Provision of Primary Frequency Control with Variable-Speed Pumped-Storage Hydropower

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***Abstract*—The development of renewable and intermittent power generation creates incentives for the development of both energy storage solutions and more flexible power generation assets. Pumped-storage hydropower (PSH) is the most established and mature energy storage technology, but recent developments in power electronics have created a renewed interest by providing PSH units with a variable-speed feature, thereby increasing their flexibility. This paper reviews technical considerations related to variable-speed pumped-storage hydropower (PSH) in link with the provision of primary frequency control, also referred to as frequency containment reserves (FCRs). Based on the detailed characteristics of a scale model pump-turbine, the variable-speed operation ranges in pump and turbine modes are precisely assessed and the implications for the provision of FCRs are highlighted. Modelling and control for power system studies are discussed, both for fixed- and variable-speed machines and simulation results are provided to illustrate the high dynamic capabilities of variable-speed PSH.**

***Index Terms*—Frequency Containment Reserves, Hydraulic machines, Power Conversion, Pumped-storage Hydropower.**

I. Introduction

This is a fact, today's and future's power systems will have to deal with more and more renewable energy sources. As the energy transition goes on, more intermittent energy sources will be connected to the grid, resulting in challenges for keeping its balance, both in short and medium terms (from seconds to hours). Pumped-storage hydropower (PSH) is the most effective technology for keeping this balance between supply and demand of electricity, which explains why 99 % of grid-connected energy storage worldwide are of this type [1]. In Europe, PSH has still a big development potential since its capacity could be increased by up to 10 times the current capacity [2].

Up to now, PSH has mainly consisted in large fixed-speed reversible pump-turbines driving or being driven by a grid-connected synchronous machine. The power in turbine mode is varied at the expense of efficiency by controlling the flow in the machine while the power is fixed in pump mode. The best efficiency point (BEP) corresponds to different speeds in pump and turbine modes, resulting in a loss in efficiency as the machine is mainly operated at one single constant speed, the synchronous speed.

In the last decades, developments in power electronics have enabled the supply of electrical machines with variable-frequency voltages, resulting in the possibility to vary the speed of PSH plants. Large machines, of more than 50 MW, use a doubly-fed induction machine (DFIM) with a power converter rated to only a few percent of the nominal power while smaller machines use a synchronous machine with a full size power converter [3]–[6].

This variable speed possibility can be used to always operate the hydraulic machine at its BEPs, as these are related to different speeds in pump and turbine modes, thereby increasing the revenues from price arbitrage on the energy markets. Sometimes the variable speed becomes a necessity in order for the pump mode to support high head variations and be able to operate between its stability and cavitation limits [3]. Besides, the increased head range can lead to a better use of the basins' capacity. Another option is to make use of the variable speed to provide transmission system operators (TSOs) with ancillary services, in particular primary and secondary frequency control, both in pump and turbine modes. The basic idea behind those services is to adjust power injections/offtakes in real-time, so as to keep the balance between electricity generation and consumption. The provision of primary and secondary frequency control imposes high technical requirements on power plants, as these must have a range of power within which injections/offtakes can continuously and rapidly be varied.

Several authors have emphasized that a significant share of PSH revenues could be achieved through the provision of ancillary services [7]–[9]. While these studies aim to give an overview of PSH, they fail to provide important technical details regarding the variable-speed operation of PSH to provide ancillary services. Economic calculus is often based on rather vague assumptions regarding the operation range of PSH, e.g. from 20 to 100 % of the nominal power in turbine mode, and from 60 to 100 % in pump mode if variable-speed operation is considered in both cases [4].

It is also noticed that the control of variable-speed PSH units and their dynamic capabilities to provide primary and secondary frequency control are not well known. This is not

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surprising since hydro turbine models and control schemes traditionally used in power system [10]–[14] studies have become inappropriate for variable-speed units.

This paper aims to clarify the technical capabilities of variable-speed PSH, and to put in light state of the art control schemes and transient modelling for power system studies. The outline of the paper is as follows. Section II discusses primary frequency control, its interest for PSH and the related technical requirements. Based on the characteristics of a real pump-turbine, Section III discusses the operation range of PSH. Section IV is dedicated to the transient modelling and control of PSH for power system studies. Simulation results put in light the dynamic capabilities of PSH for the provision of primary frequency control. The conclusions of the paper are given in Section V.

II. Frequency Containment Reserves

FCRs are the first ancillary service to be activated in order to support the grid frequency in Continental Europe (CE). The provision of FCRs requires to be able to continuously vary power injections/offtakes, within a predefined range. Power set points are proportional to local measurements of frequency deviations. FCR providers must saturate at frequency deviations of ± 0.2 Hz and they must be able to fully activate in 30 seconds. A frequency deadband of ± 0.01 Hz with no expected reaction is accepted. The largest market for FCRs includes Germany, Switzerland, Austria, the Netherlands and Belgium. It is organized as weekly tenders and currently represents more than 800 MW [15], i.e. more than 25 % of the total FCR requirement for the CE grid. During the 52 weeks between early November 2015 and end October 2016, the average capacity and average marginal capacity prices have respectively been 15.73 €/MW/h and 17.14 €/MW/h. Given these average weekly prices, it may seem a priori more lucrative for an energy storage to provide FCRs than to do price arbitrage. For each megawatt of FCR, the storage can earn at least more than 15 €/MW/h while arbitrage requires price spreads of at least 30 € to make the same revenue. However, providing FCRs leads to technical challenges since the power output should constantly be varied.

III. Operation range of variable-speed PSH

The detailed characteristics of a Francis pump-turbine are given in [16]. These characteristics are obtained from laboratory measurements on the scale model of a real machine by varying the head H [m] across the machine, for different wicket gate positions $y \in [0; 1]$, while the rotational speed N [RPM] is kept constant. Both discharge Q [m^3/s] and torque T [N m] are measured, which gives a vector (y,N,H,Q,T) for each measurement. The measurements are then provided as charts giving the relations between the unit numbers[1], namely unit speed N_{11}, unit flow Q_{11} and unit torque T_{11}, with D_{ref} [m] the pump-turbine's reference diameter [17]:

$$N_{11} = \frac{N\, D_{ref}}{\sqrt{H}} \quad Q_{11} = \frac{Q}{\sqrt{H}\, D_{ref}^2} \quad T_{11} = \frac{T}{H\, D_{ref}^3}. \tag{1}$$

The operation range of the studied turbomachinery are provided by the same reference [16], and indicated on the T_{11}-N_{11} charts, as shown in Fig. 1. In turbine mode, the limitation of the operation range is mainly due to several flow phenomena taking place in the hydraulic machine [18]. The A-B and C-D limits are defined by the manufacturer in order to avoid discharge ring swirl cavitation. Indeed, at high and low discharge, a swirling flow appears at the runner outlet, with eventually cavitation at its center, in which case the phenomenon is referred to as vortex rope. The whirl dynamics compromises the machine's operation stability, since it is the main source of pressure fluctuations in the hydraulic installation. The B-C and D-E limits are set to respectively avoid leading edge pressure- and suction-side cavitation as it can lead to a severe erosion of the blades. Finally, the A-E limit is set by the rating of the electrical generator. In pump mode, the operation range is limited by lower and upper N_{11} bounds, i.e. higher and lower head values, to respectively prevent suction- and pressure-side cavitation on the runner blade [6], [18]. In addition, the operation range is limited by another upper N_{11} bound, corresponding to the pump's stability limit [6].

The power P_m [MW] associated to each measurement can be derived from the unit numbers using the following relation:

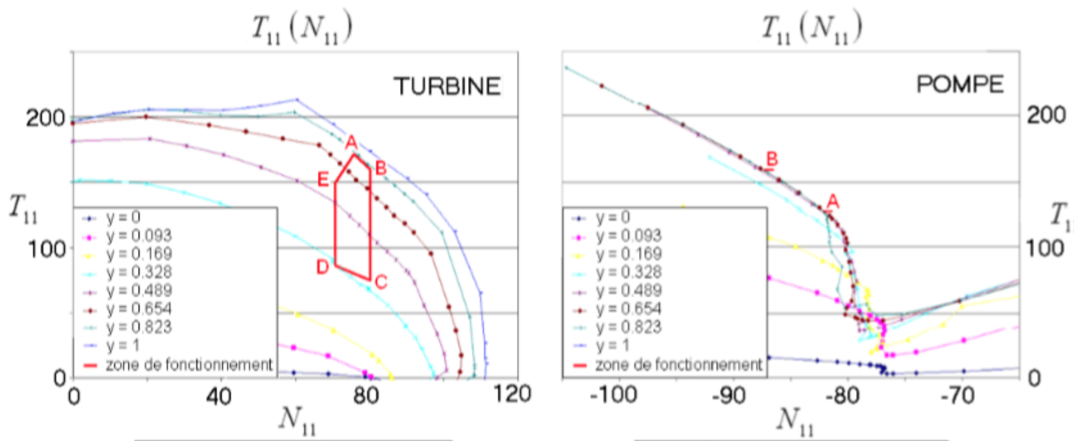
$$P_m = \frac{2\,\pi\, N\, T}{60} = \frac{\pi}{30}\, H^{3/2}\, D_{ref}^2\, N_{11}\, T_{11}. \tag{2}$$

It should be emphasized that given N_{11}, Q_{11} and T_{11} numbers, e.g. those associated with the BEP or the operation limits, one has just to set two variables among D_{ref}, N, Q, H, T and P_m in order to define a pump-turbine, i.e. to derive its diameter and operation ranges in terms of head, flow and power. Given a nominal rotational speed of 675 RPM, and a maximum mechanical power of 2 MW in pump mode, one can derive the corresponding head and reference diameter, 93.04 [m] and 1.2413 [m] respectively. The operation ranges associated with this fictitious case are indicated in Fig. 2. The grey area corresponds to the case where the rotational speed can be varied from 0 to 100 % of its nominal value, while the segments in red indicate the range if the speed is kept at the nominal value. We see from Fig. 2 that at given head, the variable-speed feature provides the pump mode with a non-zero power range, while the turbine mode does not see its power range affected. The variable speed enables both pump and turbine modes to operate at lower heads. Besides, the extent of the power range varies with head, both in pump and turbine modes, and can precisely be quantified using 2 and the ranges of Fig. 1, i.e. substituting in 2 the extreme values that can be taken by N_{11} and T_{11}.

Following the mentioned methodology for the pump mode, the allowed power range range can be derived as a ratio:

$$\frac{P_{m,A}}{P_{m,B}} = \frac{N_{11A}\, T_{11A}}{N_{11B}\, T_{11B}} = 0.76.$$

[1]The advantage of providing the characteristics of the studied turbomachinery using unit numbers is the following. If one can make the similitude hypothesis, i.e. if one can neglect the effects of both the Reynolds number and roughness, then the provided characteristics can easily be extended to (i) the same machine operated at other working conditions, and (ii) to machines having the same geometry as the scale model but a diferent size.



Point	N_{11}	T_{11}
A	75.6481569	171.685041
B	80.1760764	159.858624
C	80.1760764	74.7710555
D	70.6751414	86.1941526
E	70.6751414	149.854271

Point	N_{11}	T_{11}
A	-81.7429833	127.80629
B	-86.8688621	158.9985

Fig. 1: Characteristics and operation ranges of the studied pump-turbine, as found in [16].

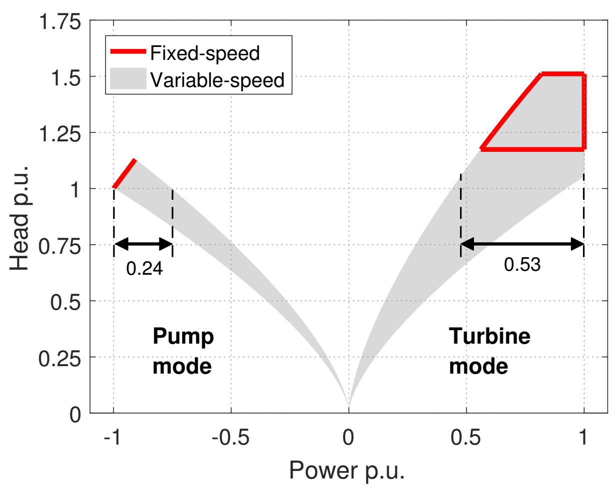


Fig. 2: Implied operation ranges of the fictitious PSH case given the operation ranges of Fig. 1.

This relation shows that given a head, the allowed power range is 24 % of the maximum allowed power at that head. It is important to emphasize that the maximum allowed power depends on the available head. The lower the available head, the lower the maximum allowed power. The same computation can be carried out for the turbine mode, which gives the following power range:

$$\frac{P_{m,C}}{P_{m,B}} = \frac{N_{11C}\,T_{11C}}{N_{11B}\,T_{11B}} = 0.47.$$

Here again, the minimum allowed power at a given head is a fixed percentage of the maximum allowed power at that head.

To sum up, given a head, the range of allowed power of the studied pump-turbine is respectively 24 % and 53 % of the maximum allowed power at the considered head. These ranges are limited to prevent undesired flow phenomena. The fact that the power range varies with head is crucial for the provision of FCRs. Indeed, as the level of water varies with time, so does the available power range and the amount of FCRs that can be provided. Whether in pump or turbine mode, the amount of FCRs that can be provided by a pump-turbine should be computed based on the minimum operation head. Another important consideration is that variable-speed PSH must be operating rather far from the zero output to be able to provide FCRs, which implies frequent switches from pump to turbine, and vice versa, in order to continuously provide this ancillary service. The energy market implications of this last point should be taken into account when assessing the revenues of variable-speed PSH for the provision of FCRs.

IV. Transient Modelling of PSH for Power System Studies

A. Hydraulic model

Various models can be used to study transients in hydropower plants. Although most of them are dedicated to hydraulic turbines, those models can be adapted/generalized to pumps and pump-turbines. They most often consist in the combination of a model for the water column, in the penstock, with the characteristics of the turbomachinery, which can be with or without guide vanes. The water column is generally considered either elastic or inelastic and is modelled, in the former case, through a first-order dynamic model. The characteristics of the hydraulic turbine is static, i.e. non-dynamic, meaning that it does not consist in differential equations.

The characteristics can be represented through two classes of models. On the one hand, several models, linear or non-linear, have been developed for small to medium deviations around an operating point [10]–[14]. Those models target hydraulic machines operated at fixed or nearly fixed speed, i.e. those connected to the grid through a synchronous machine with no power electronics converter. In the second class of models, the turbine is considered through its detailed characteristics [16], [19], [20], i.e. interpolation curves joining various operation points, as in Fig. 1. Those models are suited for the study of large variations in operating conditions, including variations in rotational speed.

Modelling the water column and coupling it to the characteristics of the turbomachinery enables to account for water hammer phenomena, as a result of variations in guide vane position and/or rotational speed, which in turn lead to undesired mechanical power variations. The water column is considered in [16], [19] but ignored in [20].

B. Control scheme

Reference [10] describes a governor for fixed-speed hydraulic machines, enabling the control of power thanks to the wicket-gate position Y. The control implements a droop characteristics whose stability is ensured by including a low transient droop. This scheme should obviously be different for variable-speed machines. In pump mode, power should be controlled, thanks to the variable speed, while in turbine mode, power can be controlled through rotational speed and/or wicket gate position. In [21], the power in turbine mode is controlled through the wicket gate position Y while the variable speed enables to completely dampen power oscillations normally resulting from changes in Y. In [16], two control strategies are proposed in turbine mode. The first strategy consists in controlling the power thanks to the wicket gate position, while the control of speed enables to optimize the efficiency. The second strategy consists in controlling the power through the electromagnetic torque while varying the wicket gate position to optimize the efficiency through the rotational speed. It is shown that the second strategy, which goes one step further than [21] through the efficiency optimisation, is by far superior to the first one in the sense that it implies a much faster dynamic response. This strategy is definitely the best suited for the provision of ancillary services. Whether it is in [16] or in [21], no information is given on the followed methodology for the tuning of the speed controller.

In order to design the control in speed of the hydraulic machine, in turbine mode, the transfer functions linking the guide vanes position to the rotational speed have to be established. As explained in the previous sections, the detailed characteristics of hydraulic turbomachinery are needed in order to carry out simulations at variable speed. Such characteristics do obviously not come as transfer functions, but instead as grids of operation points from which intermediate values are interpolated, as in Fig. 1. An approximate transfer function is therefore needed in order to design a speed controller. Besides, linear transfer functions are needed in order to be able to use linear control theory to design the speed controller.

For an ideal lossless turbine and using Laplace's formalism, the transfer function linking the mechanical torque and power to the wicket gate position is given by:

$$\frac{\Delta \bar{T}_m}{\Delta \bar{Y}} \approx \frac{\Delta \bar{P}_m}{\Delta \bar{Y}} = \frac{1 - T_w\, s}{1 + \dfrac{T_w}{2}\, s}. \tag{3}$$

with T_m the mechanical torque, P_m the mechanical power, T_w the water time constant, Y the wicket gate position, $\Delta \bar{X}$ denoting $(X - X_0)/X_0$ and subscript 0 referring to initial steady-state values. The mechanical torque can then be linked to the electromagnetic torque T_{em} and the turbine's moment of inertia J and angular speed ω:

$$T_m - T_{em} = J\,\frac{d\omega}{dt}.$$

Using Laplace's formalism in addition to linearising and normalising with respect to an operating point (T_0, ω_0) gives

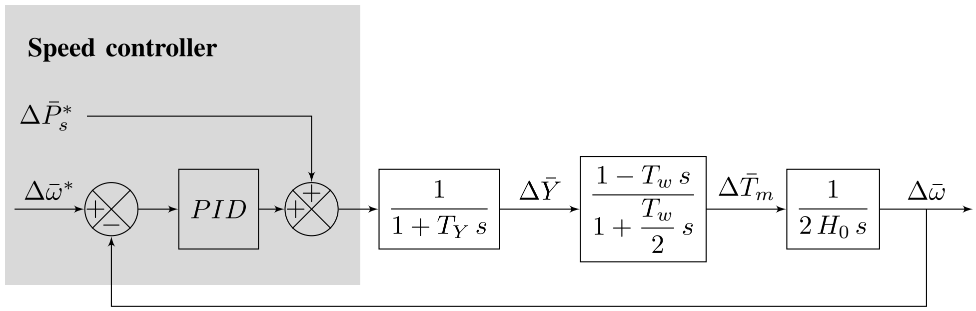
$$\frac{\Delta \bar{\omega}}{\Delta \bar{T}_m - \Delta \bar{T}_{em}} = \frac{1}{2\, H_0\, s}. \tag{4}$$

In addition to 3 and 4, and as in [10], a first-order transfer function with unit gain and time constant T_Y is considered to model the dynamics of the guide vanes. We have thus a set of transfer functions linking the rotational speed to the signal controlling the guide vanes position. From these transfer functions, one can establish the control scheme of Fig. 3a. This scheme includes an upper branch feeding the normalized deviations in electrical power reference, $\Delta \bar{P}_s^*$, in order to account for the fact that variations in power and rotational speed references occur simultaneously. Indeed, the higher the required electrical power, the higher the optimal rotational speed. As a result of an increase in power reference, the electromagnetic torque adjusts rapidly, which leads to a decreasing rotational speed while the new optimal speed increases.

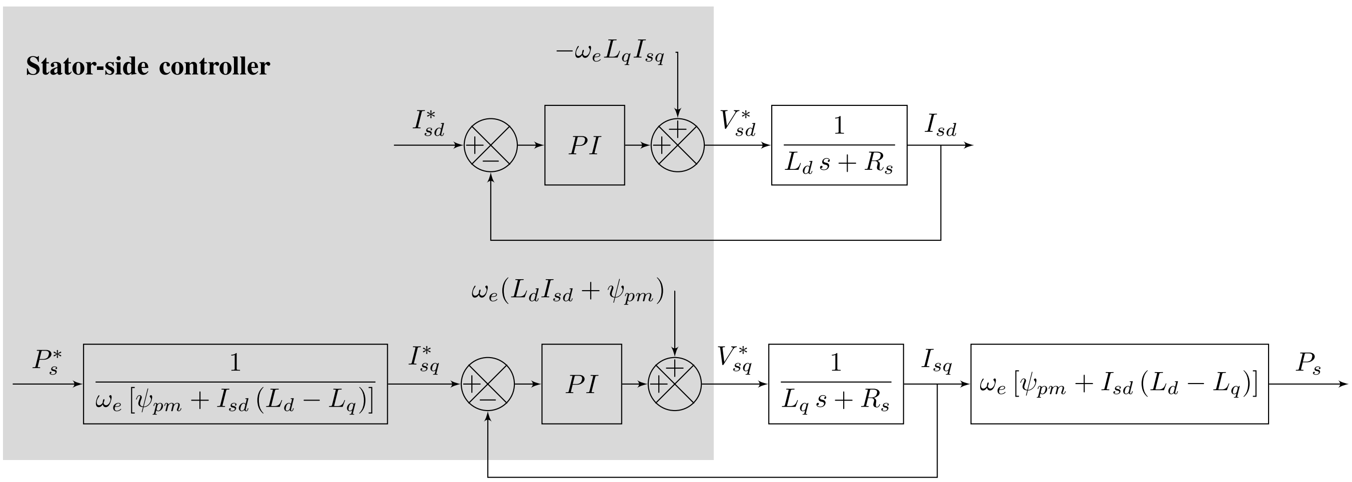
The power controller is derived from the equations describing the dynamics of the electrical machine. For a permanent magnet synchronous machine (PMSM), the resulting control scheme is indicated in Fig. 3b.

C. Simulation results

A PSH case has been set up in *MATLAB/Simulink* and the *Simscape* library in order to illustrate the dynamic capabilities brought by the variable speed. The hydraulic model consists of a rigid water column coupled to the pump-turbine characteristics of Fig. 1. On the electrical side, a PMSM is fed by two two-level voltage source inverters (VSIs) connected in back-to-back configuration with a DC link. The power converter is connected to an infinite medium-voltage bus through a short distribution line. The goal of the simulation is to observe the dynamics of the whole system while the PSH unit is providing FCRs with the control schemes of Fig. 3. Simulation results are given in Fig. 4, for the turbine and the pump modes. Based on the requirements imposed by the Belgian TSO, an



(a) In turbine mode, the rotational speed is controlled through the wicket gate position.



(b) The power is controlled through the electromagnetic torque.

Fig. 3: Block diagram for the control of speed and power.

active power reference is derived from frequency deviation measurements, the latter being in this case 2013 data at ten-second time steps. Based on frequency data indicated in Fig. 4, FCRs are provided for an amount of power equivalent to 20 % and 12.5 % of the pump rated power, respectively in turbine and pump modes. In both operation modes, the results show that power injections can effectively be varied to follow a power reference for the provision of FCRs. Mismatches between power injection and power reference are around two percentage points and are explained by the losses in the electrical machine, the power converter and the distribution line. In turbine mode, while power is controlled through the power converter, the control of the guide vanes enables to vary the rotational speed in order to follow the optimal speed reference. Variations in guide vanes position induce water hammer effects. In pump mode, the guide vanes are kept fixed while power is controlled through the power converter. Water hammer effects are reduced as compared to the turbine mode.

V. Conclusion

Recent developments in power electronics have brought the possibility to operate PSH units at variable speed. This feature enables to operate hydraulic machines over wider head ranges. For a given head, the variable speed allows power variation in pumping, while a unique power is possible at fixed speed, but leaves the power range unchanged in turbine mode. Variable-speed PSH is thus able to provide FCRs both in pump and turbine modes. Operation ranges are limited to prevent undesired flow phenomena. Given a head, the studied pump-turbine's allowed power range spreads 24 % and 53 % of and from the maximum allowed power at the given head. Whether it is in one or the other mode, the allowed power range varies with head, which impacts the amount of FCRs that can be provided. Besides, the continuous provision of FCRs requires PSH units to constantly be operating rather far from the zero output, which implies frequent switches from pump to turbine mode and vice versa. This has energy market consequences that should be taken into account when assessing the revenues of variable-speed PSH providing FCRs.

Modelling and control for power systems studies have been discussed, both for fixed- and variable-speed machines. A variable-speed PSH model has been set up based on the detailed characteristics of a Francis pump-turbine, a rigid water column, the dynamic model of a PMSM, a short impedance line and two two-level back-to-back VSIs. In pump as in turbine mode, power control has been achieved through the VSIs. The guide vanes position has been kept fixed in pump mode while its adjustment enabled speed control in turbine mode. Simulation results have shown the high dynamic performances of variable-speed PSH, in line with FCR requirements.

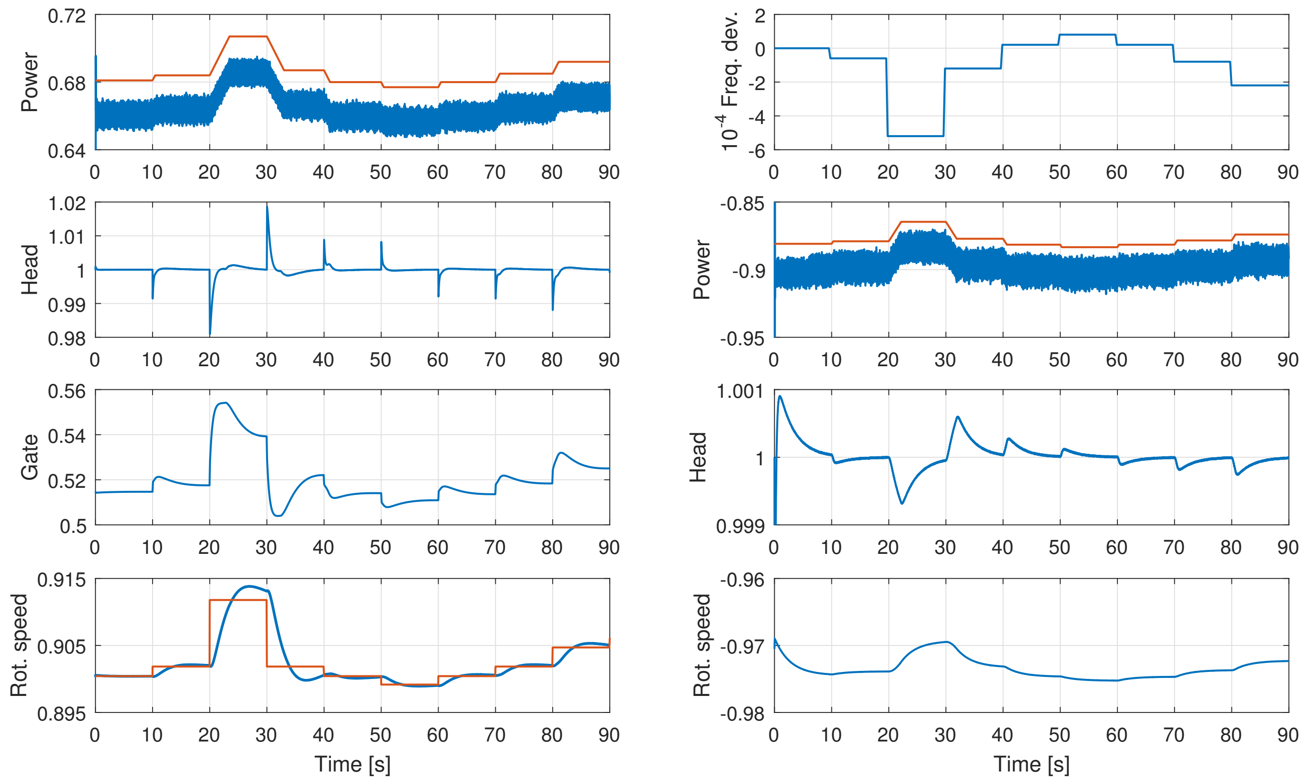


Fig. 4: Transient operation of variable-speed PSH, in turbine (left) and pump (right) modes, for the provision of FCRs. All values are given in per unit. Red curves are reference signals.

References

[1] I. E. Agency, "Technology roadmap: Energy storage," IEA, Tech. Rep., 2014.

[2] M. Gimeno-Gutiérrez and R. Lacal-Arántegui, "Assessment of the european potential for pumped hydropower energy storage," *A GIS-based assessment of pumped hydropower storage potential. EUR, Scientific and technical research series*, vol. 25940, 2013.

[3] G. Ardizzon, G. Cavazzini, and G. Pavesi, "A new generation of small hydro and pumped-hydro power plants: advances and future challenges," *Renewable and Sustainable Energy Reviews*, vol. 31, pp. 746–761, 2014.

[4] D. Beevers, L. Branchini, V. Orlandini, A. De Pascale, and H. Perez-Blanco, "Pumped hydro storage plants with improved operational flexibility using constant speed francis runners," *Applied Energy*, vol. 137, pp. 629–637, 2015.

[5] G. D. Ciocan, O. Teller, and F. Czerwinski, "Variable speed pump-turbines technology," *UPB Scientific Bulletin, Series D: Mechanical Engineering*, no. 74, pp. 33–42, 2012.

[6] K. Jürgen, K. Helmut, and S. Manfred, "Small and mid-size pump-turbines with variable speed," *Energy and Power Engineering*, vol. 2013, 2013.

[7] J. I. Pérez-Díaz, M. Chazarra, J. García-González, G. Cavazzini, and A. Stoppato, "Trends and challenges in the operation of pumped-storage hydropower plants," *Renewable and Sustainable Energy Reviews*, vol. 44, pp. 767–784, 2015.

[8] J. Pinto, J. de Sousa, and M. V. Neves, "The value of a pumping-hydro generator in a system with increasing integration of wind power," in *2011 8th International Conference on the European Energy Market (EEM)*. IEEE, 2011, pp. 306–311.

[9] R. Deb, "Operating hydroelectric plants and pumped storage units in a competitive environment," *The Electricity Journal*, vol. 13, no. 3, pp. 24–32, 2000.

[10] P. Kundur, N. J. Balu, and M. G. Lauby, *Power system stability and control*. McGraw-hill New York, 1994, vol. 7.

[11] F. Demello, R. Koessler, J. Agee, P. Anderson, J. Doudna, J. Fish, P. Hamm, P. Kundur, D. Lee, G. Rogers *et al.*, "Hydraulic-turbine and turbine control-models for system dynamic studies," *IEEE Trans. Power Syst.*, vol. 7, no. 1, pp. 167–179, 1992.

[12] G. A. Munoz-Hernandez, D. I. Jones *et al.*, *Modelling and controlling hydropower plants*. Springer Science & Business Media, 2012.

[13] W. G. on Prime Mover and E. S. M. for System Dynamic Performance Studies, "Hydraulic turbine and turbine control models for system dynamic studies," *IEEE Trans. Power Syst.*, vol. 7, no. 1, 1992.

[14] E. De Jaeger, N. Janssens, B. Malfliet, and F. Van De Meulebroeke, "Hydro turbine model for system dynamic studies," *IEEE Trans. Power Syst.*, vol. 9, no. 4, pp. 1709–1715, 1994.

[15] www.regelleistung.net, 2015.

[16] Y. Pannatier, "Optimisation des stratégies de réglage d'une installation de pompage-turbinage à vitesse variable," Ph.D. dissertation, École Polytechnique Fédérale de Lausanne, 2010.

[17] A. L. Jaumotte and G. RIOLLET, *Caractéristiques et similtude des turbomachines hydrauliques*. TI, 1991.

[18] F. Avellan, "Introduction to cavitation in hydraulic machinery," in *6th International Conference on Hydraulic Machinery and Hydrodynamics, Timisoara, Romania, October*, 2004, pp. 21–22.

[19] M. H. Chaudhry, *Applied hydraulic transients*. Springer, 1979.

[20] L. Belhadji, S. Bacha, I. Munteanu, A. Rumeau, and D. Roye, "Adaptive mppt applied to variable-speed microhydropower plant," *IEEE Trans. Power Syst.*, vol. 28, no. 1, pp. 34–43, 2013.

[21] J.-K. Lung, Y. Lu, W.-L. Hung, and W.-S. Kao, "Modeling and dynamic simulations of doubly fed adjustable-speed pumped storage units," *IEEE Trans. Energy Convers.*, vol. 22, no. 2, pp. 250–258, 2007.