Test facility for transient operation point changes of hydraulic machinery

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Abstract. With the increase in fluctuating power generation, hydropower plants have more and more taken on the task of regulating the power frequency control for the electrical grid. Therefore turbines in storage hydro power plants often operate outside their optimum and frequently change their operating points. The question in how far hydro power technology is in a position to allocate large amounts of balancing power very quickly requires research related to the dynamic behaviour of the hydraulic machines. The stability and safety of the operation must be guaranteed for a wide operating range and transient operation. Here, vibrations and oscillations as well as high stresses in the material must be avoided so that no fatigue fractures or high wear occur. A flexible test rig is being constructed in the institute's laboratory for research work related to these transient operating point changes. In this paper, the structure of the test facility and the various operating modes of the test facility are presented. Furthermore, the transfer of the plant conditions of the prototype to the model machine and the test facility is described. The requirements of the planned experiments on the test equipment are presented and special features of the test equipment for executing tests are derived.

1. Introduction
With the deregulation of the electricity market and the expansion of wind power and photovoltaics, the operation of hydropower plants has changed significantly, especially for pumped storage power plants. As a result, this type of power plant is increasingly used in secondary control and more recently also in primary control for the electrical grid. In addition, start- and stop processes are frequently performed on the machines [1, 2, 3, 4]. This inevitably leads to operation in very wide operating ranges, including pronounced off-design conditions. In this context vibrations and oscillations as well as high stresses in the material must be avoided so that no fatigue fractures or high wear occur.

For example, fatigue fractures have occurred in Francis turbines that previously operated for decades without problems or signs of fatigue failure. One reason for this seems to be that, as mentioned above, the electricity industry today operates its power plants in a market-oriented manner but under different conditions than before. This has resulted in an increase in the number of turbine starts and stops as well as an increase in the number and duration of overload and partial load periods [5].

In the future, the electrical grid will show a further increase in fluctuating power generation, such as wind power and photovoltaics. It is therefore necessary to expand the capacity of hydropower technology to provide balancing power with large capacities for stabilizing the electrical grid. The provision of large power generation ranges for a hydraulic machine inevitably leads to a large variation in discharge at a given head of the machine. As a result, the output can vary between overload and zero...
output in the limit case. In the hydraulic machine, strongly detached three-dimensional flow phenomena, possibly with cavitation, can occur leading to strong pressure fluctuations in the machine. These pressure fluctuations result in transient stresses within components of the mechanical structure. These transient stresses are superimposed on the mean stress and thus influence the operational stability of the hydraulic machine. In turn, that loading is often accompanied by a shortening of the service life.

Against this background, a new test facility is currently being erected in the laboratory at the Institute of Fluid Mechanics and Hydraulic Machinery (IHS) with the aim of investigating transient operating changes of hydraulic machines more precisely and to better understand the interaction with the piping system. Initial research work on this topic has already been carried out [6, 7, 8]. An operational transition was investigated for a pump turbine using numerical flow simulation. Starting from a pump operating point, the rotational speed of the main machine was changed to a turbine operating point. In addition, 3D CFD simulation (Computational Fluid Dynamics) was coupled with a 1D simulation method (water hammer analysis tool) to consider the interaction with the piping system. Hereby the comparison between the piping system of the real large power plant and the piping system of a closed-loop-test-rig for model tests was made. In a further research project, it was finally investigated how a closed-loop-test-rig must be controlled in order to achieve transient operation of the hydraulic machine in model size, corresponding to the properties at the real power plant under consideration of similarity rules [9, 10]. These papers show limitations in our existing closed loop test rigs and thus, the need for a new flexible test facility for the investigation of transient operational changes of hydraulic machinery.

2. Purpose of the test facility for transient operation point changes and procedure for their implementation

The test facility is designed for various and almost arbitrary transient investigations for hydraulic machinery. In a main research field, the flow behavior of transient operating point changes of hydraulic machines with effects on the mechanical structure is in focus. For this purpose, the operating conditions of the prototype machine have to be mapped onto the model machine in the test facility. In a first step, the operating point changes in the hydroelectric power plant are simulated and the resulting operating data at the hydraulic machine are determined. Such results are converted to the model machine using similitude. In order to generate a start solution for an optimization, a simulation of the operating conditions of the test facility is carried out for the operating point changes on the model machine. The simulation of the operating conditions of the test facility is adapted based on this starting solution. The optimizer adjusts the operating points of the components of the test facility during each iteration in case of discrepancies. This is done by comparing the actual and set values of the head at the model machine at each time step throughout the complete simulation. The correction at the operating points of the components of the test facility is adjusted by the optimizer, which is influenced by the deviation. To avoid unrealistic jumps, the changes are limited by a fixed maximum value. The transient process simulation is repeated using the new values. If the deviation in each time step falls below a defined limit value, the iteration loop stops and the optimization is finished. By this procedure, conditions are set on the model machine which are similar to the conditions on the turbomachine in the real power plant under consideration. For this purpose, the optimizer determines the target values of the controllers of the test facility. The test facility is operated with those target values. Detailed information about the optimization and examples for transient operating point changes are presented by Junginger et al [9, 10].

Other activities may focus on unsteady flow in all kinds of off-design operating conditions including cavitation phenomena. This also includes measures to reduce pressure fluctuations that occur in certain operation ranges. The aim is also to gain a better understanding of the basic flow processes and to generate high-quality reference measurements for the validation of simulation programs in order to further reduce their uncertainties.

3. Necessary operating conditions for transient operation point changes

For the investigation it is necessary to have geometrical similarity between model and prototype. Additionally, hydraulic similitude according to IEC 60193 test standard is required.
Prototype and model machine are operated under hydraulically similar operating conditions if the ratios of corresponding flow velocity components at every comparable position of prototype and model machine are equal. As a result, both machines have, at corresponding operating points, identical speed factors (1) and discharge factors (2) being defined as:

\[ n_{ED} = \frac{n \cdot D}{\sqrt{E}} \]  
\[ Q_{ED} = \frac{Q \cdot \sqrt{E}}{D^2} \]  

For other dimensionless terms relating to oscillating quantities like torque and force factor as well as factor of pressure fluctuation, reference is made to the IEC 60193 standard.

The Reynolds number of the model, related to the reference diameter of the machine, is smaller than that of the prototype, being unavoidable. The friction losses mainly depend on the Reynolds number, the ratio of friction losses to the total losses for the model becomes larger than the corresponding ratio for the prototype. Therefore, the efficiency of the model is slightly lower than that of the prototype. As a consequence, the model efficiencies and power factors need to be corrected if they are related to a different Reynolds number, e.g., scaling of the model results to the conditions of the prototype. Consequently, for the machines, the scale effect of the losses has to be considered for power and torque factor as well as efficiency.

For transient operating conditions, the dynamic behavior has to additionally reflect the Strouhal similitude, equation (3). For the characteristic length the diameter of the runner, as characteristic velocity the circumferential velocity at the nominal diameter of the runner and as characteristic time the time period of the transient process is used.

\[ Sr = \frac{D}{\omega \cdot \Delta t} \]  

The circumferential velocity can be eliminated by angular velocity and diameter, thus:

\[ Sr = \frac{D}{\omega \cdot \frac{2}{\Delta t}} = \frac{2}{\omega \cdot \Delta t} \]  

For the similitude the Strouhal number of model and prototype has to be the same, therefore:

\[ \frac{\omega_{model} \cdot \Delta t_{model}}{\omega_{prototype} \cdot \Delta t_{prototype}} = 2 \]  

This can be formulated as a time factor between the time period of the model size and the time period of the prototype size which is also equal to the relations in time of the corresponding operation points:

\[ \frac{\Delta t_{model}}{\Delta t_{prototype}} = \frac{t_{model}}{t_{prototype}} = \frac{n_{prototype}}{n_{model}} \]  

For the similitude of transient operating conditions, the time factor has to be selected by the factor of the runner speed between prototype and model. Typically, model runner speed is higher than the speed of the prototype. Thus, the resulting characteristic time scale on the model is smaller than that of the prototype. With a fixed runner speed of the prototype the time factor can be adjusted by the selection of the model runner speed.

For realizing a good measurement accuracy at the test rig, larger rotational speeds, e.g. around 1000 rpm, should be used. As a consequence, the characteristic time period of the model is shorter than the characteristic time period at the prototype. Thus, all processes in the test rig run faster, equation (6). Changes of flow rate cause pressure waves (water hammer phenomena) within the water piping system due to the weak compressibility of water. Cautious execution of transient tests is advisable to not exceed the mechanical limits of the test rig components. As an example, Figure 1 presents the speed curve for load rejection of a turbine with three different time factors. It may be clearly noted that a smaller time factor is associated with a higher runner speed and with a reduction of the time scale for the model machine.
4. Test facility

The test facility is designed as a closed loop test rig to also allow a definition of the absolute pressure level for cavitation assessment. The structure of the test facility and the positions of the components are presented as CAD-model, Figure 2. The schematic, Figure 3, illustrates the arrangement of the main water conduit. The test rig tower is centrally located and the test machine is attached to it. The tower consists of four concrete columns, whereby two of them are always connected at the top in a longitudinal direction. The variable speed motor-generator for connecting the test machine, maximum power of about 700 kW, is located on top of the test rig tower. Of course, both directions of rotation can be selected to enable four quadrant operation of the test machine.

![Figure 2](image)

**Figure 2.** Scheme of the test facility.

The test machines are hydraulically connected to the headwater and tailwater vessels. The headwater vessel is manufactured with a diameter of 1400 mm for a maximum pressure level of 25 bar. It is equipped with an air chamber being connected to the compressed air supply of the laboratory. Furthermore, it can be electrically adapted to the geometric conditions of the test machine by means of an adjustment device in the
height position and in the horizontal transverse direction to the connecting pipes. On the suction side, the test machine is connected to the tailwater vessel, designed with a diameter of 2000 mm for a maximum pressure level of 10 bar. The tailwater tank can be manually adjusted horizontally in the direction of the pipes to meet the requirements of the test machine. It is also equipped with an air chamber connected to the compressed air supply of the laboratory. Additionally, the air chamber is connected to a vacuum pump allowing the pressure level of the test circuit to be adjusted above and below ambient pressure.

![Figure 3. Schematic of the main parts of the test facility.](image)

Other main components of the test facility include two service pumps which can operate individually or together in serial or parallel connection. With the two service pumps, a maximum flow rate of more than 1 m³/s can be achieved in parallel connection and a maximum head of more than 140 m in serial connection. Each service pump is coupled with a variable speed motor with a power of approximately 450 kW. The energy recovery turbine with electrically adjustable guide vanes reduces the pressure in the pipes and generates up to approximately 300 kW of power with a generator. For pressure reduction and additional flow control through the bypass section, two annular piston valves are installed, one annular piston valve in series connection and one in parallel connection to the energy recovery turbine. Most of the main piping has a diameter of 500 mm. The pipes in the measuring sections of the two flow meters are designed with a diameter of 400 mm to achieve a higher accuracy of the flow measurement.

In order to guarantee a constant water temperature, a heat exchanger for cooling the water is installed and connected to the cooling network of the university. The heated water is removed from the tailwater vessel and pumped through the heat exchanger by a pump and fed back into the main pipe. To clean the water, a filter circuit is connected to the tailwater vessel. The polluted service water is removed from the tailwater vessel, pumped to a filter and fed back into the main circuit. In order to guarantee a flexible mode of operation, various motor driven valves are installed which allow switching between the different modes of operation. For the calibration of the flow meters open service tanks and a calibration device is installed in the test facility.

5. Operating modes for the test facility

The arrangement of the test facility enables test operation of turbines, pump-turbines and pumps under steady state and transient operating conditions.
To generate transient boundary conditions on the test machine, the following adjustment options are available. By changing the runner speed of the service pumps, the operating point of the pumps and thus the discharge and the head is modified. The energy recovery turbine is adjusted by varying the runner speed and by adjusting the guide vanes. As a result, the operating point of the energy recovery turbine can also be varied in discharge and head. In addition, it is possible to adjust discharge and pressure difference with the opening of the two annular piston valves.

In order to explain the functionality of the test facility, three modes of operation are described next.

5.1. **Turbine mode operation**

This operation mode is intended to investigate unsteady phenomena with constant operating point in turbine operation. Instabilities in low partial load or high partial load such as the draft tube vortex rope or overload instabilities can be investigated. In addition, of course, operating point transitions with minor changes in the boundary conditions can also be carried out. During turbine operation, the speed regulated service pumps push water through the flow measurement section to the headwater vessel, then through the turbine into the tailwater vessel and back to the pumps, Figures 4 and 5.

![Figure 4. Test facility operation in turbine mode of the test machine.](image)

![Figure 5. Schematic of the test facility operation in turbine mode of the test machine.](image)

5.2. **Pump mode operation**

That operation mode allows the investigation of unsteady phenomena with a constant operating point during pump operation. Phenomena such as rotating stall can be investigated. In addition, this mode is
used to perform operating point transitions with small changes in the boundary conditions. The test unit moves water to the headwater vessel towards the energy recovery turbine and/or through the annular piston valve back to the tailwater vessel, Figures 6 and 7. The service pumps are not in use for that operation mode.

Figure 6. Test facility operation in pump mode of the test machine.

Figure 7. Schematic of the test facility operation in pump mode of the test machine.

5.3. Transient operation

The most flexible operation mode deals with almost arbitrary transient operation conditions of a hydraulic machine used as test machine. For example, transitions from pump to turbine operation and vice versa, start up and shut down processes as well as fast changes in turbine operation or pump operation are feasible.

The aim is to gain a deeper understanding of the flow within the test machine and the resulting loads on machine components, for example during a transition from deep partial load to overload operation or vice versa. Other test options include flow measurements for validation of transient CFD simulations as well as measurements of strains for validation of stress prediction methods. This mode is the only one where transitions from pump to turbine operation and vice versa are possible.

As the control of the test rig allows a simultaneous and independent speed control of all existing motors, as well as the opening of the guide vanes on the test machine and on the energy recovery turbine and on the annular piston valves, it is also possible to replicate, e.g., start and stop processes. The additional bypass at the energy recovery turbine will vary the flow rate more rapidly. Of course, it will be a challenge to define all control tables and to predict the interaction and thus the behavior of the test rig. The entire piping system is equipped with pressure sensors for gathering experience during operation of general condition changes. In this way, the maximum occurring pressures can also be measured. This also permits the measurement of wave velocity and the possible interaction of pressure fluctuations of a test machine with the main piping system.
The annular piston valve, which is mounted in a bypass to the energy recovery turbine, allows a larger discharge change compared with the energy recovery turbine alone. The annular piston valve, which is installed downstream of the energy recovery turbine, allows a larger pressure difference to be achieved compared to the arrangement without the annular piston valve.

Some pipes may have different flow directions for arbitrary operation point transitions, Figures 8 and 9. Pipes with constant flow direction are marked with yellow arrows. Pipes with both flow directions are marked with green arrowheads for pump operation and with blue arrowheads for turbine operation of the test machine.

**Figure 8.** Test facility operation with transition between pump and turbine mode of the test machine.

As an example, the rapid transition from pump operation to turbine operation on the test machine will be described here. In the start configuration, the test machine operates in pump mode at nominal speed, nominal discharge and nominal head. At these conditions the flow direction in the pipes is in the direction of the yellow arrows and in the section of the test machine in direction of the arrows with the green arrowheads. The test machine pumps water from the tailwater vessel to the headwater vessel. From the headwater vessel, water flows to the pipe junction marked with number 1. There, the water
combines with the water coming from the pipes of the service pumps. At position ①, the water flows to the energy recovery turbine and the annular piston valve in the bypass of the energy recovery turbine. Then, the water flows to the pipe junction marked number ②, Figures 8 and 9. There, the water is divided between the two open connections. One part of the flow moves to the tailwater vessel, while the other part flows to the service pumps. The service pumps generate the necessary head at the test machine to achieve the desired operation point. Since the test machine is in pumping mode at this time, the pumps deliver a relatively low discharge during this state.

For the operating point transition, the speed of the test machine is changed according to the specification of the prototype behavior considering similitude relations. At the same time the flow through the test machine is reduced according to the prototype conditions. In order to correctly set the head according to the prototype conditions, the speed of the pumps is increased accordingly and the energy recovery turbine and the annular piston valves are varied to match. From the point where the flow direction is reversed on the test machine, the flow direction changes in the area between pipe junction number ① and number ②, Figures 8 and 9. Now the water in the pipes flows in the direction of the blue arrowheads. From this point on, the water delivered by the service pumps is divided at the pipe junction marked with number ①. Part of the water flows to the energy recovery turbine and the annular piston valve in the bypass of the energy recovery turbine. The other part flows to the headwater vessel towards the test machine. Downstream of the test machine, the water flows through the tailwater vessel to number ② junction, where the water from both pipes merges to flow towards the service pumps. The runner speed of the test machine, the runner speed of the pumps, as well as the energy recovery turbine and the annular piston valves are adjusted according to the conditions on the prototype test machine until the end of the operation transition.

During the transient operation point changes, the operating data and other measured quantities can be recorded. Among other things, it is planned to measure unsteady forces and torques on the runner being measured under water with a special measuring shaft for dynamic measurement even before the seal. In addition, pressure fluctuations in the test machine are, of course, also recorded during operational transitions.

The current progress of the test rig assembly presents the test rig tower as well as the adjustment devices of the two vessels, the platform for the assembly of the test machine and the associated motor-generator, Figure 10.

**Figure 10.** Recent ground floor photography of the test facility, work in progress.
6. Conclusion
A test facility for very flexible and almost arbitrary transient operation point changes of hydraulic machinery is presented. The design of the test facility is especially designed for fast changes of operation condition at the test machine in order to realize the smallest possible time factors. The test facility allows a simultaneous and independent speed control of all existing motors, as well as the opening of the guide vanes on the test machine and on the energy recovery turbine and on the annular piston valves. The new test facility may generate conditions at the model test machine replicating prototype machine behavior under similarity conditions. The entire main piping system is equipped with pressure sensors for measurement of wave velocity and the possible interaction of pressure fluctuations of a test specimen with the piping system. The test facility is currently under construction and is expected to be completed this year.

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Nomenclature

| Symbol | Description |
|--------|-------------|
| D      | Diameter or the runner |
| E      | specific hydraulic energy |
| N      | runner speed |
| n      | runner speed |
| nED    | speed factor |
| Q      | discharge |
| QED    | discharge factor |
| Sr     | Strouhal number |
| t      | time |
| u      | circumferential velocity |
| ω      | angular velocity |

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