Hongrin-Léman hydroelectric pumped storage plant, Veytaux II powerhouse - Developing a new generation of multistage storage pumps

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Preface of the Owner: Forces Motrices Hongrin-Léman SA (FMHL)

The Swiss company Forces Motrices Hongrin-Léman SA (FMHL) has requested VOITH HYDRO GmbH & Co. KG in Heidenheim, Germany to design, manufacture, install and commission two 120 MW pumps for the project to double the installed power of the existing hydroelectric pumped storage power plant.

As part of the scope of this mandate, having recently finished the design studies and the experimental modelling, VOITH HYDRO Holding GmbH & Co. KG comments in this article on the most recent developments in this field.

Thus it can be seen that the maximum efficiency obtained over a wide operating range, the significant guarantees given with respect to cavitation, as well as the intrinsic characteristics of the pump which limit overpressure due to hydraulic transients are indeed remarkable achievements.

Abstract

In order to double the total installed capacity of the Veytaux pumped storage power plant, the existing plant put into operation in 1970 will be enhanced by two ternary sets. Both plants are owned by the Swiss-located FMHL (Forces Motrices Hongrin Léman) belonging to Romande Energie SA, Alpiq Swiss SA, Groupe E and the city of Lausanne. FMHL representative is Alpiq Suisse SA and the operator is Hydro Exploitation SA. Each set, rated at 120 MW, consists of a five-stage storage pump coupled to a motor/generator with a gear coupler and a Pelton turbine. The lower reservoir is the "Lac Léman" - the "Lac de l'Hongrin" will be used as upper reservoir for both power stations. The project Hongrin-Léman extension will provide additional power capacity by the end of 2014. Here, the focus will be on the challenges and the development of the multistage pump.

Today the expectation regarding operating range, cavitation behaviour and, of course, efficiency level is much higher than in the last decades when most existing pumped-storage plants with ternary sets were built. For example the KopsII plant operated by the Vorarlberger Illwerke (VIW) since 2007 created a new milestone regarding capacity, flexibility and efficiency of modern storage pumps. Hongrin-Léman is a next step towards modern pumped-storage with ternary sets.

The hydraulic design challenges for the five-stage storage pump of the Veytaux II are manifold. First, a high level of efficiency must be reached for a complex hydraulic machine. Second, the cavitation safety requirements are high, and third, the operating range is stretched due to the possible parallel operation with the Veytaux I machines and the hydraulic short circuit. Furthermore, the head of over 900m in constant operation is an additional challenge for the design. These goals could only be reached by employing modern design and analysis methods for simulation and measurement.

In early 2011 Voith Hydro GmbH & Co. KG in Heidenheim was awarded the contract for the two five-stage storage pumps. The model acceptance test was conducted in the "Brunnenmühle" hydraulic laboratory of Voith Hydro Holding GmbH & Co. KG and successfully finished at the end of 2011.

1. Introduction

The increasing amount of new renewable energy sources (mainly solar and wind energy) in the energy production in Europe requires also a higher amount of storage capacity and flexibility of the networks. Still one of the most efficient methods of grid regulation and energy storage is the use of pumped-storage plants. In order to be prepared for the future tasks, in 2007 FMHL decided to extend the capacity of the Veytaux pump-storage scheme. ***Figure 1*** shows the position of the production site at the Lake Geneva near the city of Montreux, CH and the whole scheme.

The existing scheme of Veytaux I consists of four horizontal shaft ternary sets with a five stage storage pump, a feeder pump and a Pelton turbine providing a capacity of 60MW each. Veytaux II (FMHL+) will be equipped with two ternary vertical shaft sets of 120MW. Here, the focus lies on the development of the five-stage radial storage pump and the hydraulic transient analysis conducted during the project.

Concerning the requirements on pump hydraulics several tasks need to be solved to obtain a reliable and optimal solution. To minimise impact on environment Veytaux I (V1) and Veytaux II (V2) will both use

Figure 1: Position of the Veytaux power station (left) and general layout (right).

the existing penstock. Hence, pipe pressure losses will increase and play an important role for operation modes with high overall discharges. Regarding the pump this means a considerable increase in the head range to be covered. Consequently, good cavitation properties for the corresponding discharge range and a flat efficiency curve must be achieved. Furthermore the high discharges require a detailed water hammer analysis and careful definition of operating laws.

2. Project Scheme and Planned Operation

The Veytaux II plant is designed to give services for grid regulation. Hence short changeover times are not required and instead of a hydraulic torque converter a regular gear coupler is installed. For starting the water-filled pump the Pelton turbine is applied as a start-up turbine. ***Figure 2*** shows the complete Veytaux site and a cross section of the ternary set with the gear coupler between the turbine and the five-stage pump.

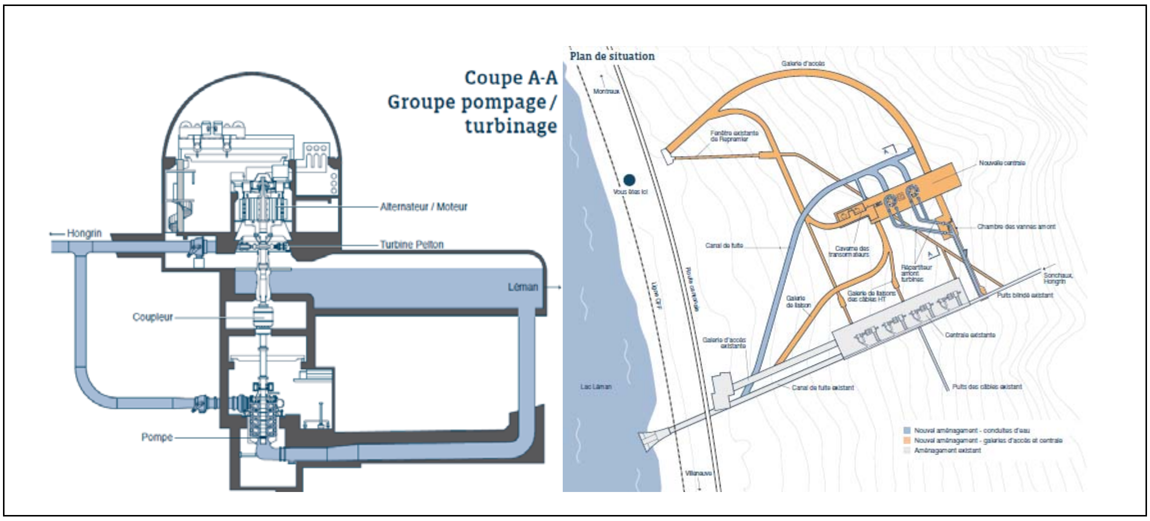
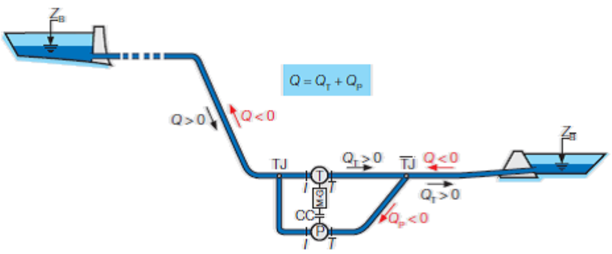


Figure 2: Cross section of the Veytaux II ternary set (left) and scheme of Veytaux I (grey) and II (orange).

Usually pumping operation causes a constant, non-regulated power input on a set. In order to achieve a controllable pump operation the new machines must be able to operate in hydraulic short circuit mode, i.e. a fraction of the pumped discharge is directed back to the turbine. Since the turbine is fully regulated it is possible to control the power demand of a set almost between 0…100%.



The concept of hydraulic short circuit operation is already implemented and functional in the pumped storage plant Kops II owned by Vorarlberger Illwerke (Austria) [1]. ***Figure 3*** shows the principle flow situation in hydraulic short circuit operation.

Figure 3: Hydraulic short circuit in principle [2].

In order to reduce environmental impact Veytaux I und Veytaux II will both use the existing penstock. Hence discharges are quite high when both plants are in operation and the head losses contribute significantly to the required pump head. Table 1 shows the main data of the Veytaux II pumps including the net head range (76m) which is 60% higher than the gross head range (46m). Correspondingly the discharge range between maximum and minimum head operation is also increased. Additionally maximum discharge will increase even further for some hydraulic short circuit operation points at minimum head. These boundary conditions have to be taken into account for the new hydraulic design of the pump. A stable operation with good efficiency over the whole operating range with good cavitation properties, i.e. a flat cavitation curve, must be ensured.

Head range (geodetic)	[m]	837.4 ... 884
Head range (manometric)	[m]	841 ... 917
Operation mode	[V1+V2]	0+1 ... 3+2
Runner diameter	[m]	2.2
Pump input P_{Pu} at min. head	[MW]	118
Rated unit discharge Q_{rat}	[m³/s]	12.5
Hydraulic short circuit Q_{max}	[m³/s]	13.3

Table 1: Main data of the Veytaux II storage pumps.

3. Hydraulic Engineering and CFD Analysis

The ever increasing demands for better performance, higher reliability and lower costs, are driving the development in the areas of fluid and structural mechanics. The principal goal of the hydraulic design of the Veytaux II multistage storage pumps is to achieve an increased efficiency level compared to former projects, good pump stability behaviour, and cavitation safety in the required operation range.

The design optimisation includes the whole hydraulically effective geometries of the pump. This includes the suction elbow, runners with labyrinth seals, bladed passages and spiral case. Of course the runners will be optimized regarding the boundary conditions mentioned above. Furthermore, the draft tube elbow, the spiral case and the bladed passages of guide and return vanes need to be investigated and adapted to fit the current operating conditions and the new runner design.

A complete 3D view of the five-stage pump is shown in ***Figure 4***. Runners are colored blue, guide vane sections pink and return channels grey.

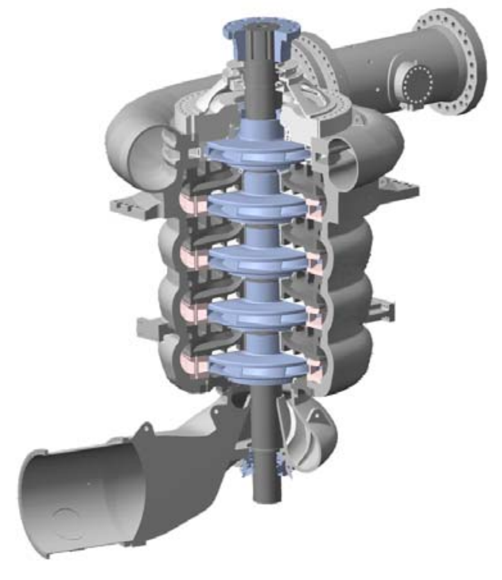


Figure 4: 3D view of the multistage pump.

Engineering methods applied for hydraulic design

The basic flow equations, the Navier-Stokes equations, are derived from the conservation of mass and momentum and describe the complete physics of the fluid. For complex geometries and flow conditions they must be solved with numerical methods. As a result of these simulations, the designer retrieves the velocity vectors and the pressure field for the entire flow region. A major aspect of this method is the relatively high computing effort, which is the main reason for applying it at an advanced point in the design process.

Generally the flow in hydraulic turbines is turbulent, which means stochastic fluctuations superpose the mean flow field. Currently for turbo machinery applications it is not possible to compute the flow including all these fluctuations. From an engineer's view it is also not necessary, hence the influence of the turbulence on the mean flow is represented by turbulence models. A variety of different turbulence models can be applied depending on the application. The turbulence model is still one of the largest error sources in modern computational fluid dynamics (CFD) software. Navier-Stokes methods can be applied for the calculation of all main components of a pump turbine: spiral case, stay vane, guide vane, runner, draft tube and seal system. The flow interaction between adjacent components is considered by using different coupling algorithms [11]. Depending on the demands concerning accuracy and computation time an appropriate coupling model is chosen. As a standard, the stay vane, the wicket gate, the runner and the draft tube are calculated in one large calculation. However, it is important to note that an exact prediction of the absolute efficiency level of a pump over the whole operation range is not yet possible, even when using the most advanced available CFD tools.

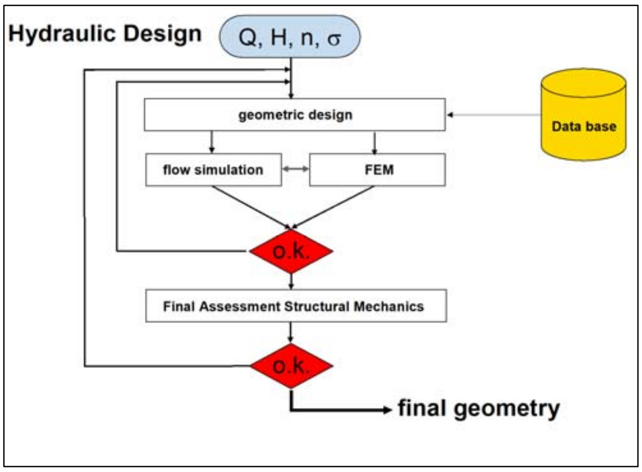


Figure 5: Workflow of hydraulic design process.

In order to be able to achieve shorter development times, better efficiency and higher structural capacity despite extended operating ranges, i.e. high quality, Voith Hydro uses an engineering system that integrates specialized design tools, general CAD and automated numerical simulation and analysis software. This environment enables the designer to analyse and modify the hydraulic and structural performance of a machine within a very short time [5][6][7][10]. Since it is not only able to represent single-stage machines but also multistage geometries it is also applied for the design of the Veytaux II pump.

Today enough computing power is available to analyse the hydraulic design applying modern simulation methods from the beginning of the design process. Starting with simple models of single hydraulic parts for promising designs finally a whole machine is investigated.

At Voith Hydro Navier-Stokes computations are performed for example with the commercial code Ansys-CFX. This CFD program is well tested and established in the field of numerical flow analysis, especially for turbo machinery. Its precursor, TASCflow, was used at Voith Hydro for more than ten years and showed very high accuracy in predicting complex turbulent flows. The accuracy of the predictions has further improved using CFX and finer computational meshes. In order to ensure an efficient process the flow simulation tool is embedded in a complete simulation environment. This includes the generation of the hydraulic surfaces, an automatic generation of a high-quality computational mesh, the set-up of appropriate boundary conditions and finally running the simulation and automatic post-processing of the simulation results. All results are plotted in standardised format to enable the engineer to easily compare different variants. Furthermore, a static and dynamic stress assessment using the finite-element method is computed for the preferred designs [9]. The whole process is shown in ***Figure 5***.

Hydraulic design for the five-stage storage pump

The requirements for modern pumped-storage sets have changed significantly in the last years [3]. Hence from the hydraulic development point of view the Veytaux II pump represents a new generation of storage pumps. Not only the increased head range due to the parallel operation with the existing units of Veytaux I but also new operation modes used for grid regulation, e.g. hydraulic short circuit, have to be taken into account. One important boundary condition that needs to be respected is the minimum achievable shaft diameter influencing hydraulic design and limited by shaft dynamics.

For mechanical reasons the pump has a guide bearing below the suction elbow as shown in ***Figure 4***. The shaft crossing the suction elbow disturbs the flow towards the pump intake. This disturbance has to be minimised by special design features in the suction cone, especially since it has a big influence not only on the pump performance and stability but also cavitation properties of the first stage [4]. Furthermore, the return channels and vanes, redirecting the flow to the following stage, have to be designed with care to obtain suitable flow conditions, e.g. minimum swirl, at the inflow of the runner.

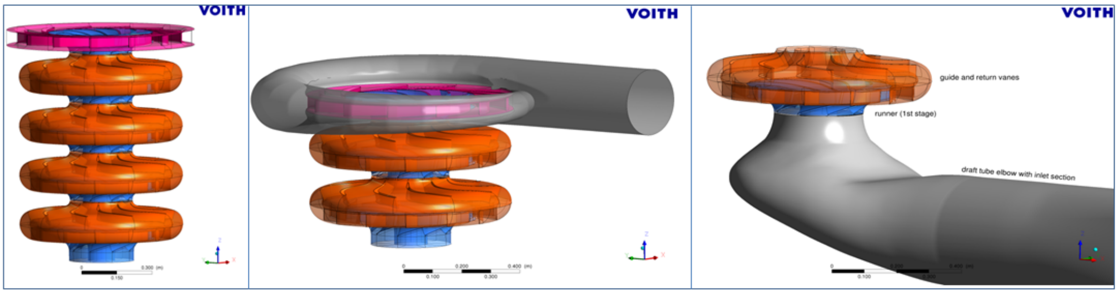


Figure 6: Applied CFD Models (left to right). 5-stage, 3-stage with spiral case, single-stage with elbow.

In ***Figure 6*** different geometry set-ups used in the CFD-based design process are shown. For normal design, steady-state simulations with a three or five-stage model composed of runners (blue), guide vanes and return channels (orange) and stay vanes (pink) are used. To check the performance of the new spiral case and the draft tube elbow, including the inflow section from the roller gate, separate models are used. In order to see inconvenient effects and to study rotor-stator excitation effects the spiral case is investigated with a fully transient analysis. Where possible in steady-state simulations sector models of the bladed passages are used for simplicity applying periodic boundary conditions and stage-averaging interfaces.

As mentioned above, a uniform circumferential flow distribution at the inlet of the first stage runner is mandatory to minimise radial forces or unfavourable cavitation behaviour. ***Figure 7*** shows the velocity distribution and the secondary flow vectors at the outlet of the draft tube elbow near the first stage runner. The optimised draft tube geometry produces an almost constant flow distribution and only minor secondary flow. Hence, the impact on cavitation and runner forces will be negligible and the inflow may be considered as uniform.

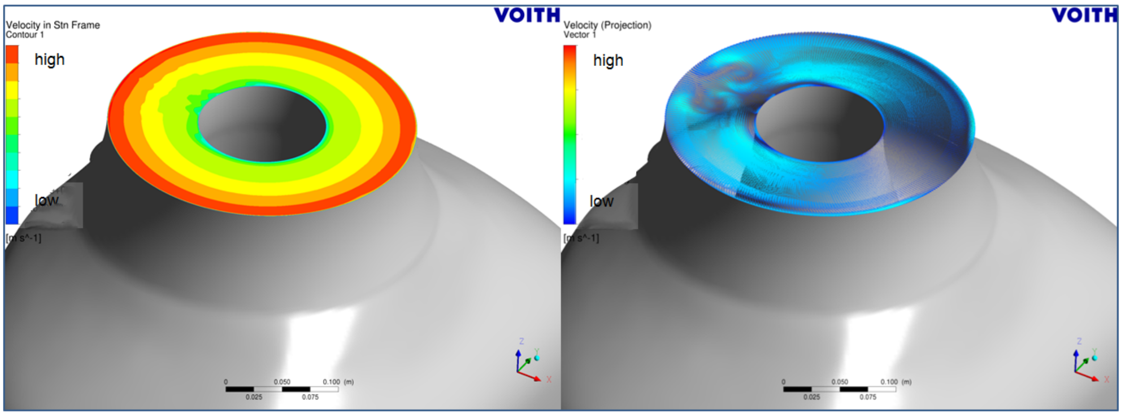


Figure 7: Absolute flow velocity (left) and secondary flow vectors (right) at the transition between suction elbow and first stage runner.

Furthermore, the spiral case is optimised and checked with a fully transient flow simulation, see Figure 8. Both pictures show the transient simulation results at a randomly picked, representative time-step. Even though the wake behind the stay vanes can hardly be avoided the impact on the mean flow in the spiral casing is almost not present. The pressure shows the expected distribution.

Regarding cavitation the first stage runner is considered to be the most critical in multistage pumps. The distribution of the local Thoma number on the blade surfaces shows a good distance between beginning cavitation and plant conditions, see Figure 9.

Even though the prediction of the behaviour of the machine with the applied CFD models is sufficient to evaluate different design variants and to optimize the geometry, the uncertainties are higher than usually known from single stage machines [8][12]. This especially applies for the part load pump characteristics. Hence, a model test is necessary for this kind of challenging type of hydraulic machine.

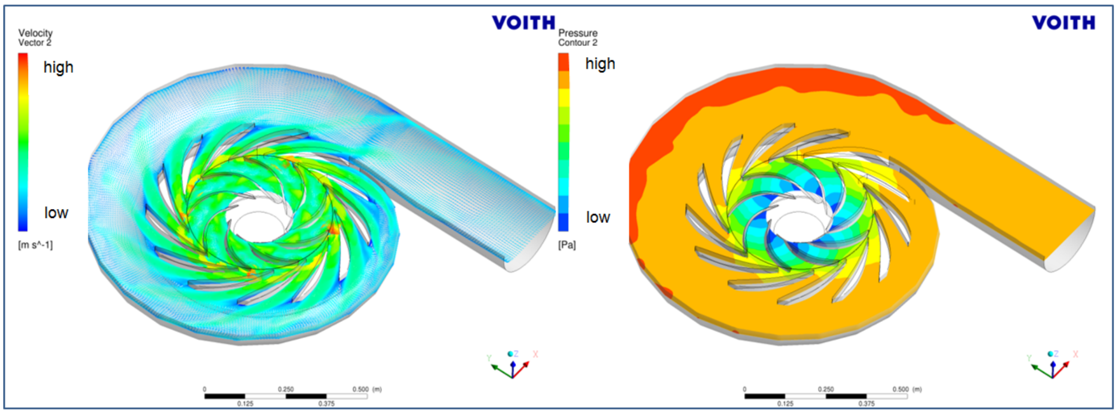


Figure 8: Spiral case centre plane cut. Velocity vectors (left) and absolute pressure (right) for optimum.

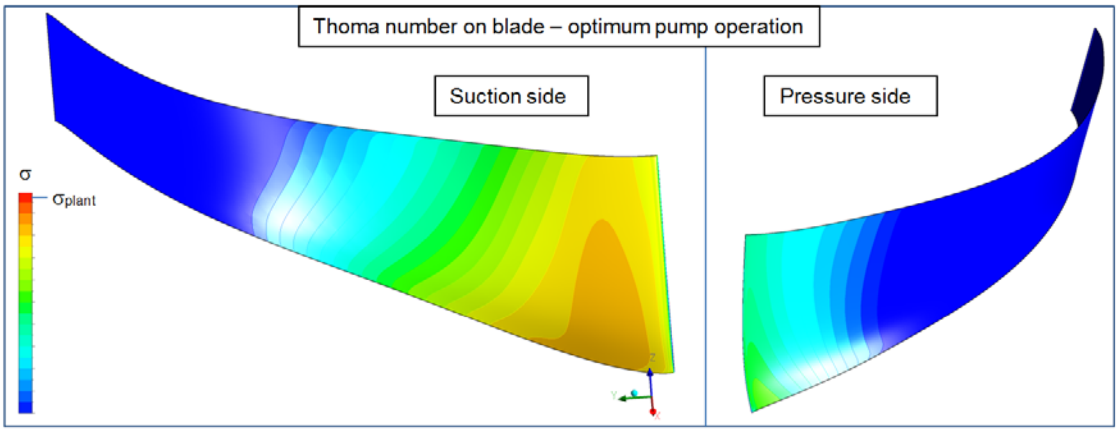


Figure 9: Cavitation index on blade surface (first stage) at optimum operation.

4. Model test results

In order to verify the contract guarantees, the cavitation behaviour and pump head curve a model test has to be conducted. For this model test the best variant from CFD-development is chosen and tested with two different sets of return vanes.

The model tests as well as the witnessed tests are performed on the VOITH hydraulic laboratory "Brunnenmühle" in Heidenheim, see ***Figure 10***. The test rig, completely renewed in 2007, allows turbine and pump operation and provides modern measurement and data acquisition equipment for best productivity.

Figure 10: Universal High Pressure Test rig at the "Brunnenmühle" hydraulic laboratory in Heidenheim (left). With the Veytaux II model machine mounted (right).

Referring to former projects and assuming the same hydraulic behaviour of all stages it is sufficient to represent each type of stage once in the model, i.e. suction stage with elbow, intermediate stage and spiral stage. Hence, the model consists of three stages. Of course the measured efficiency has to be corrected according to the number of stages in the prototype, which will have five stages.

All model parts from the suction elbow to the outlet pipe between spiral case and spherical valve are fully homologous and the tests are conducted in accordance with IEC 60193. Axial thrust plays an important role for dimensioning of the bearings and may be computed directly from the measured values without any further corrections due to the homology of the seals.

The three-stage model machine uses more parts than standard single stage models and the overall project time schedule is optimised and without time buffer. Hence, a convenient material choice and modern methods are applied to reduce the manufacturing times and therefore stay in the time frame.

Figure 11 shows the measured pump head curve and efficiency for the two tested variants and the previous simulation results. Obviously, the second variant has a better performance and is chosen for prototype engineering.

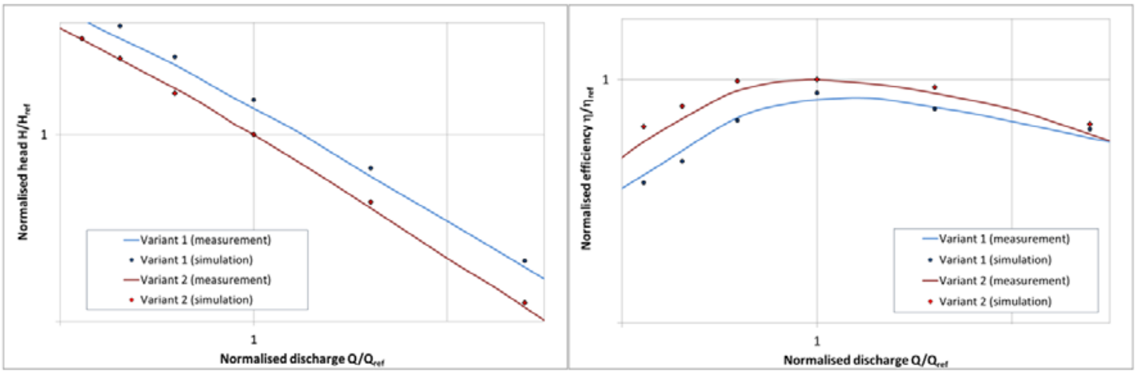


Figure 11: Comparison of simulation and measurement for two design variants.

Regarding the predicted pump head characteristics the simulation is in good accordance with the measured pump behaviour. Also the efficiency near the optimum point and for higher discharges fits well with the measurements. But at part load operation the prediction of the pump efficiency is less precise.

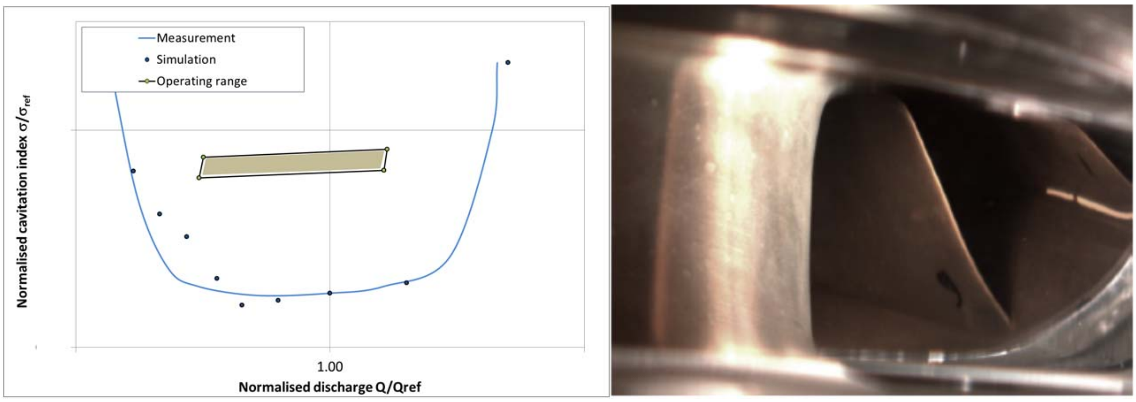


Figure 12: Normalised incipient cavitation curve (left) and observation at first stage runner inlet and plant conditions (right).

Figure 12 shows the measured incipient cavitation curve compared to the respective simulation results (left). Obviously the distance to the operation range (plant condition) is sufficient and leaves a comfortable safety margin to incipient cavitation. Hence, cavitation free operation is expected for normal pumping operation. The curve is based on measurements and observation made at the test rig through the Plexiglas cone of the draft tube. The situation at plant conditions near optimum operation is shown in the photograph. Again, simulation results fit well with the measurements. Since prediction of incipient cavitation at the runner leading edge is sensitive to mesh resolution in this area, the difference here (small discharges) is higher than for areal cavitation occurring near optimum discharge.

In summary the machine with the second set of return vanes fully meets all contract guarantees in matters of hydraulic performance and is chosen for final prototype design. It sets a new milestone in multistage pumps regarding operation range and efficiency level. Even though the schedule is optimised and very tight and still multistage pumps are challenging machines away from common standard the geometry was released in time.

5. Transient analysis and closing law

The new Veytaux II power units are an extension of the existing Veytaux I power plant, which was built in the 1970s. The existing water circuit system, means headrace tunnel, the steel lined penstock and the tailrace channel is used to provide water for the additional power units, without expanding the existing or adding a new circuit system. As the generating and pumping capacity of the total plant is almost doubled, the stationary discharge of upstream and downstream waterways can reach twice the original design discharge, as well. This circumstance requires detailed analyses of water circuit in stationary and especially in transient conditions.

The main adjustment of the upstream waterways was the adding of a second surge shaft in order to satisfy the increased discharge and the consequently required surge capacity. The preliminary conception of the new surge shaft was done at an early design stage by GILHEM (Groupement d'ingénieurs Hongrin Léman) with the help of a numerical simulation [13], [14] together with Power Vision Engineering Sàrl, PVE, as expert for Alpiq Suisse SA. The location of the surge tank is defined by various boundary conditions, such as geological and geographical data, and technic and economic feasibility. The main surge shaft dimensions, height and diameter, are given by the relative position in the waterways, as well as the discharge data of the power plant. Particular characteristics, as orifice size and loss coefficient, need to be determined by the investigation of various transient scenarios. A basic layout was done by GILHEM and PVE and will be verified by comprehensive transient analyses with final characteristics of hydraulic machines and valves.

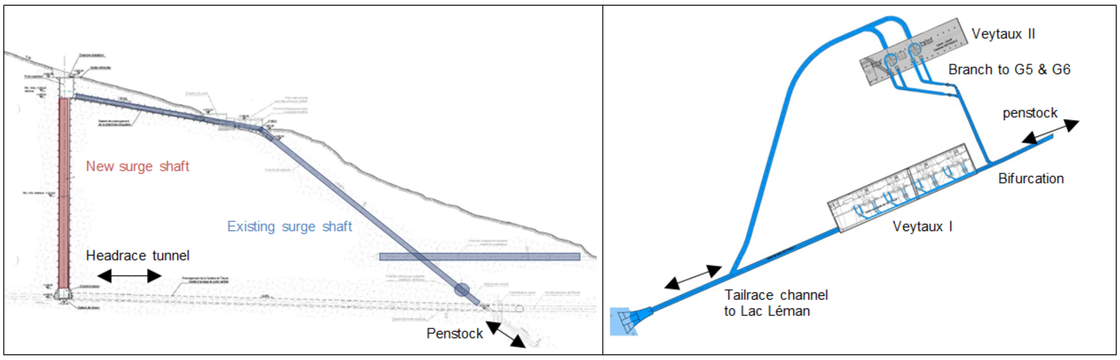


Figure 13: Location and layout of surge shafts (left), water circuit and power house layout (right).

From the upper reservoir (Lake Hongrin) headrace tunnel (8 km length, 4 m diameter) leads to the penstock. The existing surge shaft is located directly upstream the penstock (length 1.4 km, 2.9 m diameter). Another 340m further upstream the new surge chamber is excavated. Both shafts are connected to each other at their highest level as shown in ***Figure 13***. Hence, at high oscillation amplitudes, the surge levels are influenced by each other.

The bifurcation to the Veytaux II powerhouse is located at the lower end of the steel lined penstock. Further 70m downstream of the branch another bifurcation divides the water to the individual circuits of the new ternary units (Group 5 & 6). The individual circuits can be separated with the help of a common spherical inlet valve. The last penstock section (see ***Figure 2***) connects the Pelton distributor with the pump spiral case located 22m below. The pump can be disconnected by an upstream spherical valve (PDV) and a downstream roller gate. The open tailrace channel of Veytaux II is connected to the existing channel of Veytaux I as shown in ***Figure 13***. In order to represent the behaviour of the hydraulic machinery properly, four-quadrant curves obtained from hydraulic model tests are used in the simulation.

Numerical transient simulation

In order to perform a transient calculation the hydraulic characteristics of the complete water circuit and all hydraulic components (machines, valves, orifice, etc.) are modelled for numerical simulation. Voith Hydro uses the commercial simulation software 'SIMSEN', covering the analysis of electrical power networks, adjustable speed drives and hydraulic systems [15]. The computation method is based on the solution of Euler's law of motion and equation of continuity considering the elasticity of water column and piping [16].

The main task of transient simulation is the determination of operating times for all valves, Pelton nozzles and deflectors, and the analysis of dynamics and limits of hydraulic circuit. The limiting properties are the maximum occurring pressure in the penstock, the up- and down- surge level in the shafts and the minimum pressure in the headrace tunnel. To take these requirements into account, it was necessary to study a multiplicity of load cases, operating modes and conditions.

For Veytaux I and II the numerical model includes the water circuit, both surge shafts and the upper link, 3 Pelton- Turbines and 3 Pumps of Veytaux 1 with one main inlet valve each, 2 Pelton- Turbines and two 5-stage Pumps of Veytaux II with one pair of PDV and main inlet valve for the turbines and 2 common inlet valves. A case with all machines (4 Veytaux I, 2 Veytaux II) in operation is not considered, because maximum power input in operation is limited to 420MW and 60MW in reserve. Besides the pumping and generating mode, Veytaux II is designed to operate in hydraulic- short circuit mode and in parallel with 3 units of Veytaux I. All together it ends up in 14 possible combinations, taking the maximum and minimum conditions of gross head into account, only.

In the transient calculation grid separation with load rejection at generator, power unit start-up, loading scenarios and several combinations of the mentioned cases are studied.

Generating mode

The transient conditions in Pelton turbine generating mode can be handled without special procedures. Nozzle closing and opening laws are defined according to the given limits, such as permissible pressure values, minimum or maximum at start-up and shutdown, or specified operating times for various sequences. The deflectors prevent high overspeed of the generating unit without negative effects on water hammer. Since they cause considerable surge tank oscillations, especially sequences of unloading and reloading are critical. Consequently, unfavourable excitation of surge oscillation has to be prevented.

Figure 14 shows the transient behaviour for load reduction of five Pelton turbines (Veytaux I+Veytaux II = 3+2) with reloading at worst conditions for minimum surge tank levels. If the reloading is done synchronously for all machines the water level in the surge shafts reaches the minimum value and thus leads to minimum pressure in the headrace tunnel. The opening time and sequence of the injectors has been adapted in order to keep the minimum pressure in the gallery to acceptable values.

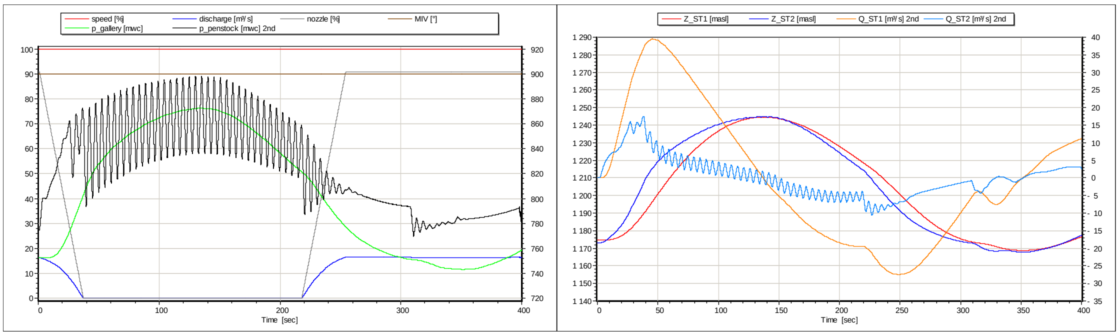


Figure 14: Generating mode, synchronous load reduction at V1+V2=3+2 and reloading.

Pumping mode

The pumping mode is of high interest in transient analyses. While the pumping units are connected to the grid the consumed electrical energy is transferred to pump head and makes the water flow upstream. After a sudden load rejection the water masses push downhill due to gravity. In less than 6 seconds the flow direction changes, and consequently the direction of rotation of the power unit changes, too. During this scenario the discharge changes direction and absolute value significantly in a short time period. Even without external intervention this causes a remarkable water hammer.

Now the challenge is to find PDV operating laws restraining the water circuit dynamics without increasing water hammer and meanwhile optimize operating times. Keeping the existing Veytaux I power units and the present closing times in mind and considering all possible combinations of operating modes and various scenarios of load rejections reduces considerably the degrees of freedom of design for this task.

The highest pressure in penstock due to water hammer is expected at delayed load rejection of all pumping units (V1+V2=3+2) at maximum gross head as shown in ***Figure 15***. The pump discharge valve needs to close stepwise in order to handle water hammer and keep operating time quick. A first, fast slope allows reducing the spherical valve opening to a low value while the flow is decelerated, changes direction and is accelerated downstream due to gravity. However, this first closing step hardly affects the discharge behaviour since the throttling of the valve is still low. Now, with the water starting to flow downstream the second slope starts. The first, unavoidable water hammer caused by the natural change of motion already happened and the downstream discharge reaches approximately 80% of rated value. In order to reduce discharge in a smooth way and to avoid superposition of pressure surges caused by Veytaux I and Veytaux II the slope of the second closing time is carefully determined [17].

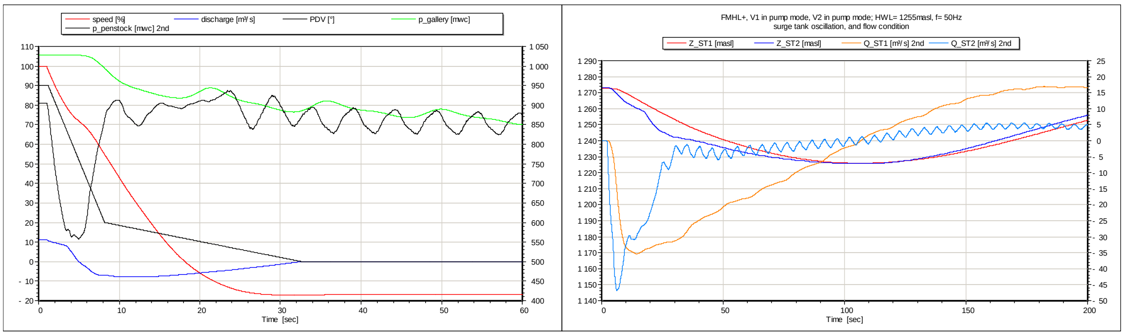


Figure 15: Pumping mode, synchronous load rejection at 3VI + 2VII

Hydraulic short circuit mode

Hydraulic short circuit (HSC) mode is a special feature to be able to vary the power consumption from the grid, to place the power unit on hold, and quickly increase the power consumption on demand. This quick change-over from various operating modes requires a transient analyses, especially in terms of high and low surge shaft oscillations. The maximum surge shaft level is reached at maximum gross head, when both new sets change from full hydraulic short circuit to pumping only in parallel operation with three pumps of Veytaux I. Even though this is the most critical case with HSC involved, the simulation results as shown in ***Figure 16*** do not indicate critical water or pressure levels.

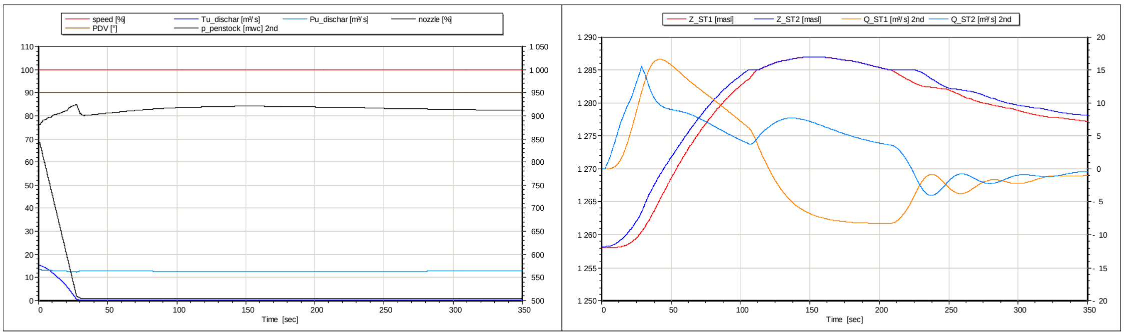


Figure 16: Hydraulic short circuit mode, Turbine close → change over to pumping mode.

6. Conclusion

The extension of the existing pumped-storage plant Veytaux I (4 ternary sets, 60MW each) with two additional sets (Veytaux II, 120MW each set) brings up several tasks regarding hydraulic transients and the development of the new five-stage storage pump.

Even though the operating range of the pump is stretched significantly due to the losses in the penstock and headrace tunnel, used in common by both plants, as well as hydraulic short circuit operation all requirements could be fulfilled. By applying modern methods during hydraulic design and for manufacturing the hydraulic model machine as well as for the measurements and data acquisition all tests could be finished within the tight time-frame. Several optimisation steps including all hydraulically effective parts, the suction elbow, runners, spiral casing and stator blades, lead to the expected increase in efficiency without reducing cavitation properties.

Furthermore, the above issues, i.e. common penstock, hydraulic short circuit operation, as well as a new surge tank are taken into account in the water hammer and transients analysis used to determine operating laws of closing devices. The most critical cases turn out to be reloading after load rejection of all turbines in operation, load rejection with all pumps in operation at maximum gross head and transition from hydraulic short circuit to full pumping combined with pumping of three of the existing pumps against maximum head. For these cases suitable operating laws are discussed and determined to reduce oscillations in the surge chambers to acceptable amplitude.

REFERENCES

[1] E. Pürer, G. Goekler. *The power plant Kops II - Basic conditions, design and details of construction.* Hydro 2005, Villach, Austria, October 17-20, 2005.

[2] F. Avellan. *Evolution des groups de pompage-turbinage.* www.bulletin-online.ch

[3] G. Penninger, F. Spitzer. *New Challenges for Modern Pump Storage Units.* 14th International Seminar on Hydropower Plants, Vienna, 2006

[4] H. Offenhäuser. *Gestaltung von Zu- und Abströmung von großen Kreiselpumpen.* Pumpentagung Karlsruhe 1973.

[5] T. Aschenbrenner, A. Jung, F. Flemming. *CFD-Aided Design of Modern Pump Turbines*. WaterpowerXVI, Spokane Washington, USA, 2009

[6] F. Flemming, T. Aschenbrenner, A. Jung, R.K. Fisher. *Optimization of an adjustable speed pump turbine using unsteady flow simulations.* Hydrovision 2008, Sacramento, CA, 2008

[7] M. Giese, A. Jung, P. Hassler. *PSP Limberg II - optimized design for a wide head range application*. Hydro 2008, Ljubljana, Slovenia, October 6-8, 2008

[8] D. Anciger, A. Jung, T. Aschenbrenner. *Prediction of Rotating Stall and Cavitation Inception in Pump Turbines*. 25th IAHR Symposium Symp. on Hyd. Machinery and Systems, Timisoara, Romania, 2010

[9] R.K. Fisher, C. Powell, G. Franke, U. Seidel, J. Koutnik. *Contributions to the improved understanding of the dynamic behavior of pump turbines and use thereof in dynamic design*. 22nd IAHR Symp. on Hyd. Machinery and Systems, Stockholm, Sweden, 2004

[10] H. Buchmaier, B. Quaschnowitz, W. Moser, D. Klemm. *Numerical optimization of high head pump-turbines*. 18th IAHR Symp. on Hydr. Machinery and Systems, Valencia, Spain, 1998

[11] T. Kaechele, C. Hauff, T. Aschenbrenner. *Discussion of Several Numerical Approaches for the Stator-Rotor Interaction*. 20th IAHR Symp. on Hyd. Machinery and Systems, Charlotte, USA, 2000

[12] M. Sallaberger, P. Bachmann, U. Henggeler. *A New Generation of Multistage Pump-turbines.* 23rd IAHR Symposium, Yokohama, 2006

[13] J.P. Taulan, M. Bourrilhon. *Synthèse des données d'entrées à considerer pour les calculs de régimes transitoires*. Rapport GILHEM n° 4216: version A.

[14] J.P. Taulan, M. Bourrilhon. *Synthèse des résultats des calculs de régimes transitoires*. Rapport GILHEM n° 4217: version A.

[15] C. Nicolet, *Hydroacoustic modelling and numerical simulation of unsteady operation of hydroelectric systems*, Thesis EPFL n° 3751, 2007, (http://library.epfl.ch/theses/?nr=3751).

[16] B. Wylie, V. Streeter, *Fluid transients*, McGraw-Hill International Book Co., 1978

[17] N. Hellstern, *FMHL+ - Transient Calculation Report*, Heidenheim, Germany, 2012

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