Comprehensive experimental and numerical analysis of instability phenomena in pump turbines

Ch Gentner[1], M Sallaberger[1], Ch Widmer[1], B-J Bobach[3], H Jaberg[2], J Schiffer[2], F Senn[2], M Guggenberger[2]

[1] ANDRITZ HYDRO LTD, P.O. Box 2602, 8021 Zurich, Switzerland
[2] Graz University of Technology, Austria; [3] HTWG Konstanz, Germany

E-mail: christof.gentner@andritz.com

Abstract. The changes in the electricity market have led to changed requirements for the operation of pump turbines. Utilities need to change fast and frequently between pumping and generating modes and increasingly want to operate at off-design conditions for extended periods. Operation of the units in instable areas of the machine characteristic is not acceptable and may lead to self-excited vibration of the hydraulic system. In turbine operation of pump turbines unstable behaviour can occur at low load off-design operation close to runaway conditions (S-shape of the turbine characteristic). This type of instability may impede the synchronization of the machine in turbine mode and thus increase start-up and switch over times. A pronounced S-shaped instability can also lead to significant drop of discharge in the event of load rejection. Low pressure on the suction side and in the tail-race tunnel could cause dangerous separation of the water column. Understanding the flow features that lead to the instable behaviour of pump turbines is a prerequisite to the design of machines that can fulfil the growing requirements relating to operational flexibility. Flow simulation in these instability zones is demanding due to the complex and highly unsteady flow patterns. Only unsteady simulation methods are able to reproduce the governing physical effects in these operating regions. ANDRITZ HYDRO has been investigating the stability behaviour of pump turbines in turbine operation in cooperation with several universities using simulation and measurements. In order to validate the results of flow simulation of unstable operating points, the Graz University of Technology (Austria) performed detailed experimental investigations. Within the scope of a long term research project, the operating characteristics of several pump turbine runners have been measured and flow patterns in the pump turbine at speed no load and runaway have been examined by 2D Laser particle image velocimetry (PIV). For several wicket gate positions, the flow fields in the vane-less space at runner inlet observed in the experiment are compared with the results of unsteady CFD flow simulations. Physical phenomena are visualized and insight to flow phenomena is given. Analyses using both results of simulation and measurement allow deriving a consistent explanation of the fluid mechanical mechanisms leading to the S-shaped instability of pump turbines.

IOP Conf. Series: Earth and Environmental Science **22** (2014) 032046 doi:10.1088/1755-1315/22/3/032046

1. S-shaped instability in turbine operation

The existence of instabilities in a hydraulic system including pump turbine and penstock is related to the slope of the head characteristic of the hydraulic machine. The critical slope is a necessary condition for the instability of the system but not sufficient. Further conditions for actual instability of the hydraulic system are elasticity, inertia and the amount of energy dissipated in the system. Once more energy is transferred to the system than dissipated, the system becomes self-excited. Instabilities can occur in both pump and turbine operation of a pump turbine [1],[2].

If the head-flow curve has a negative slope in turbine operation, the discharge drops with increasing head. This behavior results in energy being transferred to the fluid giving rise to unsteady flow structures like vortices with the potential to excite oscillations in the system.

For pump turbines at turbine no load the criterion for instability can be expressed as: $\frac{dH}{dQ} < 0$

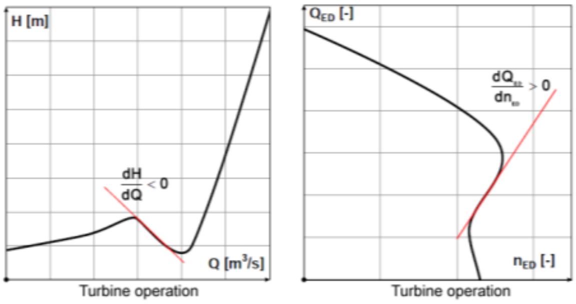

Figure 1: Slope of turbine characteristic at no load for constant wicket gate angle, in dimensional and normalized values [1].

Figure 1 shows a typical branch of a turbine characteristic fulfilling the described criterion for instability near no load at constant wicket gate opening in dimensional and dimensionless presentation. Due to the inverted S-shape of the curve in the dimensionless representation on the right, pump turbine characteristics fulfilling this instability criterion are often called S-shaped. Using the normalized quantities

normalized rotational speed $n_{ED} = \frac{n \cdot D}{\sqrt{g \cdot H}}$ and normalized discharge $Q_{ED} = \frac{Q}{D^2 \cdot \sqrt{g \cdot H}}$ (with reference diameter D and rotational speed n),

the criterion for instability in turbine operation can thus be expressed as

$$\frac{dQ_{ED}}{dn_{ED}} > 0.$$

Several authors describe the excitation of oscillations in hydraulic systems ([3], [4]) and sophisticated programs for system modeling are available. Recent publications on pump turbine flow at speed no load show an unstable interaction of flow in pump and in turbine direction in the runner channel [5]. One author describes the influence of several design parameters, such as profile shape, on the stability behavior of a pump turbine [6].

2. S-shaped instability in turbine operation

Reversible pump turbines are in general designed focusing on the pump flow, because the decelerated pump flow is more sensitive to flow separation, recirculation and losses than the flow in turbine direction. This constraint can lead to unstable operation near no load (runaway) in turbine operation in the event of unfavorable interaction of hydraulic machine and hydraulic system. Such instability is associated with fluctuations of head and discharge, which in turn lead to torque fluctuations on the shaft of the machine. During turbine startup and synchronization, such torque fluctuations are highly unwanted, as they slow down the synchronization and faster startup and switch-over times become increasingly important in today's electricity markets. In case of unstable interaction of machine and hydraulic system, stability can be accomplished by misaligning (de-synchronizing) a few wicket gates when starting the machines. This measure has been applied successfully, but it doesn't influence the root cause of the instability [8].

In order to identify the hydro-mechanic effects, which lead to the S-shape of the turbine characteristic and with the goal to identify criteria for the design of stable pump turbine runners, ANDRITZ HYDRO has initiated a long-term research project in collaboration of the Hochschule Luzern (Switzerland) ([1], [4]) and the Institute for Hydraulic Fluidmachinery at the Graz University of Technology (HFM) (Austria). The project comprises the numerical and experimental analysis of several pump turbines runners with stable and unstable behavior respectively.

3. Numerical analysis of vortex structures at runaway

In the scope of this project, two pump turbine runners were examined using CFD and measurements on the test rig. The runners have a specific speed of around n_q=44, with n_q defined as

$$n_q = n \cdot \frac{\sqrt{Q}}{H^{3/4}}$$

The number of wicket gates was 20 and the number of runner blades 9 for all examined pump turbine runner. All runners were analyzed using the same stationary components, i.e. spiral casing, stay vanes, wicket gates and draft tube (Figure 2).

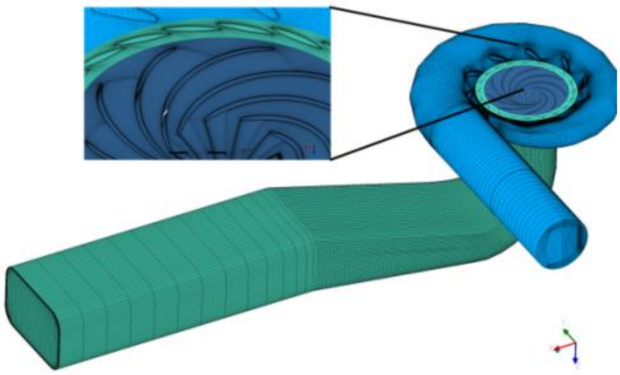

Figure 2: Simulation domain including the complete hydraulic machine.

At startup, the machines are synchronized to the frequency of the electrical grid at small wicket gate angles, usually around 6°. Instability in this operating range can increase the startup time or even inhibit synchronization completely. In order to resolve the problem at its source, detailed knowledge of the changes in the flow patterns in the pump turbine during this state of operation is essential.

When no torque is transferred to the shaft (no-load operation), input of hydraulic energy and dissipated energy are at equilibrium. For each wicket gate opening, such an equilibrium point exists, defining the runaway line. When the wicket gates are opened, the operating point moves along the runaway line with increasing rotational speed until synchronization speed is reached.

A comprehensive CFD-study of the flow field at runaway conditions led to a hypothesis about the nature of the vortex structures leading to potentially unstable characteristics [7], [9]. The excess hydraulic energy at runaway condition leads to pumping (flow opposite to the direction of the turbine flow) in parts of the runner channels. The interaction of inflow and forced outflow at the runner leading edge causes the formation of vortex structures. These vortex structures create additional losses and can block the through flow. This mechanism increases the pressure difference between the areas upstream and downstream of the vortex structure (Figure 3).

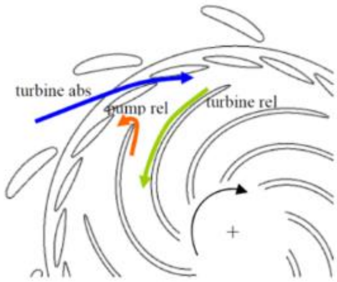

Figure 3: Flow structure driving the vortex formation in the vane-less space [1].

For flow rates at which the head discharge characteristic has a positive slope, or shows stable behavior, the described vortex structures are highly unstable. With lower discharge, the vortices sometimes become stable in time and space and literally block the through flow.

If the vortex structure is fully developed over the inlet area of all runner channels, it blocks the flow (Figure 4). This blockage leads to high pressure in the spiral casing and the high-pressure side of the system, leading to decreasing discharge at rising head (S-shape). The main driver of the primary vortex is a strong cross flow on the pressure side from hub to shroud at the leading edge. If this cross flow can be reduced or avoided, the S-shape of the characteristic can be avoided or reduced along with the associated system fluctuations.

Based on the CFD results, this blocking was identified as the reason for the negative head gradient and the pressure rise in the spiral. The stabilization of the unstable vortex formation is a likely cause for the characteristic with negative slope, i.e. potentially unstable interaction with the system.

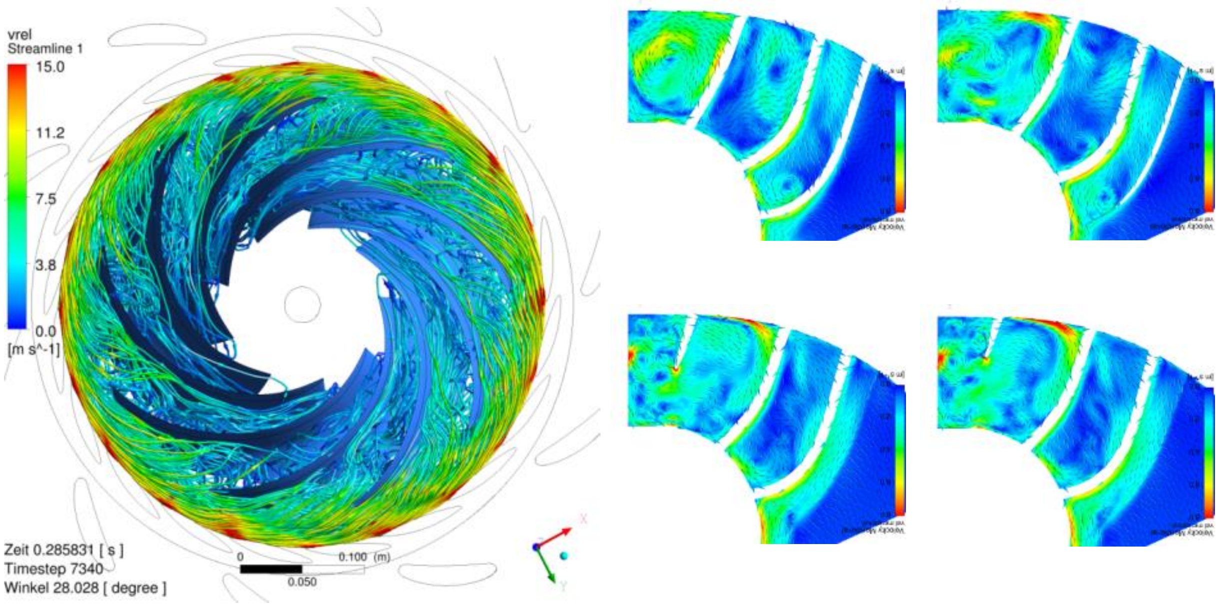

Figure 4: Stable vortex structure at runaway condition WG6° in a runner with S-shaped characteristics. Stream lines at one point in time (left) and meridional cross section at 4 time steps

4. Results of test rig measurements

In order to verify the hypothesis derived from the analysis of the unsteady CFD calculation with measurements, ANDITZ HYDRO initiated a joint project with the HFM. The scope of this project included three main measurements [10].

1. The measurement of the pump and turbine characteristics of the runners that had previously been analyzed numerically on a 4-quadrant test rig, including efficiency measurement according to IEC60193
2. The measurement of static and dynamic pressure characteristics at several locations up- and downstream of the runner, including the wicket gate channels
3. Measurement of the unsteady flow fields in the vane-less space between runner and wicket gates using particle image velocimetry (PIV)

The measurement showed s-shaped pump turbine characteristics for wicket gate angles between 5° and 15°. Depending in the stiffness of the test rig circuit – which could be changed using an upstream valve –, the operation became unstable or remained stable when passing through the s-shaped region. With the stiff test-rig circuit, it was thus possible to carry out continuous measurements for all operating points in the s-shaped region of the characteristics. The runner designed according to the criteria developed to avoid stable vortex structures in the vane-less space at runaway at the earlier stage of the project was built and measured on the 4 quadrant test rig of the HFM. The measurements of the turbine characteristics showed no S-shape at runaway, as was predicted by the previous CFD analysis (Figure 5).

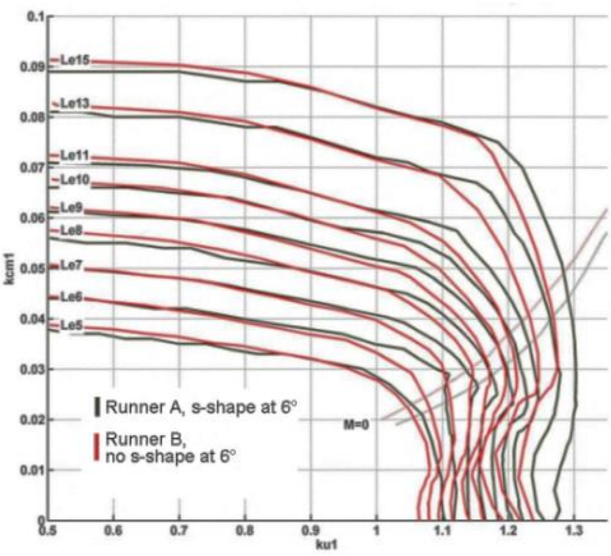

Figure 5: Turbine characteristics for runner with and without S-shape at the runaway curve.

As can be expected, for one wicket gate angle the dynamic pressure amplitudes clearly increase moving from the best efficiency operating point to the runaway curve and turbine brake operation (Figure 6). Especially at smaller wicket gate angles, the pressure amplitudes inside the machine, i.e. in the vane less space and at diameters smaller than the pitch circle diameter, are significantly higher than those farther upstream and downstream of the runner. This result indicates that the source of the instability is located in the runner or the vane less space. The frequency analysis of the dynamic pressure values measured at these locations detected no single dominant frequency other than the blade passing frequency. This result suggests the conclusion that no single characteristic frequency exists that could describe the vortex structure between wicket gate and runner.

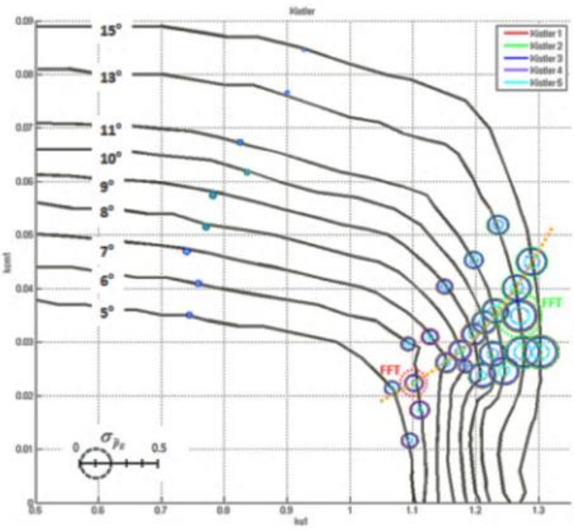

Figure 6: Characteristics of normalized discharge over speed showing the amplitude of dynamic pressure in the vane-less space and wicket gate channels

Relating to frequency and amplitude, the measurements showed no obvious difference between the pressure pulsations of the runner with a stable characteristic and that with an s-shape discharge-over-speed curve for 6° wicket gate opening.

Similarly, the records show no significant difference between the pattern of the unsteady pressure in the branch with negative slope and the branch with positive slope of the Q-n curve.

For larger wicket gate openings, the measurement showed pressure pulsations with a frequency lower than the runner rotation. This result can be taken as evidence of rotating stall cells in the upstream domain

5. Experimental analysis of vortex structures at runaway

On the test rig, unsteady velocity fields were recorded using PIV in 3 span-wise planes and one tangential, axis-parallel plane between runner and wicket gates. A publication by the Graz University of Technology describes the details of the PIV procedure [10].

For two pump turbine runners, flow fields were recorded for 3 wicket gate angles. PIV data was measured for the angle-specific best efficiency point and for several operating points around the slope change of the characteristics (Figure 7).

ANDRITZ HYDRO carried out additional unsteady CFD studies for the measured operating points in order to compare the measured and calculated flow fields. The underlying target is to validate the explanation of the fluid-mechanical mechanisms leading to potentially unstable turbine characteristics.

When comparing CFD and PIV results, it is important to consider the specifics of data acquisition for both methods. The raw data for the PIV results are recorded over a period of about 190 revolutions of the runner. The CFD simulations were performed for around 10 revolutions. This difference has to be kept in mind when comparing time or phase averaged results. During data processing, the PIV data is mapped to a regular grid whereas the CFD results in the plane of comparison are the result of spatial interpolation from the CFD calculation grid to the plane. The PIV data are available only for regions where reflections and blocked visual access don’t prevent proper acquisition while access to arbitrary planes in the calculation domain is not an issue in a CFD model.

IOP Conf. Series: Earth and Environmental Science **22** (2014) 032046 doi:10.1088/1755-1315/22/3/032046

In order to obtain meaningful values, the circumferential velocity component is normalized with the rotational speed of the runner leading edge. A value of one for the circumferential flow component thus signifies that fluid and runner leading edge have the same velocity.

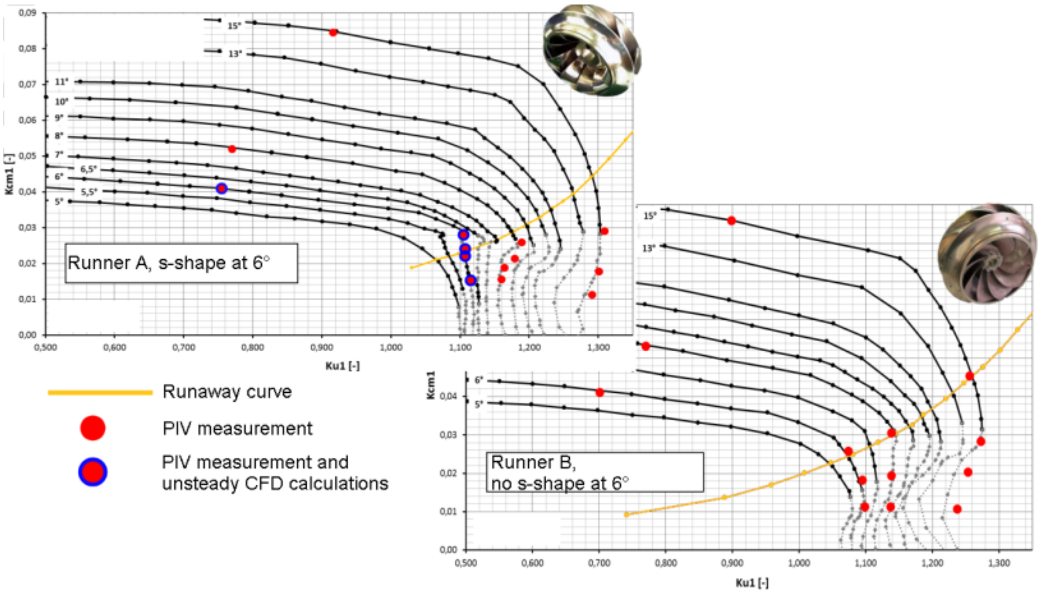

Figure 7: Discharge-speed characteristics showing the operating points with PIV and unsteady CFD results

The comparison of the PIV measured and calculated flow fields shows similar flow structures in the vane less space for the normalized phase averaged representation and the secondary flow field in the runner coordinate system, in which the average circumferential velocity is subtracted (Figure 8).

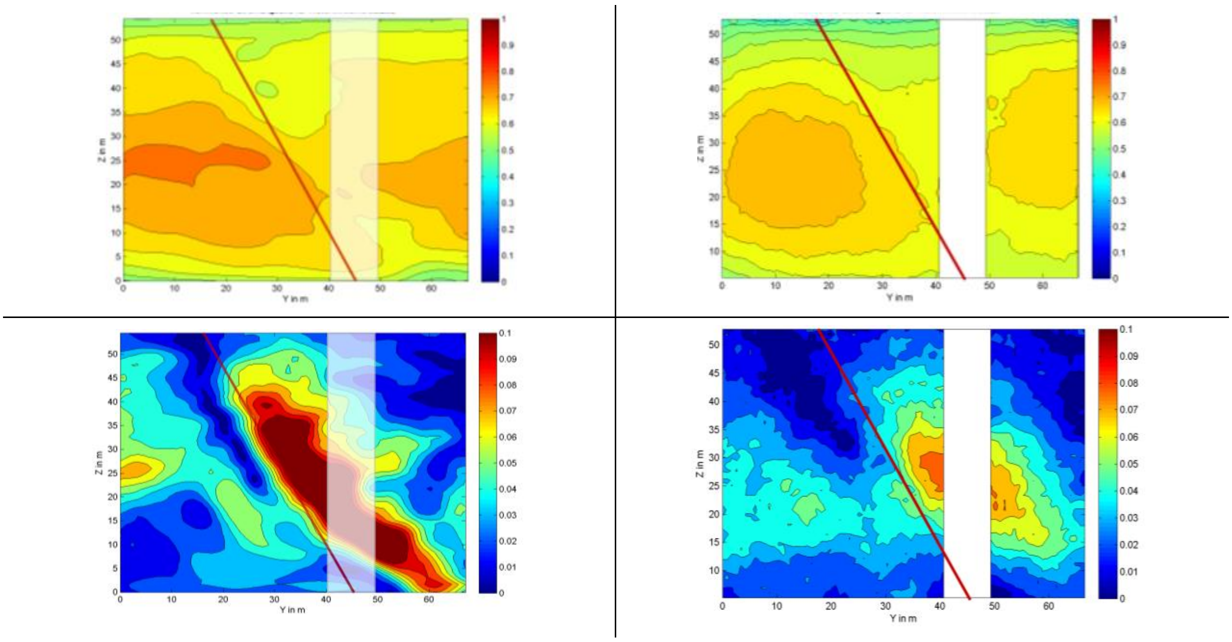

Figure 8: Comparison of the calculated (left) and measured velocity fields in the vane less space close to the no-load curve WG 6°, normalized velocity on top, secondary flow below, LE in red.

The evaluation of secondary flow field shows two regions with stronger secondary flow for the operating points close to the runaway curve. One region is located close to the leading edge and one in the center of the blade channel.

Both CFD and PIV data show a secondary flow field inside the blade channel moving with the rotational speed of the runner. The velocity of the secondary vortex is in the range of roughly 1/10 of the velocity in the phase averaged results.

6. Conclusion

In order to identify the mechanisms leading to unstable behavior of pump turbines, the turbine behavior at runaway was analyzed with numerical methods and on the test rig using PIV as well as dynamic pressure measurements. Based on earlier CFD-studies, the authors had identified a possible mechanism causing the S-shape of the turbine characteristic. In the measurements, a runner designed using criteria developed based on these findings showed stable behavior in the operating range relevant for synchronization.

The analysis of the dynamic pressure in the vane less space and in the wicket gate channels revealed a relatively broad-band pressure pattern with no specific frequency. This result suggests that there is no characteristic frequency that can be attributed to the vortex structure in the area of unstable behavior in turbine operation.

The flow survey for operating points in vicinity of the no-load curve with CFD and PIV showed similar flow fields, a finding that enhances the credibility of earlier CFD studies. This agreement supports the explanation of the mechanisms that lead to potentially unstable turbine characteristics.

References

[1] Staubli T, Widmer C, Tresch T, 2010, *Starting pump-turbines with unstable characteristics*, Hydro 2010, Lisbon, Portugal

[2] Gentner C, Smiljcic Z, Sallaberger M, 2010, *Analysis of unstable pump characteristics of pump turbines*, 16. internationales Seminar Wasserkraftanlagen, Vienna, Austria

[3] Dörfler P, Engineer A, Pendse R, Huvet P, Brahme M, 1998, *Stable operation achieved on a single-stage reversible pumpturbine showing instability at no-load*, 19th IAHR Symposium, Section on Hydraulic Machinery and Cavitation, Singapore

[4] Widmer C, Staubli T, Tresch T, Sallaberger M, 2010, *Unstable Pump-Turbine Characteristics and Their Interaction with Hydraulic Systems*, Hydrovision 2010, Charlotte, USA

[5] Liang Q, 2010, *Behaviour of Pump Turbines Operating at Speed no Load Condition in Turbine Mode*, Hydrovision 2010, Charlotte NC, USA

[6] Olimstad G, Nielsen T, Børresen B, 2012, *Dependency on Runner Geometry for Reversible Pump-Turbine Characteristics in Turbine Mode of Operation*, ASME, J. Fluids Eng., Vol. 134 (12).

[7] Widmer C, Staubli T, Ledergerber N, 2011, *Unstable Characteristics and Rotating Stall in Turbine Brake Operation of Pump-Turbines*, ASME, J. Fluids Eng., Vol. 133 (4)

[8] Klemm D, 1982, *Stabilisierung der Kennlinien einer Pumpenturbine im Bereich zwischen Turbinen-Teillast und Rückwärtspumpenbetrieb*, Voith Forschung und Konstruktion, Heft 28

[9] Gentner C, Sallaberger M, Widmer C, Braun O, Staubli T, 2012, *Numerical and experimental analysis of instability phenomena in pump turbines*, IOP Conf. Ser.: Earth Environ. Sci. Vol. 15 (3)

[10] Guggenberger M, Senn F, Schiffer J, Jaberg H, Gentner C, Sallaberger M, Widmer C, 2014, *Experimental investigation of the turbine instability of a pump-turbine during synchronization*, 27th IAHR Symposium Hydraulic Machinery and Systems, Montréal, Canada