. Note that the real/theoretical PSPPs are equipped with fixed-speed/variable-speed pump-turbine units; therefore, they cannot/can be operated at partial load in pumping mode, respectively. The main goal of the comparison is to discuss if either real PSPPs or theoretical PSPPs are exposed or not to fatigue damage in the penstock, according to the results shown in Fig. 12.

The number of hours per year that each real PSPP operates in the secondary regulation service in generating mode in the period 2010–2015, is presented in Table 4. They are referred to as PSPPs 1–7. Although the hourly schedules of the real PSPPs are publicly available in the information system of the Spanish transmission system operator, the authors prefer not to reveal their names. The number of hours per year in the secondary regulation service in generating mode of the real PSPPs presented in Table 4 corresponds to the number of hours in which the PSPPs were assigned some reserve in the secondary regulation reserve market.¹ Note that a real PSPP of the Spanish market cannot participate in the said

market in pumping mode.

There are several PSPPs in the Spanish system that barely operate in the secondary regulation reserve market: PSPPs 2–3 and 5–7. The main reason for this is the operation strategy of each PSPP, having the possibility of operating in many markets, not only in the secondary regulation reserve market. However, PSPPs 1 and 4 have been operated in the reserve market between 2088–3153 and 438–2579 h per year in the period 2010–2015 (see Table 4), respectively. If a similar operation strategy was followed in pumping mode, the fatigue damage in the penstock would not be discarded. It is important to remark that the gross head of the real PSPPs 1 and 4 are 345 and 980 m, respectively.

The number of hours per year that each theoretical PSPP operates in pumping mode in the secondary regulation reserve market of the Spanish electricity system (hSM) in the period 2013–2015 is presented in Table 5. It is also shown the number of hours out of hSM in which the PSPPs are required by the transmission system operator to be operated at partial load in pumping mode (hSM-pmin), i.e., between the minimum power pmin and pmin+0.15 (pmax-pmin). The theoretical PSPPs are referred to as A, B and C-PSPPs, and come from unpublished results from the authors in collaboration with General Electric, obtained with the optimization model presented [12]. The A/B/C-PSPPs are equipped with two variable-speed pump-turbine units, and can be operated in pumping mode between 59%/59%/82% and 100% of the maximum power, respectively.

The results presented in Table 5 are consistent with the operating experience in the Swiss variable-speed PSPP of Grimsel II (head of 400 m), which was operated from May 2013 to March 2014 during 3500 h as a controllable load according to [37], or the German variable-speed PSPP of Goldisthal (head of 302 m), which is operated since October 2004 for 19 h per day for frequency-control purposes in generating and pumping mode according to [38] (assuming 9 h per day in pumping mode, it means 3285 h per year).

Besides, the theoretical PSPPs are operated between 676 and 1221 h per year at partial load in pumping mode (close to the minimum power) according to the assumptions made in Ref. [12] with respect to the percentages of the real-time use of the committed secondary regulation reserves. These results preliminary confirm that fatigue damages in the penstock of PSPPs equipped with variable-speed pump-turbine units are not likely to

¹ In the secondary regulation reserve market of the Spanish electricity system, the market participants have to fulfil a ratio between the upward and the total reserve. This ratio is typically 45–60%. Besides, the committed reserves must be ready to be used in real time by the transmission system operator, what can lead the plants to operate close to the minimum or the maximum power. However, the percentages of the real-time use of the committed reserves are not publicly available in the Spanish system.

Table 3
Maximum number of cycles to avoid fatigue damages.

Gross Head [m]	σ'_{eq} [MPa]	Maximum cycles
500	13.89	∞^a
550	16.09	60,865,932
575	16.46	54,317,643
600	17.66	38,195,188
625	19.14	25,541,735
650	20.47	18,285,665
675	20.67	17,413,917
700	22.52	11,322,613

^a For $H_b = 500$ m is lower than the minimum stress (14,57 MPa) to consider fatigue damages.

occur according to Fig. 12 (the net head of the A, B and C-PSPPs is 253 m). Note that the procurement of the secondary regulation service is different depending on the power system. Therefore, the number of hours per year here presented (hSM and hSM-pmin) could change in other power systems different from the Spanish one. It is important to remark that the results shown in Table 5 for theoretical PSPPs are obtained considering only the participation in the day-ahead energy market and in the secondary regulation service of the Spanish electricity market whereas the real PSPPs also participate, among others, in the intraday market or the tertiary regulation service.

6. Conclusions

The risk of fatigue damage in the penstock of a variable-speed pumped-storage power plant due to the propagation of the pressure pulsations originated in the pump-turbine when it operates in pumping mode at partial load has been analysed in this paper by means of a Monte Carlo simulation-based approach. A dynamic simulation model of the PSPP has been developed for the said purpose. The PSPP conduits have been modeled by using the Characteristics Method. The pump-turbine has been modeled through its steady-state operation curves (head-flow-rotation speed) and a sinusoidal function to reproduce the pressure pulsations developing in the pump-turbine runner at partial load. Both the operation curves and the sinusoidal function were up-scaled by hydraulic similarity from those obtained from the results of a CFD simulation carried out on a reversible pump-turbine model installed in the University of Padova.

The Monte Carlo simulations were carried out considering different random temporal gaps in the sinusoidal function that

Fig. 12. Maximum number of hours per year in pumping mode at partial load to avoid fatigue damage for different gross heads and power plant lifetimes.

Table 4
Number of hours per year that the real PSPPs 1–7 were assigned secondary regulation reserve in generating mode.

PSPP	2010	2011	2012	2013	2014	2015
1	2356	2088	2758	2884	3153	2765
2	312	100	387	229	410	262
3	366	137	470	344	470	306
4	827	438	812	1253	2306	2579
5	0	0	0	0	0	0
6	101	33	108	49	104	112
7	0	0	18	0	58	0

Table 5
Number of hours per year that the theoretical PSPPs operate in pumping mode in the secondary regulation reserve market (hSM) and at partial load in pumping mode (hSM-pmin), i.e., between the minimum power pmin and $pmin + 0.15 \cdot (pmax - pmin)$.

	PSPP	2013	2014	2015
hSM	A	3902	3736	3363
	B	4370	4128	3758
	C	3652	3497	3163
hSM-pmin	A	1086	1043	676
	B	1221	1138	792
	C	1088	1025	722

reproduces the pressure pulsations, as well as different power plant gross heads.

An equivalent tangential stress has been obtained from each set of Monte Carlo simulations (i.e. for each gross head) by the RFCM. The maximum number of stress cycles to avoid fatigue damage in the penstock has been computed for each considered gross head, by entering the fatigue curves of the Eurocode of Steel Structures with the corresponding equivalent tangential stress, and has been later translated into the maximum number of hours per year the PSPP can operate in pumping mode at partial load to avoid fatigue damage in the penstock for practical purposes.

By comparing the obtained maximum number of hours per year the PSPP can operate in pumping mode at partial load with the hours per year that real fixed-speed PSPPs and theoretical variable-speed PSPPs participate in the secondary regulation and operate at partial load in pumping mode, respectively, it can be concluded that fatigue damages are not likely to occur due to the participation of the PSPP in the secondary regulation service of the Spanish electric power system. Further research is necessary to accept or reject this conclusion in other power systems.

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Appendix

The LPa has been checked by using expressions (28) and (29) [20–22].

Fig. 13. Comparison between MoC and LPA.

$$\frac{dq_{p,i}}{dt} = \frac{n_t + 1}{T_w} \left(h_i - h_{i+1} - \frac{r}{2(n_t + 1)} q_{p,i} |q_{ip,i}| \right) \quad (28)$$

$$\frac{dh_i}{dt} = n_t \frac{T_w}{T_e^2} (q_{p,i} - q_{p,i+1}) \quad (29)$$

This method allows to obtain pressure value in each elastic element introduced along the conduit.

As it shown in Fig. 13, LPA acts as a filter of high frequency pressure signals neglecting their effects, while MoC is able to transmit these signals. Therefore, MoC is closer to reality than LPA and it has been selected to model water hammer effects.

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