Sustainable Hydraulics in the Era of Global Change – Erpicum et al. (Eds.)

Pumped hydroelectric energy storage: A comparison of turbomachinery configurations

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ABSTRACT: The technical and economic feasibility of Pumped Hydroelectric Energy Storage (PHES) is increasingly under discussion. Its important capital cost may be mitigated by its relevance in balancing the electrical grid. Indeed renewable energy intermittency forces electric power industry to find new solutions for the regulation of the electric grid. PHES systems are built around hydraulic and electrical machines. The site head and the flow rate primarily define the type of hydraulic machine used. In this paper three configurations are discussed: traditional applications that include separate hydraulic machines (turbines and pumps), Reversible Pump Turbines (RPTs) and Pumps as Turbines (PATs) that can operate in both modes of operation. A detailed estimation of the PATs performance is computed to predict and interpret their behaviour during reversed operation: therefore different prediction models are listed and analyzed. A case study compares different system efficiencies and capacity for different configurations. This PHES consist in a unused mine with a depth of 350 meters and it is equipped with one or more Francis turbine or with PAT unit. The available head heavily varies with interittency due to the peculiar mine configuration. Consequently the hydraulic machines adopted could often work in off-design conditions. Results show that regarding micro and small power plants the solution using PATs is an available and suitable option.

1 INTRODUCTION

In the past decade, the concerns about world environmental issues produced by fossil fuel exploitation have increased. The use of renewable energy sources aims to reduce the dependence on thermal power plants, that nowadays satisfy most of the global energy demand. Among all the technologies usable for this goal, hydro-power is by far the most used in electric energy production.

Although hydro-power brings up several problems linked to its social and environmental implications, such as disturbing wildlife and biodiversity or altering landscapes, it has several important positive effects. It maintains a stable price of energy that is very dependent on political and geographical questions (as the crude oil or the gas market); it is a long-term investment and guarantees low emissions for all the working duration with a relatively high efficiency (Barnes 2011). In such context, the Pumped Hydroelectric Energy Storage (PHES) finds its role as the most mature technology regarding energy storage (Deane 2010). PHES systems obviously have many similarities to conventional hydro-power plants but differ by the fact that the flow is bidirectional. A PHES unit exploits the potential energy stored in the upper reservoir as a conventional hydro-power plant, but it converts electric energy from the grid to refill this reservoir, by pumping back water from the lower reservoir when it is economically profitable. The system working conditions are then designed according to the electricity trading and regulations services needed by the local energy market; for instance, PHES systems dedicated to peak load would be designed for generating, while pumping during light load.

To provide a better overview of the possible solutions in a PHES systems, the authors describe the main features of the hydraulic turbines generally installed and their comparison in terms of efficiency and operating range. This paper also gives an overview on the technology of the centrifugal Pumps As Turbines (PATs), proposing a review of the model dedicate to forecast pumps behaviour in reverse mode operation.

A non-conventional test case analysis where an existing unused mine of 350 meters deep would be used as a lower reservoir in PHES. The available head heavily varies with interittency due to the peculiar mine configuration. Consequently the hydraulic machines adopted could often work in off-design.

2 TURBO-MACHINES IN HYDRO-POWER PLANTS

The first important criterion in order to define a turbomachine for a selected hydroelectric plant is the available head H [m] and the plant power size P [kW] or energy generation [kWh] target (Gulliver & Arndt 1991).

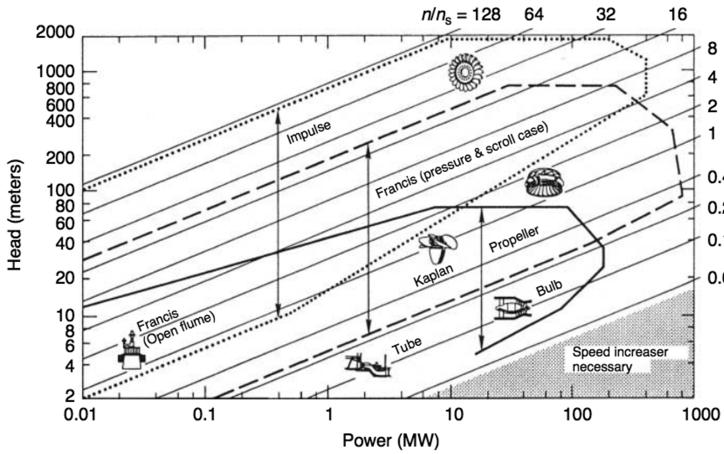

Figure 1. Application chart for hydraulic turbines (Gulliver & Arndt 1991).

In general for micro-hydro as shown in Fig.1, the Pelton turbines cover the high head domain down to 50 m. The Francis turbine covers the largest range of head under the Pelton turbine domain with some overlapping and down to 10 m head. The lowest domain of head for micro-hydro (below 10 m) is covered by Kaplan turbines with fixed or movable blades. For low heads and up to 50 m, the cross-flow impulse turbine is also used.

In large power plants the most used turbines are reaction turbines and are used in low (<40 m) and medium (40–500 m) head applications (Fig.1). Reaction turbines are moved by water, which changes pressure as it moves through the turbine and gives up its energy. In reaction turbines loss is generated in both fixed and moving blades.

Impulse turbines are often used in very high head applications (>300 m). In impulse turbines the jet pushes on the turbine's blades which changes the direction of the flow. The fluid flowing over the rotor blades is constant and all the work output is due to the change in fluid kinetic energy.

2.1 *Turbo-machines options in PHES*

PHES uses different hydraulic machines. Turbines and pumps, respectively, produce energy (generation) and consume energy restoring the capacity of the upper reservoir (pumping). The basic arrangement is a ternary pumped storage system that consists of a set of pumps and turbines. It is composed by thus two hydraulic machines and one electrical machine.

Reversible pump-turbines can also be used: the system consists of a reversible hydraulic machine, a fixed coupling and a motor-generator for storing and generating energy. In this system there is only one hydraulic machine, which works as either a pump or a turbine (with flow direction reversed), and one electrical machine, which works either as a motor or a generator. The reversible set is much simpler than a ternary set. Nevertheless, it has to stand fluid dynamic issues due to its double function and this might raise its costs.

Another challenge for the reversible machine is to provide a good efficiency η in partial load. The most used standard pump-turbine is the centrifugal and diagonal types and it must be furnished with special electrical equipment that allows variable speed in order to cover the required operating range in both modes of operation. This is realized by means of a doubly-fed asynchronous generator or full frequency converter connected to a synchronous generator.

An alternative solution for PHES is to install diagonal pump-turbines of the Deriaz type, mostly installed in the 1960s. These machines are able to cover a wide operational range, thanks to their adjustable runner blades, without requiring special electrical equipment. Deriaz machines can offer an alternative solution to the Francis machine in smaller PHES plants with large variation of the required load (Morabito 2014) with the drawback of more mechanical issues.

Another possible configuration for a hydroelectric plant is to use a Pump (diagonal or centrifugal) As Turbine (PAT). The advantages are that pumps are robust, mass-manufactured, more readily available and less expensive than manufactured micro/small-hydro turbines (Ramos, Borga, & SimÃ£o 2009). Micro and small hydraulic plants are mentioned because PAT is able to work in turbine mode to the detriment of high efficiency: the performances are far from the best efficiency point (BEP) due to the fixed internal geometry and absence of flow regulation. For this reason PATs are not recommended for big and medium size plants where a single percentage point of efficiency has a strong impact on the economic feasiblity of the installation (Orchard & Klos 2009) (Derakhshan & Nourbakhsh 2008b).

A single installation is not efficient over a wide range of flows; having multiple PATs of different sizes, each optimized for a different flow and suited for specific flow regimes, can overcome this limitation. PATs are studied in more details in this paper.

3 PUMP AS TURBINE

The use of pumps as turbines has been a research topic for over 80 years when engineers accidentally found that pumps were able to operate very efficiently in turbine mode when they were trying to evaluate the complete characteristics of pumps (Kittredge & Thoma 1931, Knapp 1937). In the 1950s and 1960s, the concept of PHES plants evolved mainly in developed countries to manage peak power requirements and support nuclear power plants design. In the later years chemical industries became another area for the application of PATs. In certain chemical processes it was necessary to dissipate the energy of high-pressure fluids through small pipe lengths. Instead of simply throttling, PATs were installed to recover some energy. Small Hydroelectric (SH) power stations became attractive to generate electrical energy with the increasing interest in a smart use of the energy. However the cost per kWh of these power plants was higher than for hydroelectric power plants with large capacity. Therefore using PATs emerged as an attractive and important alternative. Due to the large market for pumps of all possible capacities it is easily available, cheap and reliable (Orchard & Klos 2009). Therefore the maintenance has many advantages compared to custom-made turbines. Pumps are relatively simple machines, easy to maintain and readily available in most developing countries. From an economical point of view, it is often stated that pumps working as turbines are profitable in the range of 1 to 500 kW (allowing capital payback periods of two years or less which is considerably less than that of a conventional turbine (Paish 2002)).

Pump manufacturers do not normally provide the characteristic curves of their pumps in reverse operation. Therefore, establishing a correlation enabling the switch from the pump characteristics to turbine characteristics is the main challenge in using a PAT. The hydraulic behaviour of a pump rotating as a turbine changes. In general a pump will operate in turbine mode with a higher head and discharge for the same rotational speed. Many researchers have presented some theoretical and empirical relations for predicting the PAT characteristics in the best efficiency point (BEP). Most recent attempts to predict the performance of PATs, have been made using Computational Fluid Dynamics (CFD) simulations. However, without validating the CFD results by experimental data, they are not fully reliable. Besides, all these simulations included only hydraulic losses.

Many prediction techniques have been published. A few of the early contributors to these techniques were Kittredge (Kittredge 1945) and Stepanoff (Stepanoff 1957). In the later years many more techniques were developed by many researchers namely Nautiyal (Nautiyal 2011) Sharma (Sharma 1985), Schmiedl (Schmiedl 1988), Grover (Grover 1980), and more recently by Williams (Williams 1995), Alatorre-Frenk (Alatorre-Frenk & Thomas 1990) and Cohrs (Cohrs 1977). There are many uncertainties associated with the various prediction methods nevertheless they have served as a starting point in recent disseminations investigated for instance by Sigh and Nestmann (Singh & Nestmann 2010) (Singh & Nestmann 2011) and by Derakhshan and Nourbakhsh (Derakhshan & Nourbakhsh 2008b).

3.1 *Performance prediction of pumps in turbine mode*

The PAT behaviour is very complex and it is difficult to find a unique corelation for all pumps in reverse mode. Several problems have to be solved, linked to hydraulic losses in the volute and the impeller, mechanical losses related to packing and bearing cases, disc friction losses in gaps between rotor and stator and volumetric losses related to leakage from clearances between rotor and stator (Derakhshan & Nourbakhsh 2008b). One of the serious problems of PAT technology is to individuate the right best-input parameters for predicting the performance of the rated pump. Authors have published on the use of PATs in which the turbine performance is predicted using the values of head and flow at the pump's best efficiency point, obtaining scaled values of head and flow rate values in turbine mode. The different formulas, which have been proposed by the various authors, are considered in the following sections. When a pump operates in the turbine zone, the motor will operate as a generator. During pump or turbine operations the discharge, Q, is a function of the rotating speed, N, and the pumping head, H, whereas the alteration of the speed will depend upon torque, T. In order to characterize the machine, power P and efficiency η have to be identified. The relationships between some of these parameters in pump and turbine mode are presented in dimensionless form through the ratio with operational condition:

$$q = \frac{Q_t}{Q_p}, \qquad h = \frac{H_t}{H_p}, \qquad P = \frac{P_t}{P_p}, \qquad \lambda = \frac{\eta_t}{\eta_q} \quad (1)$$

with t for turbine and p pump.

Different methods can be used to estimate the best efficiency point in turbine operation from the manufacturer's pump performance data. One of the earliest available paper that presents such a method was published in 1962 by Childs (Childs 1962). He stated that the turbine best efficiency and pump best efficiency for the same machine are approximately equal. He further assumed that the turbine output power for best efficiency operation was equal to the pump input power. Hence:

$$P_t = \rho g Q_t H_t \eta_t = \frac{\rho g Q_p H_p}{\eta_p} \quad (2)$$

and

$$\frac{Q_t H_t}{Q_p H_p} = \frac{1}{\eta_p^2} \quad (3)$$

It is also assumed that $\frac{Q_t}{Q_p} = \frac{H_t}{H_p}$ and hence

$$q = \frac{Q_t}{Q_p} = \frac{1}{\eta_p}, \qquad h = \frac{H_t}{H_p} = \frac{1}{\eta_p} \tag{4}$$

Stepanoff (Stepanoff 1957) also proposed a method that depends on the pump efficiency. It relates the ratios of turbine and pump head and flow to the *hydraulic efficiency* of the pump, by the following equations:

$$q = \frac{1}{\eta_{hp}}, \qquad h = \frac{1}{\eta_{hp}} \tag{5}$$

Since the hydraulic efficiency is not normally known, the following simplification is incorporated:

$$\eta_{hp} = \sqrt{\eta_p} \tag{6}$$

Sharma (Sharma 1985) has developed a prediction method (preferred by Williams (Williams 1994)), that also uses ratios dependent on the pump efficiency. He uses the initial assumptions, as in the method presented by Child (Childs 1962), that $P_p = P_t$ and $\eta_p = \eta_t$, which results in equation (3) and relates the two specific speeds for pump and turbine operation by

$$n_{st} = \sqrt{\eta_p}\, n_{sp}, \qquad \frac{\sqrt{Q_t}}{H_t^{0.75}} = \sqrt{\eta_p}\, \frac{\sqrt{Q_p}}{H_p^{0.75}} \tag{7}$$

where specific speed is defined by $n_{st} = N_t \frac{\sqrt{Q_t}}{H_t^{0.75}}$ The following equations can thus be derived:

$$q = \frac{1}{\eta_p^{0.8}}, \qquad h = \frac{1}{\eta_p^{1.2}} \tag{8}$$

The method presented by Alatorre-Frenk (Alatorre-Frenk & Thomas 1990) is based on fitting equations to a limited number of PAT data. The equations are again based on the pump efficiency, and are expressed in the form

$$q = \frac{0.85\eta_p^5 + 0.385}{2\eta_p^{9.5} + 0.205}, \qquad h = \frac{1}{0.85\eta_p^5 + 0.385} \tag{9}$$

Several other authors have proposed equations that relate the head and flow ratios for pump and turbine operation to the pump or turbine specific speed.

The method by Grover (Grover 1980) is restricted to the specific speed range $10 < n_{st} < 50$:

$$q = 2.379 - 0.0264 n_{st}, \qquad h = 2.693 - 0.0229 n_{st} \tag{10}$$

Hergt's method is presented in graphical form in (Lewinsky-Kesslitz 1987), which shows a range of head and flow ratios, each given as a function of turbine specific speed. Hergt's equations are

$$q = 1.3 - \frac{1.6}{n_{st} - 5}, \qquad h = 1.3 - \frac{6}{n_{st} - 3} \tag{11}$$

As showed by different experimental researches, the results obtained by these relations deviate almost ±20% from experimental data (Williams 1994).

As described by Nestmann and Sigh (Singh & Nestmann 2010) the water inlet angle to the impeller in reverse mode is equal with volute angle. In fact, the volute operates as a guide channel. Assuming there is no swirl effect at the pump discharge, the water outlet angle from the impeller in turbine operation is equal with the impeller inlet angle. So we can consider the same Euler heads for turbine and pump modes: $H_{pEuler} = H_{tEuler}$. Due to the slip of finite blade number, pump and turbine theoretical head can be written as:

$$H_p = \mu H_{pEuler}, \qquad H_t = \frac{H_{tEuler}}{\nu} \tag{12}$$

Where μ is the slip factor for pump operation $\mu < 1$, and ν is the slip factor for turbine operation. The slip factor for reverse mode is approximately equal to 1.0 (Chapallaz, Eichenberger, & Fischer 1992). Considering its hydraulic efficiency, it can be written:

$$h = \frac{H_t}{H_p} = \frac{b}{\eta_p^a} \tag{13}$$

$\eta_p^a = \eta_{ph}\eta_{th}$, $b = \frac{1}{\mu\nu}$ with a and b bigger than 1 Since it is possible to affirm that $v^2 \propto H$, and including the direct and reverse flow with their leakages $Q_{p,l}$,$Q_{t,l}$

$$\frac{Q_t}{Q_p} \approx \sqrt{\frac{H_t}{H_p}} = \frac{b^{0.5}}{\eta_p^{a/2}} \tag{14}$$

The best fit from the experimental data brings to rewrite the chosen equations as:

$$h = \frac{1.2}{\eta_p^{1.1}}, \qquad q = \frac{1.2}{\eta_p^{0.5}} \tag{15}$$

with $a = c = 1.1$ and $b = 1.2$

3.2 *Proposed predicted model*

Derakhshan and Yang (Derakhshan & Nourbakhsh 2008a) (Yang, Derakhshan, & Kong 2012) have modeled a more complex method using experimental data to predict the best efficiency point of a pump working as a turbine based on pump hydraulic characteristics. In this study, a mini hydro-power test-rig was installed, and four centrifugal pumps ($n_s < 60$ (m,m^3/s)) were tested as turbines. Experiments showed that a centrifugal pump can appropriately operate as a turbine in various rotational speeds, heads and flow rates. A PAT works in higher head and flow rate than those of the pump mode at the same rotational speed. Efficiency is almost the same in both pump and turbine modes.

Combining the experimental data provided by other autors (Derakhshan & Nourbakhsh 2008a, Joshi, Gordon, & Holloway 2005, Singh 2005, Williams 1992)

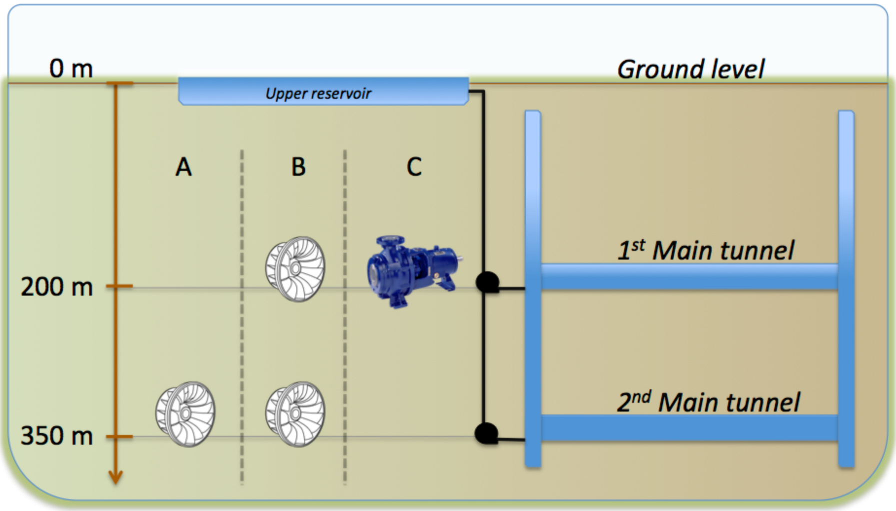

Figure 2. Sketch of the mine structure: it is composed mainly by two vertical pits that are crossed by two large tunnels. Test cases: A) one Francis turbine located at 350 m depth; B) Two Francis turbines located at 350 m ad 200 m depth; C) One PAT at 200 m.

it is apparent by tests that at the same specific speed different values for h and q may be obtained (Yang, Derakhshan, & Kong 2012). This leads to the conclusion that for smaller values of n_{sp}, the model based solely on the classification of the specific speed can not make a perfect prediction (Fig.3).

It is possible to obtain a better correlation of prediction than other various methods if one collects subgroups defined by impeller diameter. Searching for a trend line of this model through all the tests reported with the same diameter (i.e. diameter about 0.250 [m]), the following equations are found:

$$h = 5.1956 n_{sp}^{-0.323}, \qquad q = 3.1276 n_{sp}^{-0.219} \tag{16}$$

PAT efficiency value floats between the value in pump mode proposed by the most optimistic models mentioned before and more a mitigated approximation as the following equation by Gulich (Gulich 2007):

$$\frac{\eta_{opt,T}}{\eta_{opt,P}} = 1.16 \frac{n_{sp}}{200}, \qquad \pm 5\% \tag{17}$$

3.3 *Variable speed*

Units with variable speed are more used, especially for Reversible Pumps Turbines (RPTs). RPTs are hydraulic machines designed to work either as a pump or as a turbine depending on the direction of rotation. Furthermore, a well-designed, compact power house save equipment and civil costs.

The use of variable speed in PHES represents a valuable option to deal with off-design working condition. These systems couple the motor/generator to a frequency changer enabling a wider variable speed pumping or generating (± 12%) (Thoni & Schlunegger 2009). Compared to a conventional configuration at a constant speed, the variable speed offers several advantages. It ensures a rapid response to a daily unsteady demand of electrical power. This technology ensures to combine in a single machine a multiplicity of different design configurations. Adjusting the rotation speed allows to operate as close as possible to an optimal condition.

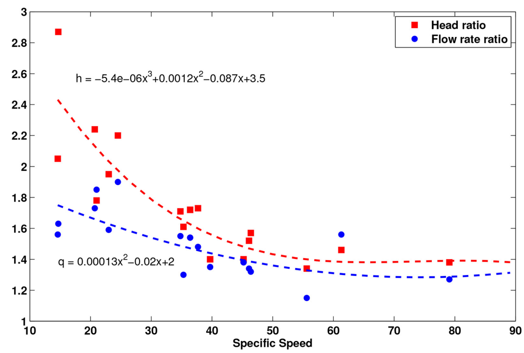

Figure 3. Experimental value of h and q.

However, the characteristic curve or set of characteristic curves for any turbo-machine can not be determined only theoretically (Round 2004). Better results are obtained by using empirical data available from performance tests at various speeds. Moreover, one must always consider that the calculations include some errors. The reference of −5% to +10% defines the range of rotational speeds where the affinity laws are applicable for pumps (Schoenung 2003, Nourbakhsh, Jaumotte, & Hirsch 2008). The affinity laws are very useful in small speed changes, but they have become predictive tools for VFDs (Variable Frequency Devices) operation where changes may be more dramatic.

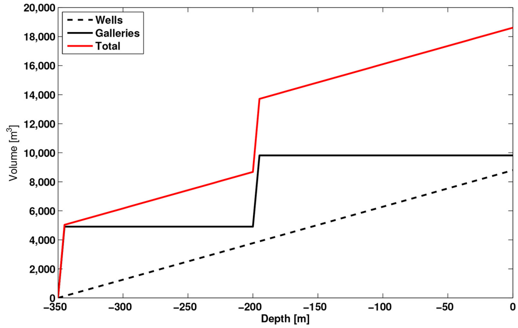

Figure 4. Volume of stored water in the mine of the presented test case.

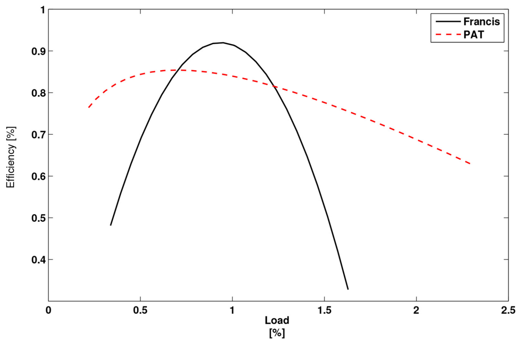

Figure 5. Efficiency curve for the Francis turbine selected and estimation of the performance of the PAT with variable speed.

Nevertheless, there are operational speed limits due to a physical boundary of operation (e.g. cavitation, vibrations).

4 CASE STUDY

The case study presented is based on an unconventional solution for PHES: the lower reservoir is an old mine traversed by a multitude of pits and tunnels. As shown in Fig.2, it mainly consists of two main horizontal tunnels (about 14700 m^3 each) connected by two wells (about 12500 m^3 each). The mine extends over a depth of 500 meters (Archambeau & Erpicum 2015) but in this case the maximum available head used is limited to 350 meters. Due to its relevant depth, it is possible to outrange the nominal design conditions in power generation mode, generally limited in a small range of available head. The mine structure is simplified by secondary minor tunnels and not relevant cavities. The upper reservoir is located at ground level.

In fact, when the gallery on the bottom is filled (reaching a volume of about 4900 m^3), the wells will be refilled, which, due their vertical extension, quickly reduce the available head. At the depth of 200 meters there is another tunnel. The water volume increases considerably when the principal tunnels are reached (Fig.4) at 350 meters and 200 meters depth (Fig.2).

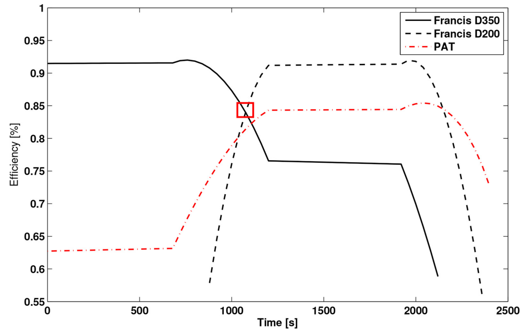

Figure 6. Efficiency trend of the turbines while generating and filling the mine. The red square marks the switch of working turbine in the configuration B.

It is a synthetic case that can be used for preliminary pumping or turbining phases study. Mines have unlikely a good geometrical description, especially for the most ancient: in fact they much more secondary tunnels and wells for ventilation or additional services. Moreover, the soil has its porosity which, acting as a large buffer and prohibiting the total filling of the mine, affects the process of pumping and generating.

Therefore three turbomachinery options are studied for this site can be summarized as following:

A) a single power station located on the bottom of the lowest tunnel equipped with a Francis turbine;
B) option A with a second Francis unit at 200 m depth ready to operate once the first is in extreme off-design condition;
C) using only a centrifugal pump in turbine mode with variable speed.

4.1 *Scenario A*

In the case with a system using only a single powerhouse at the bottom of the reservoir, a Francis turbine is selected for a nominal head of 350 m. The flow rate used is fixed to 7 m^3/s in order to produce around 20 MW according to the project goals. With this flow rate value the working time is about 36 minutes before the efficiency drops under the 60% due to varying head. This power plant set (configuration A Fig.2) has to deal with the turbine variable efficiency as the head changes with the filling of the mine. The available head decreases while the reservoir is getting full, faster while the vertical pits are crossed (Fig.4). The time ranges where the efficiency looks almost constant occur when the large tunnels are being filled and, consequently, the head remains stable (Fig.6).

4.2 *Scenario B*

The second scenario aims to take advantage of the specific configuration of the mine (Fig.2). Two units have been designed: one with a Francis turbine at nominal head of 350 meters (FrancisD350) and a second unit designed for 200 meters (FrancisD200), both ready to better exploit the stationary head condition provided by the large buffering of the tunnels. A switch point is found on their variable global efficiency. It defines the moment where FrancisD200 efficiency is higher than FrancisD300 efficiency (Fig.6). At this instant the FrancisD300 shuts down while the second one starts: this allows the system to keep high efficiency where a system of the one machine would be in extreme off-design conditions. Hence the global efficiency of the configuration stands mostly above 90% (Fig.6).

4.3 *Scenario C*

Despite the fact that the predicted power capacity for this case study is out of the recommended range for PATs (Derakhshan & Nourbakhsh 2008a, Paish 2002), this solution could represent an important alternative. A single stage commercial pump, matching the conditions requested in turbine mode for an available head of 350 meters, has not been found by the authors. A multistage radial flow may not fit in the model discussed before: the behaviour of the first stage of the PAT will trickle down to all the following stages, amplifying the effects and increasing the model uncertainty. Hence a PAT for an available 200 meter head is chosen. The centrifugal pump selected has an impeller of 980 mm diameter and a discharge section of 700 mm. Based on the selected prediction model discussed before (Par.3.2) and according to the characteristic curves provided by the pump manufacturer, its behaviour in turbine mode is predicted. In order to maintain sufficiently high efficiency for large range of load, its rotation speed is changing in the range of 1000-850 rpm thanks to a VFD.

Unlike the previous scenarii, in this option the flow rate is depending on the variable available head according to trend-line of estimated characteristic curves, given by the equation (18):

$$Q = 0.1547H^{0.6153} \quad (18)$$

In contrast to a conventional hydraulic turbine, a PAT is not equipped with movable guide vanes, hence the profile of the flow rate with partial load differs from a Francis turbine. According to equation 18 the flow rate drops from the highest value of 5,5 m^3/s to 2,1 m^3/s and this justifies also the reduced power production compared to the previous scenarii (Fig.7).

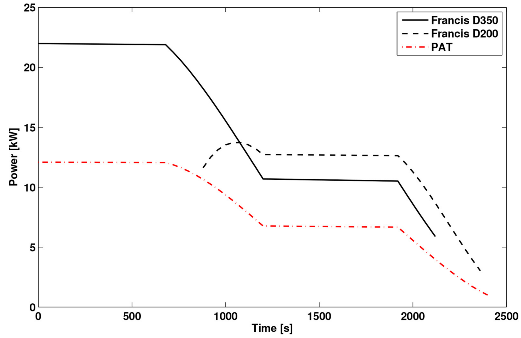

Figure 7. Power generated by the different configurations.

5 CONCLUSIONS

In this report important overviews are described regarding the possible hydraulic machinery configurations used in PHES. The most used but also the most customized and expensive is the reversible machines. PATs can offer an interesting option and, although they do not ensure a high efficiency, they have a lower capital cost. The possible efficiency detriments of the PATs might be mitigated and in cases off set by the detailed research conducted on forecasting PATs behaviour and thanks to VFD.

PATs show a value of power generation reduced compared to the other options, due to variable mass flow rate over time. Concerning the efficiency, PATs supported by the VFD are able to work more efficiently than turbines in the relevant off-design range. The performance of hydraulic turbines appears to be higher than PATs as shown in Fig.5 and Fig.6.

Regarding to the market, accessibility and maintenance, dealing with a hydraulic centrifugal pump is categorically simpler and economically more slender (Ramos, Borga, & SimÃ£o 2009). The pump market is well developed and several experiences have shown that centrifugal pumps can operate in turbine mode without any important difficulties (Carravetta & Ferracotta 2014, Orchard & Klos 2009, Ariaga 2010). Compared to a ternary pumped storage system, PATs, as RPTs, have the advantages of saving room and the consequent excavation costs. That is an important issue for hydroelectric plants with power-house built deep in the ground as in the case study.

The configurations proposed for the case study had to demonstrate the profitability of this non-conventional PHES plant. While the configuration B can properly operate in steady state conditions for a longer period of time than the other options, the configuration C guarantees a relevant energy capacity for the mine exploitation with the most reduced investment among the configurations. Therefore the integration of two options may occur, giving as result the installation of a Francis turbine designed for 350 meters head and a PAT operating at 200 meters depth.

It would be worthwhile detecting all the structural advantages of working with generating units located at different available heads. Pipelines and excavation costs have been voiced in this regard. If this analysis finds success, it will be reasonable to design even a third power-house deeper due to the fact that the selected mine reaches deeper. This arrangement might be an example to laud the profitability of the PHES plants and revaluation of spaces unused.

PHES system have all the features needed for being a suitable solution for the impelling requests of smart use of energy. There are still questions regarding the economic profitability of these applications and their social and environmental effects that still happens for hydro-power. This paper gives an overview of the configurations and settings for PHES even in a non-conventional case and enables further issues and challenges for the future.

ACKNOWLEDGEMENT

This research was supported by Region Wallonia. The data presented, the statements made and the views expressed have been achieved thanks to the *SmartWater* project.

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