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To cite this article: J Junginger *et al* 2019 *IOP Conf. Ser.: Earth Environ. Sci.* **240** 082010

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Transfer of transient conditions from prototype to closed-loop model test rig

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Abstract. This paper describes a method for determining the sequence of transient model test experiments in a closed loop test-rig, using previous 1D-simulations. A real power plant with reversible pump turbine in the Austrian Alps serves as baseline. A fast transition with load rejection and guide vane closing has been simulated, as it is state of the art. Furthermore, it was assumed that this power plant also has a full size frequency converter for fast transition from pump mode to turbine mode and vice versa applying linear speed variation. These simulation results were transferred to the scaled model size via similarity laws. Of particular importance is the choice of model to prototype speed ratio whose influence has been studied more closely. Subsequently, the transient processes are simulated in a closed loop test-rig. The simulations are controlled by a temporal power variation of the service pump. An iteration loop with optimizer has adapted the power in order to achieve similar conditions as in the prototype simulation. The findings of the pump behavior can then be used for the transient model test experiments.

1. Introduction

The increasing share of renewable energies requires flexible operation of pumped storage power plants [1]. Variable-speed units with doubly fed induction generators increase the flexibility for a better control of power range, especially during pump mode. Further, current research assumes that future power plants will be equipped with full frequency converters. These units allow fast transition maneuvers without disconnecting from the grid or dewatering the impeller [2].

To operate these power plants safely transient processes are investigated by 1D simulation. However, these 1D simulation studies provide no information about the actual mechanical loads of machine components and pressure fluctuations during a transient process. Transient loads within the hydraulic machine are possible to predict by coupling 1D pipe simulation with 3D CFD simulation [3].

Transient model test experiments provide the possibility to replicate the transient behavior of the prototype hydro power plant. With the results of these experiments it would be possible to optimize the behavior of a fast transition and allows a better determination of the lifetime of the components for the increasing number of load changes [4]. As the main piping system is completely different between a closed-loop model turbine test rig and a real hydro power plant (Figure 1), the behavior of pressure wave propagation in water will be significantly influenced. Thus, the challenge is to transfer transient conditions from the power plant to conditions at the model turbine within a closed-loop test rig considering similarity rules.

The HYPERBOLE European Research Project has investigated the feasibility of fast transition in reduced scale model tests and numerical simulation [5], [6]. In this present paper an approach is shown



how to replicate the transient behavior of a fast transition from a 1D-power plant simulation in a closed-loop model test rig. The simulation is controlled by the electric power input of the service pumps. The aim of this work is to find an appropriate behavior of the test rig pumps which can then be transferred to the model test experiment.

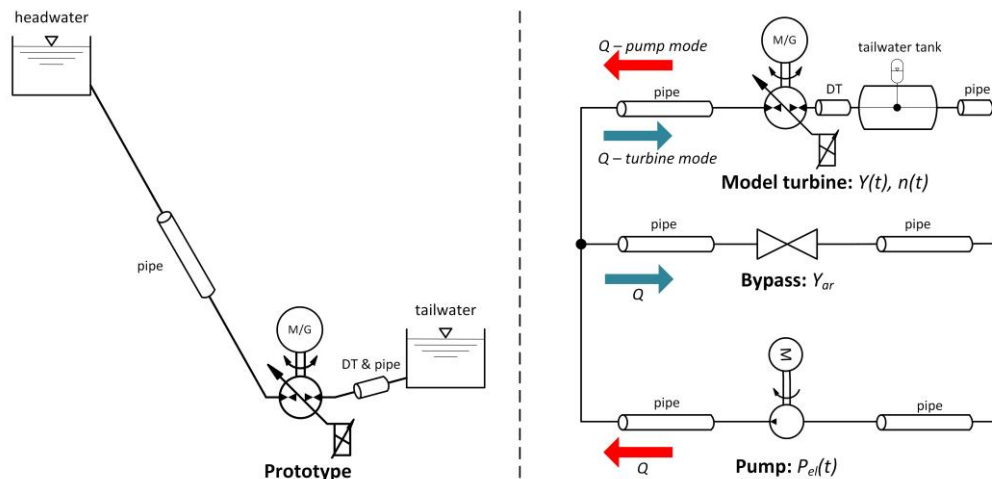


Figure 1 Piping system of a hydro power plant (left) and closed loop test-rig with bypass (right).

2. Preparation for the simulation of fast transitions in closed loop test-rig

The prototype data of the transient processes which should be simulated in the test-rig come from 1D-power plant simulations of an existing pumped storage power plant (about 280 MW) with a reversible pump turbine. The power plant calculations as well as the test-rig calculations were carried out with the in-house program SIPROHS (simulation program of hydraulic systems) developed at IHS [7]. The program is based on the Method of Characteristics (MoC) typically used for plant dynamic investigations [8] and has been used and further developed at the Institute for many years. For these simulations, two different concepts of a fast transition are assumed.

2.1. Fast transition concepts

The electrical equipment of a hydro power plant pretends the concept of fast transition. In the current state of the art, the generator of a reversible pump turbine is synchronously connected to the electrical grid or designed for marginal speed variation around the rated speed as doubly fed induction machines (DFIM). The concept of fast transition from pump mode to turbine mode is realized by load rejection that leads to a reversal of the rotation direction. At the same time, the guide vanes are closing to a position achieving synchronous speed in turbine rotation. After synchronization with the grid, the guide vanes open again to hit the specified turbine power. For more flexibility and faster transition times, current development concepts attempt to equip pump turbine units with a full-size frequency converter (FSFC). A more detailed description can be found in [9]. With these concepts a disconnection from the grid or the closing of the guide vanes are not necessary. In addition to the power control in pump mode, one of the biggest advantages is the possibility of a fast transition from turbine mode to pump mode.

Figure 2 compares the fast transition with FSFC (red lines) and synchronous speed DFIM units (blue lines). The left diagram shows guide vane opening Y (dashed) and rotational speed n (solid) versus dimensionless time of the transition. For fast transition with FSFC a linear change of rotational speed is assumed. Additionally, the reverse fast transition from turbine mode to pump mode for FSFC units (red solid line) is included. The right diagram shows the behavior of the operating point in the $n_{11} - Q_{11}$ hill chart.

For the future development of reversible pump turbines both concepts will be relevant. Therefore, the method described in this work for simulating fast transitions in closed-loop test rigs should be generally applicable.

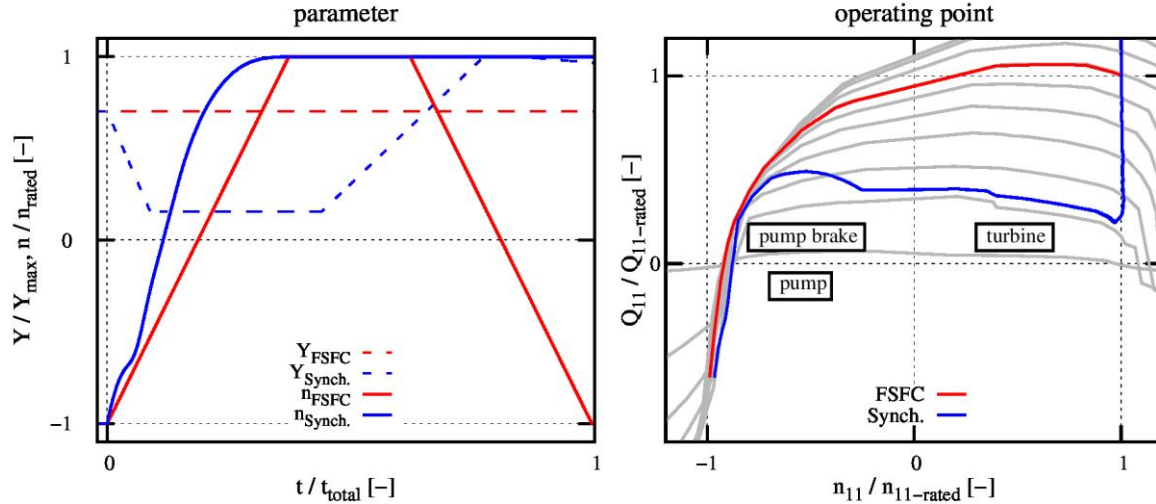


Figure 2 Comparison of fast transition with (blue lines) and without (red lines) full-size frequency converter. Left side: variation in time of guide vanes (dashed lines) and rotational speed (solid lines); right side: operation points in the $n_{11} - Q_{11}$ hill chart.

2.2. Transfer of prototype data to model scale

The prototype simulation of a fast transition delivers time-dependent parameters as head H , discharge Q , rotational speed n and torque T . The transfer of these parameters from prototype to reduced scale model is realized by similarity laws, equations (1) - (3).

$$n_{11} = \frac{n \cdot D}{\sqrt{H}} \quad (1); \quad Q_{11} = \frac{Q}{D^2 \cdot \sqrt{H}} \quad (2); \quad T_{11} = \frac{T}{D^3 \cdot H} \quad (3)$$

At each point in the simulation, it must be assumed that the characteristic values of the prototype match those of the model machine: $n_{11-proto}(t_{proto}) = n_{11-model}(t_{model})$, $Q_{11-proto}(t_{proto}) = Q_{11-model}(t_{model})$ and $T_{11-proto}(t_{proto}) = T_{11-model}(t_{model})$. Particularly important is the time scale of the transient process, which must be adapted. This is done with the Strouhal number Sr where the characteristic length L is represented by the impeller diameter D and the characteristic speed v is replaced by the circumferential speed u (4).

$$Sr = \frac{L}{v \cdot t} = \frac{D}{u \cdot t} = \frac{D}{r \cdot \omega \cdot t} = \frac{2}{\omega \cdot t} \quad (4)$$

According to (4) the time scale can be adjusted by the ratio of the rated speed of the Prototype n_{proto} and the initial speed of model turbine n_{model} (5).

$$\frac{\omega_{model} \cdot t_{model}}{\omega_{proto} \cdot t_{proto}} \rightarrow \frac{\omega_{model}}{\omega_{proto}} = \frac{n_{model}}{n_{proto}} = \frac{t_{proto}}{t_{model}} = x \quad (5)$$

The transient operation change runs x -times faster in a closed-loop test rig than in prototype scale, for a chosen initial speed of the model turbine. The speed ratio also determines the head of the turbine model, since the diameter ratio of model turbine and prototype is known, equation (1).

$$\frac{n_{proto} \cdot D_{model} \cdot \sqrt{H_{model}}}{n_{model} \cdot D_{model} \cdot \sqrt{H_{proto}}} \rightarrow \left(\frac{n_{model} \cdot D_{model}}{n_{proto} \cdot D_{proto}} \right)^2 = \frac{H_{model}}{H_{proto}} \quad (6)$$

Figure 3 shows the effect of different time scale factors x to rotational speed and head. Higher x -values lead to higher speed gradients and to higher pressure gradients, too. But this also results in higher Reynolds numbers being closer to prototype reality.

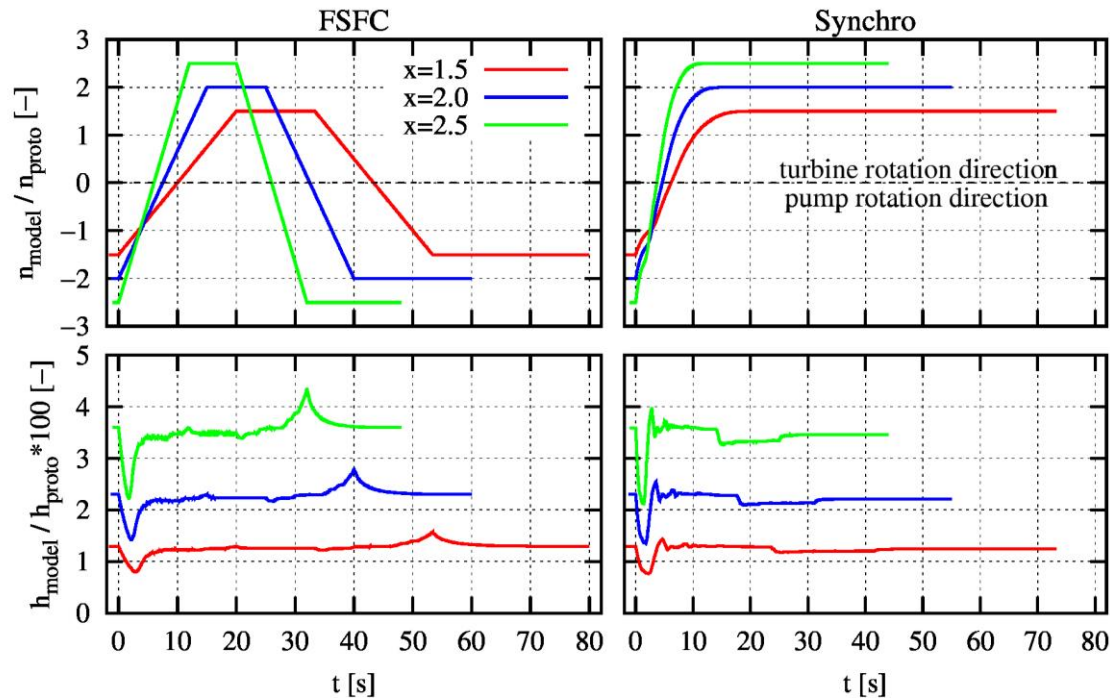


Figure 3 Transfer of rotational speed and head ratio from Prototype to model turbine by varying the initial rotational speed of the model turbine.

With the same procedure using equations (2) and (3) the model discharge and the torque can be calculated. Thus, all relevant parameters for the closed loop test-rig simulation are known.

2.3. Closed-loop test rig simulation model and solver

For the transient simulations, the closed-loop test rig at the Institute of Fluid Mechanics and Hydraulic Machinery (IHS) is modeled with all important properties like pipe dimensions, speed of sound, losses, etc. The configuration is schematically shown in Figure 1. The test rig contains three connected branches for model turbine, service pump and bypass. The bypass allows a discharge reversal at the model turbine during the transition from pump to turbine operation. To hold the head at the turbine, the bypass valve must be partially closed. The opening position Y_{ar} must be appropriately selected that the discharge of the model turbine and the pump can be processed during pump mode. Since the valve cannot respond to the change in discharge quickly enough, Y_{ar} remains constant over the entire simulation.

In the simulation, the set-point values for guide vane opening $Y(t)$ and rotational speed $n(t)$ for the model turbine are implemented, Figure 3, as well as the characteristic of turbine and pump.

The simulation is controlled by the electric power $P_{el}(t)$ at the motor of the service pump. This power can be set for each time step to reach the given set-points for head H and discharge Q at the model turbine. The temporal power variation is determined by an optimization which uses the properties of a controller.

3. Optimization via controller structure

The structure in Figure 4 displays the iteration loop to achieve the ideal pump power to reach the given set-points. This structure contains two main programs. The solver, which runs the transient simulation, and the optimizer, which adjusts the pump power for each iteration. The loop starts on the left by comparing the actual and set point values of head on the model turbine in each time step over the entire simulation. The electrical power correction at the pump $\Delta P_{el}(t^*)$ is adjusted by the optimizer being influenced by the deviation $e(t)$. To avoid unrealistic power jumps, this value is restricted by an arbitrary maximum value. The corrected value $\Delta P_{el}(t^*)_{i+1}$ is added to the previous power value $P_{el}(t^*)_i$ and then passed on to the solver. The transient process is simulated again with the new electrical power of the pump. If the deviation $e(t)$ falls below a certain limit in each time step the iteration loop stops.

Since water is assumed to be a weakly compressible fluid and the moment of inertia of motor and pump is taken into account in the simulation, a change in the electrical power does not result in an immediate change in head at the model turbine. Therefore, the timing of power input must be corrected to an earlier time step $t^* = t - \Delta t$. The value for Δt is determined via parameter studies.

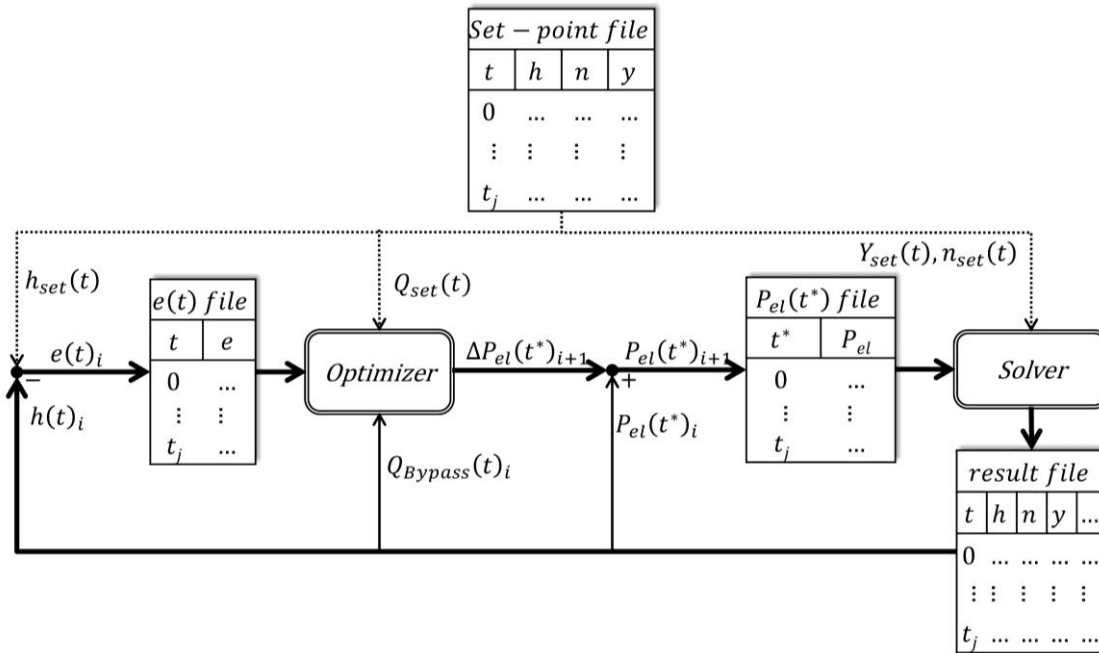


Figure 4 Iteration loop structure for determining the optimum pump power variation in time

A detailed look at the optimizer structure is shown in Figure 5. This starts on the left with a decision whether the relative deviation of head in a certain time step exceeds a fixed limit. If not, the power will not be adjusted. Otherwise a PI-controller with a proportional and an integral term modifies the signal. The result is a modified pump head $\Delta h_p(t)_{i+1}$, according to equation (7). Stable parameters for K_p and T_n were determined by parameter studies. With the modified pump head, the set point discharge $Q_{set}(t)$ and the discharge in the bypass $Q_{Bypass}(t)_i$, the adaptation of the power $\Delta P_{el}(t^*)_{i+1}$ can be determined as hydraulic power, equation (8).

$$\Delta h_p(t)_{i+1} = K_p \cdot \left[e(t)_i + \frac{1}{T_n} \cdot \int_0^t e(\tau)_i d\tau \right] \quad (7)$$

$$\Delta P_{el}(t^*)_{i+1} = \rho \cdot g \cdot [Q_{set}(t) + Q_{Bypass}(t)_i] \cdot \Delta h_p(t)_{i+1} \quad (8)$$

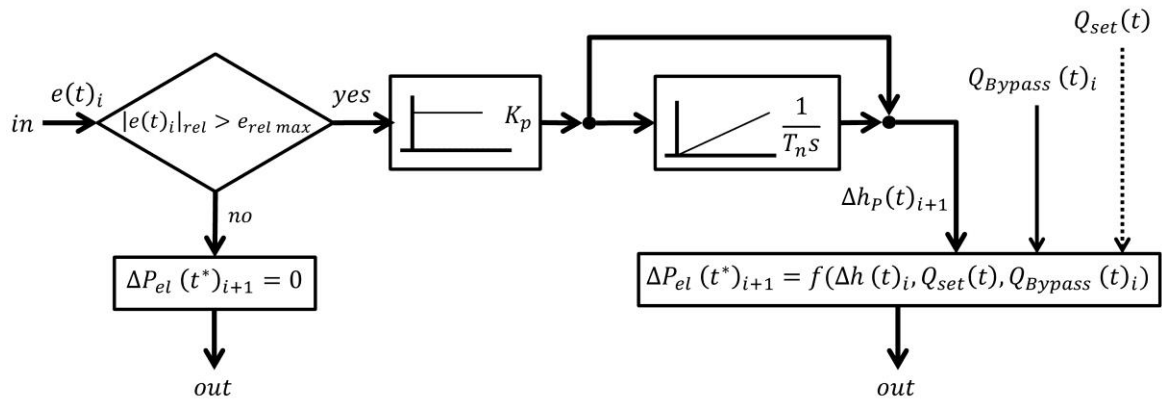


Figure 5 Structure of the optimizer for adjusting the pump power

4. Simulations and results

All six cases shown in Figure 3 were investigated. Depending on the case, the optimizer needs between 200 and 500 iteration loops until the permissible minimum error of $e = 0.5\%$ has been reached in each time step. The power generated by the optimizer show some fluctuations. To obtain a feasible speed curve for the model test experiments, all power curves were smoothed with a moving average over about one second and then calculated by the solver one last time. The results of these simulations are evaluated separately for two different fast transition concepts. Decisive for the quality are on one hand, how exactly can the test-rig simulations reproduce the fast transition and on the other hand, whether the behavior of the service pump can be transposed in the experiment.

4.1. Results of FSFC fast transition

Figure 6 (left) shows power curves normalized with maximum pump power for varied x -values after smoothing. Speed curves of the service pump normalized with maximum speed is displayed on the right. For both diagrams, neither power nor speed show significant jumps or physically unrealistic behavior. But only the experiment will show whether the high power gradients in the first few seconds of the case $x = 2.5$ are possible.

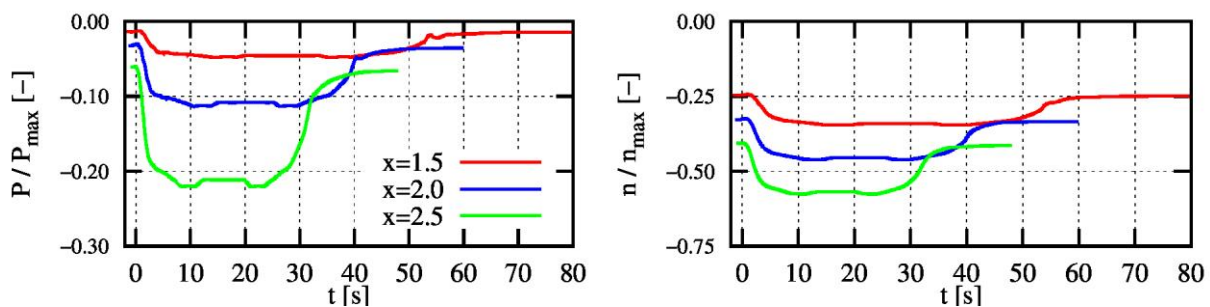


Figure 6 Normalized power and speed of the pump for linear fast transition with different x -values.

Figure 7 (left) shows head (normalized to prototype head) and n_{11} (normalized by rated n_{11} in turbine mode) at the model turbine for various x -values. The colored solid lines represent the simulation results and the dashed black lines the corresponding set point. An optical evaluation indicates no differences. By the help of the relative deviations (solid gray line) it is indicated that the error for all cases predominantly lies within $+/- 1\%$. Within a short time span in the first few seconds of the simulation with dominant pressure change causes the error to rise above 4% for head and -2% for n_{11} (case $x = 2.5$).

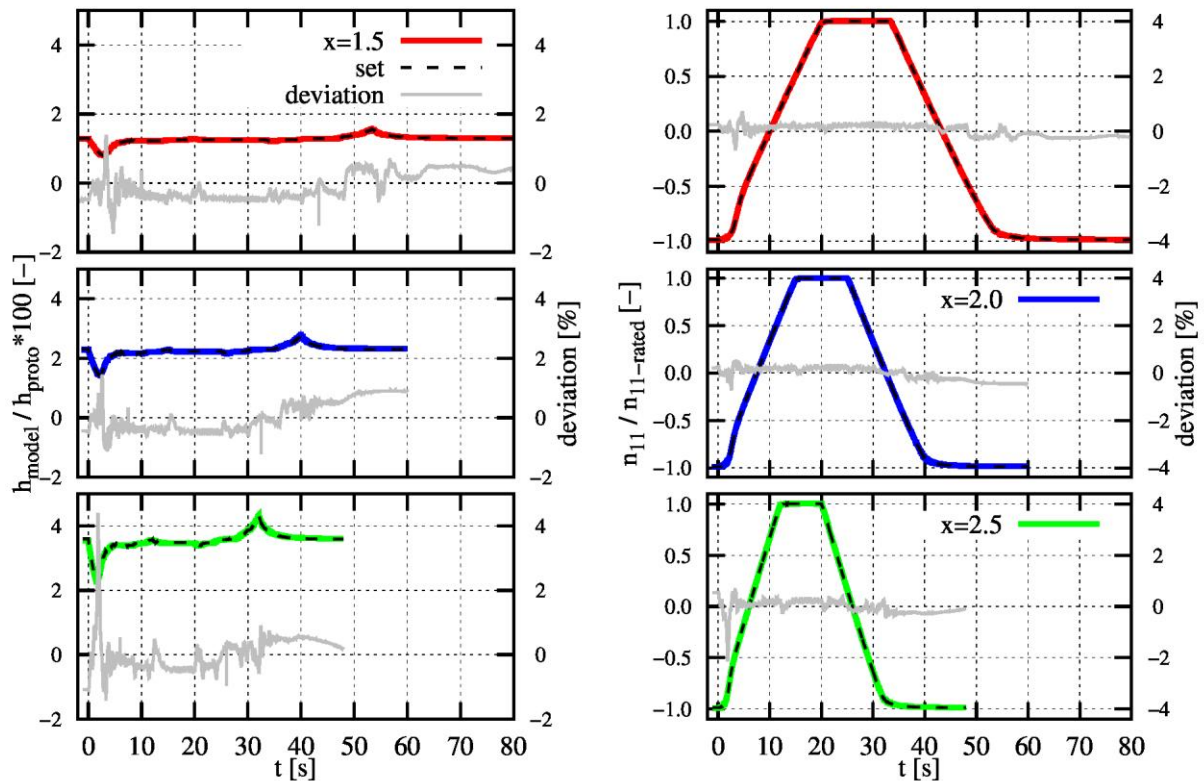


Figure 7 Normalized head and speed n_{11} of the model turbine of linear FSFC fast transition for different x -values.

To evaluate the quality of the results it is also necessary to have a look on the discharge behavior of the model turbine. Figure 8 shows the normalized Q_{11} versus time for cases $x = 1.5$ and $x = 2.5$. Case $x = 2$ is omitted due to a similar manner. By comparing the simulation results and the set point, slight deviations can be detected at the end of the simulations. This is confirmed by the relative error. However, the deviation remains well within $+/- 4\%$ over the entire process.

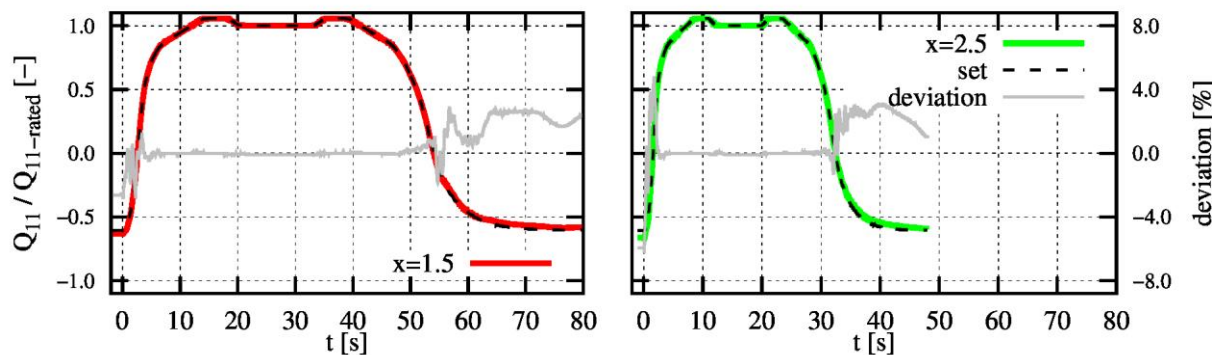


Figure 8 Normalized discharge Q_{11} of the model of linear fast transition for cases $x = 1.5$ and $x = 2.5$.

In summary, the linear fast transition in a closed loop test-rig can be simulated with high accuracy. For the most part the relative error of head deviation is smaller than 1% while the error for flow deviation is bound within a 4% range. Such values appear to be fairly acceptable for transient operation changes.

4.2. Results of fast transition with synchronous generator

Despite of the pump power curve smoothing the fluctuations are higher in cases of fast transition with synchronous generator compared to the cases with linear FSFC fast transition. This is especially pronounced for case $x = 2.5$. All power curves are shown in Figure 9 on the left while the associated speed curves are displayed on the right.

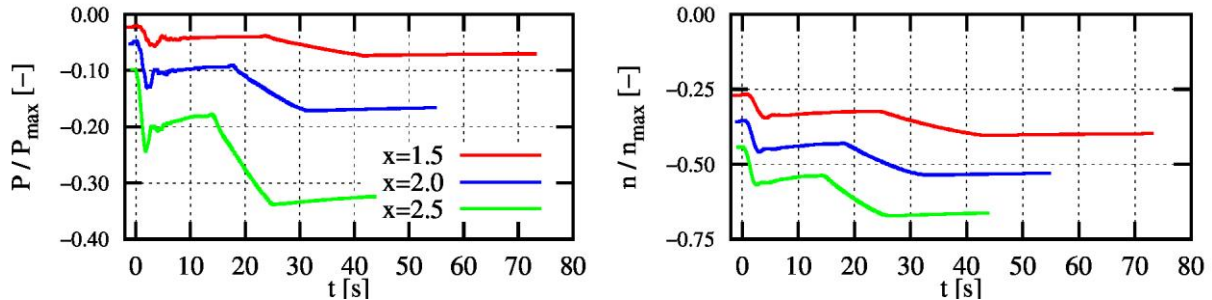


Figure 9 Normalized power and speed of the pump for fast transition with synchronous generator for different x -values.

Figure 10 shows the characteristics of head, n_{11} and the relative deviation at the model turbine. The pronounced pressure fluctuations also lead to higher deviations of head up to -6% (case $x = 2.5$). Nevertheless, the error of n_{11} exceeds a value up to $+/- 2\%$ in a few regions only. In the case of $x = 1.5$ the error remains predominantly smaller than $+/- 1\%$.

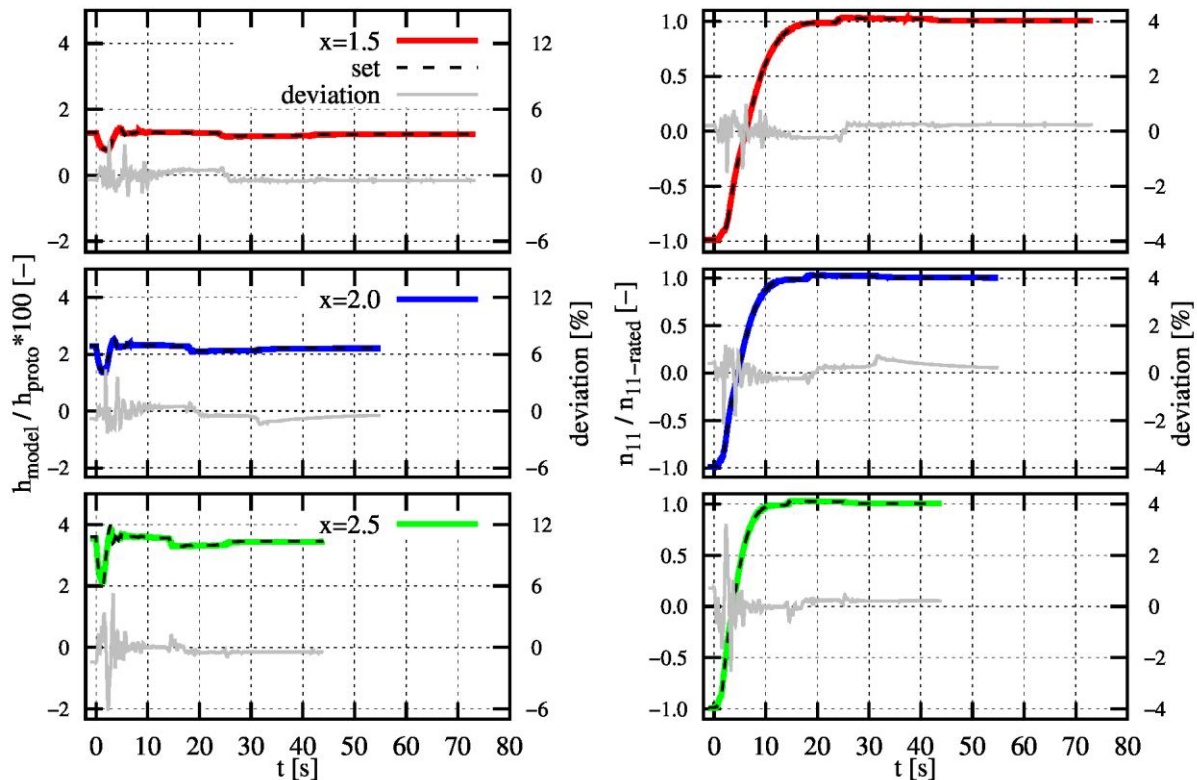


Figure 10 Normalized head and n_{11} of the model turbine for fast transition with synchronous generator for different x -values.

The difference becomes clearer by comparing Q_{11} values (Figure 11). Since the discharge change is significantly larger compared to the FSFC case at the beginning of the transition, deviations amount to $+/- 3\%$ for $x = 1.5$ and up to $+/- 9\%$ for $x = 2.5$.

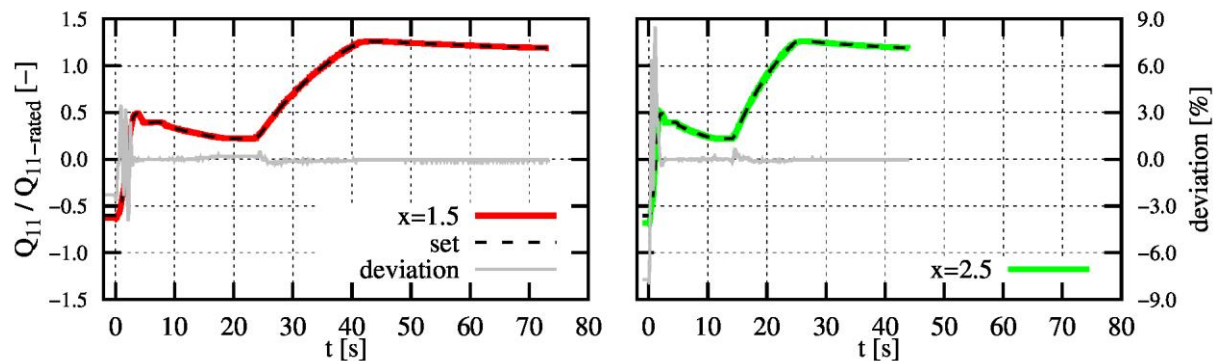


Figure 11 Normalized Q_{11} of the model turbine for fast transition with synchronous generator for different x -values.

The simulation results of the fast transition with synchronous generator show that even these transient processes are reproducible in a closed loop test-rig with reasonable good accuracy, too. Due to the higher pressure and discharge changes, there are also higher deviations increasing with a rising time scale factor x .

5. Conclusion and outlook

In order to carry out transient model test experiments in a closed loop test-rig, the relevant conditions on the model turbine must be known for the whole process. In addition, the service pump must be controlled in such way to achieve these conditions.

With the help of similarity laws, simulation results of fast transitions of an existing pumped storage power plant with reversible pump turbine have been transferred to the model turbine size. A special role played the Strouhal number, which determines the speed of the transient process and also influences the model turbine head. A fast transition with a linear speed change and a classic fast transition with load rejection and closing of the guide vanes were investigated. For both cases, three different process speeds were assumed. An optimizer adapted pump power to achieve the defined set point head at the model.

By comparing the results of the linear FSFC fast transition with those of the classic fast transition, much higher requirements on the behavior of the service pump for the classic transition become evident. Speed and head changes seem to be more moderate in the case of a FSFC transition. The time scale given by the process factor x has a strong influence. As x increases, the pressure gradients increase. In regions with higher pressure gradients, the highest deviations of head, n_{11} and Q_{11} occur. Nevertheless, it can be said that all the investigated cases could be reproduced in a closed loop test-rig with good accuracy.

In the next step, transient processes should be investigated in the real closed loop test-rig of the Institute applying the findings of this work.

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