Pump to turbine transient for a pump-turbine in a model test circuit and a real size power plant

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Abstract. Ongoing development in variable speed technology for hydraulic power plants improves the flexibility and allows new manoeuvres. In this context, the present work investigates a fast transition from pump mode to generating mode in a reversible pump-turbine by means of CFD and one-dimensional plant dynamic analysis. Calculations are carried for a transition at model scale and finally for a transition in an existing power plant. At model scale, simulation is compared to data measured in the framework of the HYPERBOLE project. For the simulation of the transition at model scale, experimental data for rotational speed and flow rate are prescribed as boundary conditions. To include interaction with the pipe system, the open-source code OpenFOAM® is subsequently coupled to an in-house code for one-dimensional system analysis based on the Method of Characteristics. The coupled method is validated by a comparison to the reference simulation at model scale, and applied to the existing power plant. Results include the evaluation of pressure values and fluctuations at various locations in the machine and the description of different flow phenomena. Furthermore, results for steady operating conditions are compared to the results from transient calculations to identify dynamic effects.

1. Introduction

Hydropower plants are ideally suited to serve as regulating power plant based on their underlying physical principle. Power variations are realized by a change of flow rate for the available head condition. Large power changes imply major alterations of discharge. Thus, hydraulic turbines in hydro power plants are increasingly operated in severe off-design states. Eventually, turbines continuously work in power ranges between no-load and maximum admissible power generation. The corresponding flow rates are controlled by guide vane opening for Kaplan and Francis-type turbines or by nozzle position in case of Pelton turbines. Those off-design operation conditions often induce vortical flow fields, e.g. leading edge and gap vortices as well as part load and overload vortices, activating high dynamic loads on the mechanical structure.

Pump storage plants are equipped with pump-turbines or with a separate pump and turbine mounted on a common shaft. While these ternary machine arrangements with two main hydraulic machines relatively easy change their operation condition from pump operation, i.e. maximum power consumption, to maximum power generation, classical pump-turbine equipment does not include that capability. Ternary machine sets such as Kops II in Austria (six jet Pelton turbines and three-stage
pumps) are designed to operate in hydraulic short circuit mode allowing fast power changes between pump and turbine limits. However, such operation mode requires a pipe layout of the main water passageway designed for that operation purpose.

The development of full size frequency converters up to about 500 MVA enable classical pump-turbine equipment containing synchronous Motor-Generator to also operate with variable speed technology [1]. From an electrical point of view, the machine is able to run at any speed, with a large torque already available at standstill for a fast start-up in pump mode without blow-down of the water within the impeller. Moreover, a fast change from pump mode to generating mode, and vice versa, can now be performed without disconnecting the machine from the grid.

The prediction of loads for such wide off-design operation conditions serves as basic input to assess the impact on changes of equipment life time. For load prediction on hydraulic machines, typically numerical flow field simulation methods (CFD) are applied considering the complete hydraulic passage of the machine [2]. Considering geometry details such as gaps [3] or improved physical modelling for cavitation phenomena [4] improves the accuracy of load predictions. Owing to the unsteady behavior of these flow fields within the off-design operation regime an interaction between the hydraulic machine and the attached piping system will occur. That interaction mechanism needs to be considered for further increase of accuracy, too.

The scope of this work is to investigate a fast transition from pump mode to generating mode in a reversible pump-turbine considering the interaction between pump-turbine and the complete piping system. In the case of the real size power plant the piping system includes the surge tank and a parallel unit as well as conditions for head and tail water reservoir. For the closed-loop model test rig, all relevant pipes and service pumps are considered. The complete pump-turbine, spiral case inlet to draft tube exit, is always simulated by CFD methods, while the piping system with all its components is described and calculated with an in-house one-dimensional model based on the Methods of Characteristics (MoC) typically used for plant dynamic investigations, e.g. [5]. The investigated operation transient contains pump mode, pump brake mode and turbine mode presuming complex three-dimensional unsteady flow fields within the machine. Selected preliminary results for CFD simulations of the pump-turbine along the four-quadrant characteristic are presented in [6, 7] neglecting pipe coupling.

The inclusion of pipe coupling for such transients creates a better understanding of the entire system behavior being able to provide hints for some open questions, e.g. (a) Is the steady state four-quadrant characteristic representative for transient conditions? (b) How can results from model test measurement be transposed to prototype conditions, and vice versa? (c) What are the differences to a pure one-dimensional analysis? Another question dealing with the possibility whether a closed-loop test rig is able to duplicate transient conditions of a prototype is addressed in another publication [8].

2. Similitude under transient conditions
The transposition of model test data to prototype data for steady state operating conditions relies on geometric and kinematic similitude. Geometric similitude applies a constant scaling factor to the wetted surfaces thereby fixing the transposition factor for the length scale. Kinematic similitude between two machine sizes guarantees that the velocity triangles at the runner are similar. One set of possible dimensionless similarity parameters are unit speed and unit flow, equation (1).

\[ n_{ed} \equiv \frac{n D}{\sqrt{g H}} \quad \text{and} \quad Q_{ed} \equiv \frac{Q}{D^2 \sqrt{g H}} \quad (1) \]

The operation condition of a hydraulic machine is characterized by this parameter set indicating similar velocity triangles for same pairs of \(n_{ed}\) and \(Q_{ed}\). Once a rotational speed is defined for model testing and prototype application, the scaling factor for time is fixed between both machine sizes because of the geometry scaling factor. This factor scaling time may be used for the transposition of frequencies, e.g. pressure fluctuations.
From a purely kinematic point of view, variable scaling factors for time are also acceptable during the transient from pump mode to generating mode, i.e. the ratio \( \frac{n_M}{n_p} \) is allowed to vary as long as all points on the characteristic are passed. In this case, each operating point in the model still has a corresponding state with similar velocity triangles in the prototype, although these states are not reached at the same relative time \( t/T \).

Dynamic similitude considers similarity of force polygons between model and prototype [9]. The unsteady momentum equation contains five different forces, namely forces from steady and unsteady inertia, gravity forces, pressure forces and viscous forces [10]. A relevant ratio for hydraulic machinery is the one between forces caused by steady inertia and pressure forces, called Euler number \( Eu \) or pressure coefficient \( c_p \) [9], equation (2). That equation is fulfilled because of the application of similarity parameters, equation (1), geometric similarity and same specific speed of both sizes.

\[
Eu = \frac{\Delta p}{0.5 \rho u^2} = c_p
\]  

Replacing velocity \( u \) in equation (2) by the mean flow velocity at the runner outlet as calculated from \( u = Q/A_{ref} \) and using equation (1) with equal \( Q_{ed} \) for model and prototype results in equation (3) for steady state conditions.

\[
\frac{\Delta p_p}{\rho g H_p} = \frac{\Delta p_M}{\rho g H_M}
\]

Under transient conditions, pressure forces are additionally affected by unsteady inertia. To guarantee that the pressure coefficients are nonetheless applicable, the ratio of forces from unsteady inertia to pressure forces is considered. As the model and prototype have Euler similitude, this is equivalent to comparing the force from steady to unsteady inertia, given by the Strouhal number, equation (4).

\[
St = \frac{u_{ref} \Delta t}{l} = \frac{n \pi D_{ref}}{u \Delta t}
\]

For the current case of a fast transition from pump mode to generating mode, the reference velocity is taken to be the peripheral velocity of the runner at its outlet, i.e. \( u_{ref} = n \pi D_{ref} \), while the timescale \( \Delta t \) is equal to the transition time \( T \) and reference length \( l = D_{ref} \). Enforcing Strouhal similitude between model and prototype is equivalent to requiring \( n_p/n_M = T_M/T_p \) under these circumstances. This condition is an important limitation regarding the choice of rotational speed for the model test. Once the ratio of \( n_M/n_p \) or alternatively \( T_M/T_p \) has been fixed, it must be maintained throughout the transient. As a further consequence the unit speed, equation (1), requires that the model head provided by the test rig must be in a fixed ratio to the one of the prototype. This is the main difference to steady state model tests, where head or rotational speed for each new operating point can be chosen at will as long as the combination leads to the required dimensionless values. Failure to comply with Strouhal similitude will lead to different pressure coefficients between model and prototype due to different contributions from unsteady inertial forces, which may be negligible in cases where gradients over time are small.

Additional restrictions may be introduced by enforcing Froude similitude, the ratio between steady inertia and forces caused by gravity, equation (5). Similitude in Froude defines the value for the time scales to be \( T_M/T_p = \sqrt{D_M/D_p} \) [9] and leads to equal pressure gradients in space within the machine [11]. Compliance with Froude similitude is often difficult to achieve as it will lead to very low or very high head in model tests for large capacity and high head turbines, respectively [10]. In this work, it is employed in CFD only.

\[
Fr = \frac{u_{ref}}{\sqrt{g \rho u_{ref}}}
\]
Changes in flow rate and rotational speed during the transient affect head at the pump-turbine, causing a deviation between the steady state characteristic and the transient curve in a four-quadrant characteristic if the measured head is used for calculation of $n_{ed}$ and $Q_{ed}$.

In a straight pipe of length $L$ with constant cross section $A$, the pressure difference from the change of flow rate can be deduced from the forces from unsteady inertia, which are equal to the fluid mass $\rho AL$ times the acceleration according to Newton’s second law [12], equation (6). For pumps and turbines, an equivalent length may be used in combination with a reference area [13, 14]. In addition, Acosta del Carpio accounts for the effects of the change in rotational speed and summarizes the contributions from the different components, resulting in the total head at a specific time, equation (7). The constants $K_{hn}$ and $K_{hq}$ are derived from geometry and scale between model and prototype according to $K_{hn} \sim D^2$ and $K_{hq} \sim D^{-1}$. If model and prototype comply with $n_P/n_M = T_M/T_P = t_M/t_P$, inserting these relationships into equation (7) shows that the ratio between dynamic head contribution and head from the steady state characteristic is equal in model and prototype.

$$\Delta H \rho g A = \rho AL \frac{dQ/A}{dt} \iff \Delta H = \frac{L}{g} \frac{dQ}{dt}$$

$$H = K_{hn} \frac{dn}{dt} + K_{hq} \frac{dQ}{dt} + H_{steady}(n, Q)$$

The fact that $H_{steady}(n, Q)$ is included in equation (7) is a result of the assumption that the steady state pump-turbine characteristic is valid under transient conditions. This is a common approach, compare e.g. [15, 16].

3. Numerical setup of pump-turbine and piping system

The investigated pump-turbine is shown in Figure 1. A full 3D computational mesh is generated from spiral case inlet to draft tube exit. Detailed views present the location of pressure sensors used during the model test. Measurements were carried out for steady state conditions as well as for a transient operation change.

![Figure 1. Pump-turbine model and location of pressure sensors.](image-url)
The current Computational Fluid Dynamics (CFD) simulations rely on OpenFOAM® 2.3 software. It is based on the Navier-Stokes equations for incompressible flow with a k-ω SST turbulence model. A block structured mesh with approximately 20 million nodes is generated for the analysis. The geometry is split into four domains, spiral case (SC), twin cascade (SV/GV), runner (RUN) and draft tube (DT). Domains are connected with each other via the “arbitrary mesh interface” (AMI) for non-matching grids. During the meshing process, special attention is paid to keeping all cells close to the walls comparable in the guide vanes and the runner. Average y⁺ values are between 20 and 65 in the runner and between 20 and 50 in the guide vanes, depending on the operating point.

Flow rate is prescribed at the respective inlet, i.e. at the draft tube outlet in pump mode and at the spiral case in generating mode, together with a zero gradient condition for pressure. At the remaining outlet, a constant average pressure and a zero gradient condition for velocity are applied. Time varying values for flow rate and rotational speed during the transient are prescribed via look-up tables.

For the entire piping one-dimensional system analysis with Methods of Characteristic (MoC) computation software is carried out via an in-house code, which includes a large number of components common to hydraulic power plants [17]. The basic governing unsteady equations (mass and momentum) are modelling compressible liquids typically used for plant dynamic analysis to investigate transient operation changes of hydro power plants.

For the coupling of CFD and MoC, the conventional treatment of a hydraulic machine in a pipe system requires head and flow rate of the machine to be compatible with those of the adjacent grid points [16]. This means that H and Q at the coupling interface need to be equal for both codes within a certain error limit for any given time step. From this condition, head values for both ends of the 1D code can be derived provided a head difference from an external code is known. Flow rate is then calculated using the internal equations from the MoC. For simplicity, the same time step is used for both systems. A coupled simulation allows identifying if and how the pump-turbine interacts with the test rig or the real power plant piping.

4. Power plant scheme, model test rig and transient condition

The fast transition from pump mode to generating mode in a reversible pump-turbine is displayed in a four-quadrant plot, Figure 2.

![Figure 2](image)

The transient at model scale investigated in this work is defined with a constant guide vane opening of 25° being the same setting as in the steady state operating points. A large guide vane opening offers a maximum difference between power consumption in pump mode and power generation in turbine mode. Transition time is chosen to be T = 8 s and the start of the transition is at 1 s.

Variation of rotational speed is linear from a stable operating point in pump mode (n_{ed} = −0.35) to stable operation in generating mode (n_{ed} = 0.3). At the same time, also the test rig pump undergoes a linear variation of rotational speed. This combination leads to the development of head and flow rate presented in Figure 3 (left). Head experiences a significant drop within the first two seconds, followed
by a rather constant recovery during the remaining period. The evolution of flow rate is roughly divided into two phases, a period of rapid changes up to $t = 4.4$ s and a more moderate increase in the second half of the transient. According to a 1D plant simulation, this behaviour is comparable to the evolution at prototype scale [18]. The characteristic head drop and the fast reversal of flow rate are also consistent with the events during pump failure as described by e.g. [15, 16].

A representation of the transient in a four-quadrant characteristic is depicted in Figure 3 (right), assuming the transient follows the steady state characteristic. The transient starts in the lower left corner with negative flow rate and negative rotational speed, i.e. normal pump mode. As the rotational speed decreases, the machine passes to the next quadrant. Flow direction is now from the spiral to the draft tube, while the runner continues its rotation in the same direction as before (pump brake or dissipation mode). Finally, the runner reverses its rotational direction and the upper right quadrant is reached. This represents normal generating mode.

Figure 4. Numerical model of the test rig. Each labelled box represents a continuous part of the test rig with pipes of constant cross section.
The test rig for the model tests consists of three parallel branches, with the pump-turbine model located in the first one, Figure 4 [19]. In the second branch, a variable speed pump provides the head for the pump-turbine, while the third branch includes a perforated plate for energy dissipation. While the pump-turbine is running in generating mode, the speed of the service pump is increased to maintain a steady flow through the perforated plate. Flow rate is measured by a magnetic-inductive flow meter suitable for transient conditions.

The power plant analysed in the present work is described in [18, 20] and schematically shown in Figure 5. It has two pump-turbines homologous to the tested model with a nominal turbine power of 2 x 210 MW. Each of the units is linked directly to the upper reservoir via a separate short penstock. On the downstream side, the two separate tail race tunnels share a common surge tank and reunite after a certain distance before ending in the lower reservoir. Nominal rotational speed is 200 rpm with a turbine nominal net head of 133.5 m and a nominal discharge of 175.1 m$^3$/s. The length ratio between prototype and model is $D_P/D_M = 15.82$.

The lower pump-turbine PT 1 is represented by its steady state characteristic and runs in generating mode at the nominal rotational speed of 200 rpm. The upper unit PT 2 undergoes the 32 s transition from pump mode to generating mode and is modelled by CFD in the coupled analysis. Upper and lower reservoirs maintain fixed water levels throughout the transient.

5. Results
For both, model and prototype setup, pump steady state conditions have to be determined. Once the transient starts, the rotational speed of the pump-turbine is varied according to Figure 3 for model and prototype. For the model, in addition, the rotational speed of the service pumps is adapted. In both cases head and flow rate at the pump-turbine as well as within the pipe system is now an additional outcome of the simulation and no longer prescribed. Results based on measured data, i.e. CFD simulation only without pipe system, are presented in [6, 7]. Here, the focus is on coupled simulation result presentation.

This section presents the results of the prototype simulation, with the results from model scale included in the graphics for comparison. The time scales are chosen such that the ratio of $n_P/n_M$ is equal for two arbitrary vertical lines between 0 and $t_P=33.5$ s, which means that the 32 s for the prototype transient correspond to 8.6 s for the model transient. At $t_M=9$ s ($t_P=33.5$ s), the model is at $-0.86 n_0$. Therefore at equal $dn/dt$, 8.6 s are required to reach $-n_0$ as in the prototype transient. This
results in a scaling factor of $t_M = 3.72 \ t_P$. As the model transient starts at 1 s, the prototype transient is set to start at 3.72 s.

![Figure 6](image.png)

**Figure 6.** Global results, prototype versus test rig. Red / blue dashed lines mark $Q=0$ for model / prototype, black dashed line marks $n_P = n_M = 0$, black solid line signals the end of constant $n_P / n_M$.

*Figure 6* summarizes the results for head, flow rate and torque together with a representation in the four-quadrant characteristic. The latter one does not contain a time scale and shows a very good agreement between the experiment at model scale and the results in the power plant. However, the remaining three figures clarify the differences.

The most important of these is the difference between the head curves. After a similar start with decreasing head, the prototype simulation starts to recover earlier than the model simulation, leading to a less pronounced head loss. As the head minimum for both curves coincides with the end of pump mode, this implies that the reversal of flow rate is faster, which can be verified in the neighbouring figure. The following increase is comparable in slope for both cases, but ends for the prototype at a head slightly over the final value in generating mode. Consequently, while the model test shows a steady increase in head for the rest of the transient, the prototype head falls again until reaching a plateau with the start of generating mode. Flow rate in the prototype shows a steeper slope during the first part of the transient, which results in the above mentioned earlier transition from pump mode to pump brake mode. Also, the change from fast to moderate increase occurs before the one in the model test. In the second part of the transient, the plateau in the head curve is matched by a slow, continuous increase in flow rate.

The different behaviour between model and prototype in head and flow rate is attributed to the difference between the test rig setup and operation mode and the behaviour of the fluid in the power plant. The test rig setup with a linear variation of rotational speed of the test rig pump does not
reproduce the head curve and thus, flow rate found in the power plant. This failure to follow the dynamic head curve of the prototype is associated with a failure of reproducing the steady state head at each point of time. While this is no problem for steady state operating tests, the direct comparison of the transients with the chosen time scales is now questionable. On a vertical line, model and prototype still present the fixed speed ratio \( n_P/n_M \), but are no longer in kinematic similitude due to the different steady state head values. For subsequent model tests, enforcing a head curve identical to the one expected from the prototype should therefore be considered. This requires a simulation of the power plant to obtain the head curve to reproduce, and additional simulations of the test rig to optimize the rotational speed curve of the test rig pump until the desired head curve at the pump-turbine is obtained [8].

![Figure 7](image.png)

**Figure 7.** Comparison of simulation and measurement for the guide vane sensor pm06.

Figure 7 presents the results at the Rotor-Stator Interaction (RSI) sensor pm06. Comparable to the results for the model test, the mean curve closely follows the head curve, with a higher minimum at the transition from pump mode to pump brake mode and larger mean values throughout pump brake mode. The first fluctuations appear shortly before the transition from pump mode to pump brake mode and are attributed to a short period of rotating stall, detected by the RSI sensors (not shown). The largest fluctuations appear in pump brake mode, with blade passing frequency being the dominant factor. In generating mode, fluctuations continuously lose intensity until reaching the final values.

Results at the different sensor locations in the runner are shown in Figure 8. With the exception of pm17, all curves are similar to those obtained from the model tests, with the same characteristic intervals for which fluctuations appear in each of the locations. On the leading edge, both sensors show the first fluctuations as the flow in the guide vane channels and the vaneless space becomes unstable in pump mode. On pressure side, this is followed by a sharp increase of pressure in pump brake mode, accompanied by large amplitude fluctuations. The main contributions are at stochastic frequencies near the runner frequency (not shown). The small decrease in head between 17 s and the beginning of generating mode and the change in the slope of the flow rate curve (Figure 6) translate to a significant drop in pressure at pm17. Compared to the model test, the prototype simulation shows higher pressure gradients as well as a higher maximum value in pump brake mode. In generating mode, the signal shows a steady decrease until reaching its final value.

On the trailing edge, the highest fluctuations are again found on the pressure side. Besides the large amplitude fluctuations in generating mode caused by the vortex travelling through the runner channel, large amplitudes are also found in pump brake mode starting from 11 s with stochastic frequencies. They are caused by a vortex near the pressure side (not shown). Depending on the relative position of the vortex with respect to the pressure side, flow is either attached or detached at the pressure sensor location. This is in agreement with the behaviour found at model scale (not shown), but the onset is earlier, i.e. at lower rotational speed.
Figure 8. Comparison of simulation and measurement for the pressure at the pressure sensors in the runner. Pressure side (left) and suction side (right).

Figure 9. Flow field results within guide vane at t = 2.45 s, compare to Figure 3 for time scale versus operation region relationship.

To also give an impression on the flow field within the pump-turbine during the transient, some selected plots are added here. Figure 9 presents a detailed view on the velocity vectors within some guide vane channels for operation within the pump instability regime. It is clearly visible that the velocity distribution is non-uniform not only between channels but also across channel height. Locally, back flow already occurs. There is no long-term stable rotating stall phenomenon as the rotational speed continuously changes [6].
Streamlines for the local velocity field within the runner at mid channel height are presented at different points in time, Figure 10. At the beginning of the transient very smooth streamlines indicate high quality flow behaviour. Within the pump instability regime at $t=3\text{ s}$ with flow in pump direction, a non-uniform behaviour with strong vortices is clearly visible. At times $t=4\text{ s}$ and $t=5\text{ s}$ runner rotation is still in pump direction while flow direction reversed to turbine direction resulting in large channel vortices. These vortices gradually decrease with progressing time until reaching a very smooth streamline distribution close to the targeted steady state turbine mode. Such vortical flow field is responsible for the dynamic pressure distribution presented in Figure 8.

![Figure 10](image-url)

**Figure 10.** Flow field results within runner at mid channel height in model test at indicated points in time, compare to Figure 3 for time scale versus operation regime relationship. Time difference of about one second from top left to bottom right starting with close to steady state pump mode ($t=1.1\text{ s}$) and ending with steady state generation mode ($t=9\text{ s}$).

6. **Conclusion**  
The present work investigates a fast transition from pump mode to generating mode in a reversible pump-turbine with a constant guide vane opening and a linear variation of rotational speed. A coupled model including the pipe system and the service pump of the test rig is set up and validated. The methodology is then applied to a real power plant with two units at 210 MW each. Although the
transient in the power plant is faster than the tested one considering similitude laws, the same flow structures are found. Simulation is found to be in good agreement with measurement considering head and several pressure sensors in the vaneless space, the runner and the draft tube.

Looking at head and pressure in the machine, dynamic effects are clearly identifiable as the values are shifted to a different level, even where general flow phenomena are comparable. However, the differences between steady state results and transient results are small compared to the difference between the operating points.

For a more detailed experimental investigation, test rig conditions should mirror the real power plant. This implies that a fixed time scale is to be used, either defined by the initial ratio of rotational speeds $n_{M,0}/n_{P,0}$ or by the ratio of transition times $T_M/T_P$. Equal dynamic contributions in model and prototype are obtained for $n_{M,0}/n_{P,0} = T_P/T_M$. Furthermore, the test rig should be able to reproduce the (scaled) real head curve. This requires a-priori knowledge of the head curve in the power plant, e.g. from one-dimensional system analysis, and a control unit able to prescribe arbitrary rotational speed at the test rig service pump and the pump-turbine model.

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