Measurements of Fast Transitions at a Reversible Pump Turbine Model in Closed Loop Test Rig

J Junginger O Kirschner and S Riedelbauch
Institute of Fluid Mechanics and Hydraulic Machinery, University of Stuttgart, Pfaffenwaldring 10, 70569, Germany
E-mail: johannes.junginger@ihs.uni-stuttgart.de

July 2021

Abstract. It is common practice to experimentally investigate the performance characteristics of hydraulic fluid machinery in power plants with steady-state operating points by scaled homologous model machines in closed-loop test rigs. In this case, the piping of the main water conduit from the power plant and the test rig plays a minor role. However, for studies of operating transient, e.g. load rejection, the influence of the weak compressibility of water in the pipes is considered by 1-D water hammer analyses. Due to differences in the main water conduit between the power plant and the test rig, the test rig is specifically controlled by new procedures to replicate the relevant conditions at the fluid machine in the test rig under similarity conditions.

The results of this newly developed method for specific control of the test rig for operating transients are presented and successfully validated on a closed-loop experimental system. In future, suitable instrumented model machines then allow the measurement of dynamic pressures or even strains in and on machine components in order to elaborate, as a goal, the influence of different operating transients on the service life of machine components. Furthermore, these experimental data can also be used to validate Computational Fluid Dynamics (CFD) and Finite Element Analyses (FEA) simulations.

Keywords: Hydraulic Transients, Variable Speed Machines, Transient Measurement Techniques, Closed Loop Test Rig

1. Introduction

Hydropower plants are increasingly employed for electrical grid stabilization purposes such as Frequency Containment Reserves (FCR) and Frequency Restoration Reserve (FRR) due to their fast response and ramping capabilities [1, 2]. Therefore, their hydraulic turbomachines are operated under extreme off-design conditions including frequent operation transients. The arising dynamic pressure loads result in a high number of load cycles challenging the structural integrity of machine components. Thus,
the high-level goal of the current research work is to assess the influence of operation transients on the life time of hydraulic machinery components.

The way of carrying out load changes with hydroelectric power plants largely depends on the generator type used. Synchronous generators, mostly present in existing hydro power plants today, are tied to a fixed rotational speed independent of the operation range once synchronized with the electrical grid. Turbine load changes are produced by mass flow changes through the hydraulic machine, in turn changing the shaft torque and thus, power. However, mass flow changes are not possible for pump operation using synchronous generators directly consuming a certain amount of power depending on the pump characteristic. Coupling the synchronous generator with a full-size frequency converter (FSFC) significantly increases operation flexibility as the rotational speed is not in a fixed relation to the electrical grid frequency any more [3,4]. That is particular advantageous for pump-turbines in pump storage plants as the entire rotor (hydraulic and electrical machine) can achieve almost any rotational speed in the nominal speed range for both rotation directions [5,6].

That concept, synchronous generator combined with FSFC, allows the acceleration of the entire rotor (generator and turbine power unit) with almost rated torque from standstill to operational speed, requiring no time delay for synchronization [6]. That property allows a highly flexible power plant operation and, especially, for plants with pump-turbine equipment, an operation transient from pump operation to turbine operation and vice versa is feasible by a change of the rotational speed. Such operation maneuver was never realized in a full-size power plant yet. However, very few available recent research studies focused on that operation transient applying a linear speed change. Measurements on a scaled model turbine [7] and 1-D water hammer simulations of the power plant confirmed the feasibility of a fast transition with linear rotational speed change [8,9]. Applying measurement results as predefined boundary conditions at the spiral case inlet and the draft tube outlet, unsteady flow field simulations of the complete pump turbine present many flow vortices with high pressure fluctuations within the pump turbine during that transient [10–13]. Real time coupling of the 1-D water hammer simulation of the complete piping system with the 3-D unsteady CFD simulation of the pump turbine produces relevant boundary conditions for the CFD part as result of the entire simulation [14,15]. Other applications use that method to analyse pump systems [16]. Applying such procedure for a closed-loop model test rig and a real size power plant with geometrical similar pump turbines reveals differences in head, flow rate and pressure pulsation characteristics [15,17]. That clearly indicates a substantial influence of the pipe system on the flow properties at the pump turbine during a transient. Such different loads on the structure are then to be analyzed with regard to an impact on service life.

Consequently, the test rig control must be governed to mimic the time-dependent head and flow conditions of the scaled model machine as defined by the real power plant satisfying hydraulic similarity conditions. A new method to predict and define the test rig control using 1-D water hammer simulations of the power plant and the test rig was
recently proposed [18,19]. The current paper validates the proposed method and verifies the feasibility with experiments carried out at our small closed loop test rig. Details of the results are presented subsequently.

2. Prototype test case and transfer to model scale

2.1. Plant dynamic test case

The prototype 1-D water hammer simulation is based on a real power plant in the Austrian Alps, which is equipped with one reversible pump turbine. The nominal power plant output in turbine mode is approximately 280MW. The 1-D water hammer simulation model includes all hydraulic components existing at the power plant. It is also assumed that this power plant has a fictitious FSFC which is also implemented in the model. A fast transition from pump to turbine mode and vice versa via rotational speed change at constant guide vane opening is simulated. The rotational speed change based on a spline function is specified as the only controlling variable for the transient process.

The duration of the transition process on the prototype $t_{\text{proto}}$ is 30 seconds. Between the two transitions, the power plant is in constant turbine operation for another 30 seconds. The defined change of rotational speed $n$ results in time-dependent functions of discharge $Q$, shaft torque $T$ and head $H$ at the machine, Figure 1. Those parameters are strongly influenced by the main water passage connecting the machine with head water and tail water reservoir.

2.2. Transfer from prototype to model size

In order to replicate the prototype process exactly, the similarity laws for turbomachines, such as geometric similarity, kinematic similarity and dynamic similarity must be observed.

The geometric similarity is realized by using a homologous model of the prototype with a scale factor of approximately 1 : 13.34. Kinematic similarity is achieved by maintaining the velocity triangle from the prototype runner to the model machine runner throughout the transient process [20]. This is ensured by reaching the normalized values for rotational speed factor $n_{ed}(t)$ and discharge factor $Q_{ed}(t)$ at each time step (according to the IEC standards [21]). The torque factor $T_{ed}(t)$ is used to observe the mechanical performance level.

Due to the huge differences in size, it is practically impossible to achieve the same Reynolds number and thus the dynamic similarity in the model test. The typical achievable Reynolds number for the model machine is in the range $5 \cdot 10^5$ to $5 \cdot 10^6$ and for the prototype the range is between $10^7$ to $10^9$. The lower Reynolds number at model test conditions result in a slightly lower efficiency due to thicker boundary layers of the model machine relative to the prototype at the plant. Nevertheless, an attempt is made to reduce these differences by using a higher rotational speed for scaled model
testing. But, in combination with the kinematic similarity, this leads to an adjustment of the time scale for the model test. In this paper, the relationship between initial speed and time scale is established via the Strouhal number, equation (1). The Strouhal number is calculated with the reference diameter $D$ (outlet diameter in turbine mode) and the initial value of circumferential speed $u$ at the reference diameter of the runner. The time span $t$ corresponds to the duration of the complete transient process. By transforming the equation, a ratio of the rotational speeds of the prototype and model machines is obtained which corresponds to the two time spans of the transient processes of both machines. Therefore, if $x_t$ results from the initial values of the chosen model speed versus the prototype speed, the duration of the transient model process decreases in the same ratio. This factor is calculated with the initial speeds at the beginning of the transient process and remains constant to the end, see also [15,17–19].

With equations (2) through (4), it is now possible to convert the results from the prototype simulation to the model size. These converted data correspond to the set values to be achieved in the simulation of the test rig and afterwards in the model test.
\[ Sr = \frac{D}{u \cdot t} = \frac{D}{\omega \cdot \frac{D}{2} \cdot t} = \frac{2}{\omega \cdot t} \Rightarrow \frac{n_{\text{proto}}}{n_{\text{model}}} = \frac{t_{\text{model}}}{t_{\text{proto}}} = x_t \] (1)

\[ n_{\text{ed-proto}}(t_{\text{proto}}) = n_{\text{ed-model}}(t_{\text{model}}) \] (2)

\[ Q_{\text{ed-proto}}(t_{\text{proto}}) = Q_{\text{ed-model}}(t_{\text{model}}) \] (3)

\[ T_{\text{ed-proto}}(t_{\text{proto}}) = T_{\text{ed-model}}(t_{\text{model}}) \] (4)

Figure 2 shows the first 30 seconds of the previously described fast transition from pump to turbine mode (Fig. 1) in the \( n_{\text{ed}}-Q_{\text{ed}} \) machine characteristic at prototype time scale. This behavior must now be reproduced in the model test with the adapted model time factor.

2.3. Test rig simulation and optimization structure

Like the power plant, the test rig of IHS is modeled using the 1-D water hammer simulation method with all important components (Fig. 3). This test rig consists of...
three different sections. One water passage contains the model machine, another pipe branch the service pumps and the third water passage is used as bypass.

The bypass is necessary to enable the discharge reversal at the model machine without disturbing the service pumps in their direction of rotation. A partly opened valve in the bypass ensures a resistance for the pressure difference between the high-pressure and low-pressure sides.

In the simulation, the opening positions of all valves and the guide vanes, the initial pressure level as well as the rotational speed curve of the model machine are specified. The setpoints for head and the discharge versus time at the model machine, which were converted from the prototype are generated by the rotational speed and thus by the driving power of the service pumps. But the needed time depending behaviour for the control of the service pumps is unknown.

To generate the required rotational speed of the service pumps, an optimizer analyses the deviation of a completed water hammer simulation in each time step and adjusts the performance of the service pumps. Then, a new simulation is carried out.
After a few loops, the final result shows a rotational speed curve of the pumps versus time that reflects the required behaviour at the test specimen. This speed curve can now be prescribed in the test rig controller as setpoint curve for the model test. Figure 4 shows the evolution of an optimization process with up to 600 iteration loops. On the left part the development of the pump speed versus time is presented. The right side shows the setpoint curve begin transferred from the prototype data (red dashed) and the development of head (solid grey shaded) versus time for some iteration loops.

3. Measurement technique at closed loop test rig

In contrast to conventional model acceptance tests, where the sensors are mainly attached to the model machine, the sensors involved for transient tests are distributed throughout the entire test rig (Fig. 3). This is mainly to gain information about the behaviour of the entire test rig and not only at the model machine. In order to obtain quantitatively correct and meaningful results, the fluid mechanical energy losses of the simulation model and the real losses in the test rig need to be brought into consistency along the piping system. This is the only way to ensure that the results generated by the optimizer actually match the measurement.

To determine the flow losses and to monitor the propagation of pressure waves, eight absolute pressure sensors (p₁−p₈) are distributed in the closed loop test rig. In addition to the differential pressure measurement at the model machine (ΔH_{model}), the differential pressure of the pumps (ΔH_{pump1} and ΔH_{pump2}) and the differential pressure on the bypass valve (ΔH_{bypass}) are recorded.

The discharge measurements take place in the water passage of the service pumps (Q_{total}) and in the bypass (Q_{bypass}). The time constant of the electromagnetic sensor for the Q_{total} measurement is R_{ems} = 0.1s. The measurement in the bypass is carried out by measuring the pressure difference at an orifice. The calibration of the orifice measurement was done at steady state conditions and the calibration curve is
additionally checked at transient conditions. Both measuring systems are not ideal for rapid volume rate changes, but these are the only systems available in this test rig. The discharge at the model machine \((Q_{\text{model}})\) is calculated by a difference of total and bypass discharge (Eqn. 5).

\[
Q_{\text{model}}(t) = Q_{\text{total}}(t) - Q_{\text{bypass}}(t)
\]  

The rotational speed measurements at the pumps \(n_{\text{pumps}}\) and the model machine \(n_{\text{model}}\) verify whether the given speed courses are realized. For these, inductive proximity sensor send an impulse whenever one of the eight markers mounted at the machine shafts passes the sensors. The measurement software calculates the frequency and thus, the rotational speed. In order to gain information about the mechanical performance of the model machine the torque \(T_{\text{model}}\) is recorded by a torque transducer. A magnetic angle sensor measures the guide vane position \(\alpha_{\text{model}}\) to guarantee the correct opening and thus, the desired operating point. Finally, it should be mentioned that all sensors are calibrated for their intended use prior to installation. An overview of these sensors can be found in the Table 1. The uncertainty analysis of the sensors are carried out according to [22] and the results are normalized to the initial measured values of the test case. Since the measuring range of some of the sensors is only used to a small extent, the relative data sometimes lead to high values.

| Example | type | output signal | sampling rate | uncertainty |
|---------|------|---------------|---------------|-------------|
| \(p_1 - p_8\) | absolute pressure | analog | 2 kHz | 0.134–2.127% |
| \(\Delta H_{\text{model}}, \Delta H_{\text{pump1}}, \Delta H_{\text{pump2}}, \Delta H_{\text{bypass}}\) | differential pressure | analog | 10 Hz | 0.31–0.394% |
| \(Q_{\text{total}}\) | electromagnetic sensor | analog | 10 Hz | 1.459% |
| \(Q_{\text{bypass}}\) | differential pressure | analog | 10 Hz | 1.979% |
| \(n_{\text{pumps}}, n_{\text{model}}\) | inductive proximity sensor | digital | – | – |
| \(T_{\text{model}}\) | torque transducer | analog | 10 Hz | 3.243% |
| \(\alpha_{\text{model}}\) | magnetic angle sensor | analog | 10 Hz | 3.333% |

4. Measurement Results

The results for a time factor of \(x_t = 2/3\) are considered representative of approximately 100 different test sequences of a fast transition. In this example, the guide vanes are set to a constant opening.

First, the behaviour of the rotational speed and normalised machine parameters \((n_{\text{ed}} \text{ and } Q_{\text{ed}})\) versus model time respectively prototype time are considered (figure 5).
Since the courses of speeds are fixed in the simulations and in the model test, their excellent agreement is not surprising. The blue and green lines fall on each other. Below are the courses of the speed factor $n_{ed}$ and the discharge factor $Q_{ed}$ normalized to the rated values in turbine mode. Like the speed curve, the speed factor can also be reproduced very well. Thus, it can be concluded that the course of the measured head at the model machine agrees well with the setpoints, resulting from the test rig simulation. All three lines essentially fall on each other.
More conspicuous are the deviations of the discharge factor, especially in sections where the flow rate changes rapidly. However, if the guide vane opening, rotational speed and head of the model machine are correct, the corresponding discharge of the machine will result automatically. Since the discharge, as mentioned above, is determined from the difference between total discharge and discharge in the bypass (Figure 6), it can be assumed that the differences between simulation and measurement originate from the time constant of the sensors. This can be recognised by the fact that the measurement signal fits very well in the region of slow changes and lags behind the setpoint in the region of fast changes.

The behaviour of the service pumps can also be predicted with good accuracy by the test rig simulation (Fig. 7). As with some other parameters, the measurement and the simulation fit well together and the curves show only minor deviations from each other. Also the slight time offset is noticeable which - as in the discharge measurement - most likely results from the damping of the sensors. It can be assumed that these courses would have less deviation from each other by using a faster sensor.

In conclusion, it can be said that the transient behaviour of the model machine in the test rig can be well predicted by the simulation.

5. Conclusions

The validation of the new method to mimic operational transients of a real size pump turbine with scaled model pump turbine in a closed loop test rig was successful. The desired functions of head and flow rate at the pump turbine as a function of time for a pump to turbine and back operational transient are achieved with the control of the
service pumps. The method to predict this control relies on a number of 1-D water hammer simulations combined with optimization strategies. The developed approach is general enough for being also applied to other transient processes.

Head and flow factor identity between prototype and model result in similar flow pattern and behavior within the pump turbine along the time line. However, the time scale changes depending on the ratio of the rotational initial speeds of prototype and model. For typical conditions, model runner speed is higher than prototype runner speed, the transient process in the test rig runs faster requiring some caution during the experiments. That similarity then allows to transfer measured results with suitable instrumented model machines from test rig to prototype size to determine the influence of operation transients on service life of prototype machine components. And, moreover, these experimental data can also be used to validate CFD and FEA simulations.

At the test rig, thoughts need to circulate to improve the accuracy of flow measurements within the different pipe branches as that measurement produces the highest uncertainty.

Tests with higher time factors are certainly of interest in general, but also specifically to reduce differences of dynamic similarity, Reynolds number effect, between model and prototype processes.

6. Discussions

The aim of creating an accurate replication of the transient process of the prototype pump turbine in the model test was achieved. Deviations between the test rig simulation and the measurement are mainly found in the case of strong changes in discharge. These deviations can be attributed to the damping of the sensors.
Since the model turbine discharge is calculated from two different measurements the deviations are favoured. However, due to limited space in the pump turbine branch of the test rig, it is not possible to attach another flow sensor and determine the discharge directly. Further it is to be expected that the deviations will increase as the processes become faster.

The use of fast sensors or other measurement techniques could counteract this and should be considered in further investigations.

Nomenclature

| symbol | unit       | description                      |
|--------|------------|----------------------------------|
| α      | [°]        | guide vane angle                 |
| D      | [m]        | reference diameter according to IEC60193 [21] |
| H      | [m]        | head                             |
| n      | [min⁻¹]    | rotational speed                 |
| n_{ed} = n \cdot D / \sqrt g \cdot H | [-] | speed factor                     |
| p      | [m]        | pressure                         |
| Q      | [m³ s⁻¹]   | discharge                         |
| Q_{ed} = Q / D^2 \cdot \sqrt g \cdot H | [-] | discharge factor                 |
| ρ      | [kg m⁻³]   | density                          |
| R_{ems} | [s]      | time constant of the electromagnetic sensor |
| S\text{r} = D / u \cdot t | [-] | strouhal number                  |
| t      | [s]        | time                             |
| T      | [N m]      | torque                           |
| T_{ed} = T / \rho \cdot D^3 \cdot H | [-] | torque factor                    |
| u      | [m s⁻¹]    | circumferential speed            |
| ω      | [s⁻¹]      | angular speed                    |
| x_t    | [-]        | time factor                      |

References

[1] Brauner G 2008 New role of pumped storage hydro plants for balancing of wind and renewable micro systems 15th Int. Sem. on Hydropower Plants.
[2] Hell J, Egretzberger M, and Lechner A 2014 Grid frequency response contribution of hydro power for grid stabilization Proc. of Hydro2014.
[3] Nicolet C, Braun O, Ruchonnet N, Beguin A, and Avellan F 2016 Full size frequency converter for fast francis pump-turbine operating mode transition Proc. of HydroVision Inter. Conf. 2016.
[4] Schafer D, and Simond J.-J 1998 Adjustable speed asynchronous machine in hydro power plants and its advantages for the electric grid stability. CIGRÉ Report.
[5] Hell J, Lechner A, Schiërhuber R, and Vaillant Y 2012 Full size converter solutions for pumped storage plants: a promising new technology Proc. of Hydro2012.
[6] Hildinger T, and Köidding L 2013 Modern design for variable speed motor-generators - asynchronous (DFIM) and synchronous (SMFI) electric machinery - options for pumped storage power plants Proc. of Hydro2013.
[7] Ruchonnet N, and Braun O 2015 Reduced scale model test of pump-turbine transition *Proc. of 6th IAHR Inter. Meeting of the Working Group on Cavitation and Dynamic Problems in Hydraulic Machinery and Systems*.

[8] Nicolet C, Braun O, Ruchonnet N, Hell J, Bégnin A, and Avellan F 2017 Simulation of pump-turbine prototype fast mode transition for grid stability support *J. Phys.: Conf. Ser.*, **813** p. 012040 URL https://doi.org/10.1088/1742-6596/813/1/012040.

[9] Ruchonnet N, and Braun O 2017 1d simulation of pump-turbine transition *J. Phys.: Conf. Ser.*, **813** p. 012039.

[10] Stens C, and Riedelbauch S 2015 CFD simulation of the flow through a pump turbine during a fast transition from pump to generating mode *Proc. of 6th IAHR Inter. Meeting of the Working Group on Cavitation and Dynamic Problems in Hydraulic Machinery and Systems* pp. 323-330.

[11] Stens C, and Riedelbauch S 2016 Investigation of a fast transition from pump mode to generating mode in a model scale reversible pump turbine *IOP Conf. Ser.: Earth and Environmental Science*, **49** p. 112001 URL https://doi.org/10.1088/1755-1315/49/11/112001.

[12] Stens C, and Riedelbauch S 2016 CFD analysis of fast transition from pump mode to generating mode in a reversible pump turbine *High Performance Computing in Science and Engineering '16: Transactions of the High Performance Computing Center, Stuttgart (HLRS)* 2016 pp. 487-498.

[13] Stens C, and Riedelbauch S 2017 Influence of guide vane opening on the flow phenomena in a pump turbine during a fast transition from pump mode to generating mode *J. Phys.: Conf. Ser.*, **813**(1) p. 012024 URL https://doi.org/10.1088/1742-6596/813/1/012024.

[14] Wu D, Yang S, Wu P, and Wang L 2015 MOC-CFD coupled approach for the analysis of the fluid dynamic interaction between water hammer and pump *J. of Hydr. Engineering*, **141**(6) p. 06015003.

[15] Stens C 2018 Investigation of a fast transition from pump mode to generating mode in a reversible pump turbine PhD thesis University of Stuttgart.

[16] Yang S, Wu D, Wu P, and Wang L 2016 Investigation on the transient characteristics of the pump system using MOC-CFD coupled method *16th Inter. Sym. on Transport Phenomena and Dynamics of Rotating Machinery*.

[17] Riedelbauch S, and Stens C 2019 Pump to turbine transient for a pump-turbine in a model test circuit and a real size power plant *IOP Conf. Ser.: Earth and Environmental Science*, **240** p. 072039 URL https://doi.org/10.1088/1755-1315/240/7/072039.

[18] Junginger J, Riedelbauch S, and Kirschner O 2019 Transfer of transient conditions from prototype to closed-loop model test rig *IOP Conf. Ser.: Earth and Environmental Science*, **240** p. 082010 URL https://doi.org/10.1088/1755-1315/240/8/082010.

[19] Junginger J, Kirschner O, and Riedelbauch S 2020 Method to transfer transient prototype conditions to closed loop model test rig *Inter. J. of Fluid Machinery and Systems*, **13**(1) pp. 1-11 URL https://doi.org/10.5293/1JFMS.2020.13.1.001.

[20] Dick E 2015 *Fundamentals of Turbomachines* Vol. 1. Springer Netherlands.

[21] International Electrotechnical Commission 1999 INTERNATIONAL STANDARD IEC60193 - Hydraulic turbines, storage pumps and pump turbines - Model acceptance tests Tech. Rep. 60193:1999 3, rue de Varembe Geneva, Switzerland.

[22] Abernethy R. B, Benedict R. P, and Dowdell R. B 1895 ASME measurement uncertainty *ASME Journal of Fluids Engineering*, **107**(2) pp. 161-164 URL https://doi.org/10.1115/1.3242450.