Instability of Pump-Turbines during Start-up in Turbine Mode

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Introduction

During the last decade the deregulation in the European electricity market has resulted in rapidly changing conditions on the market. Due to the growing demand for balancing power and frequency control an investment in increased pumped storage capacity became economically feasible. Reversible pump-turbines seem to be in many cases the most cost-effective solution.

Occasionally torque fluctuations of reversible pump-turbines are encountered in power plants during start-up in turbine mode operation. Such fluctuations can slow down the process of synchronization what is highly undesirable when fast peak power production is required.

During start-up there is practically no load on the turbine shaft and the turbine operates close to the runaway characteristic. The guide vanes are opened only a few degrees during this phase.

A first case study of such oscillations on a model pump-turbine was presented by Yamabe [1] and [2]. He observed oscillations with pronounced hysteretic behavior which interacted with unsteady cavitation patterns. A case study and a simple cure of the problem by detuning some guide vanes are given by Klemm [3]. A linear stability analysis to predict the occurrence of the oscillations was successfully introduced by Martin [4] and [5]. Also Doerfler [6] presented a case study on how stable operation could be achieved in spite of the instability at no load.

Recent experiences with single stage reversible pump turbines are published by Billdal and Wedmark [7]. They propagandize multiflow guide vanes (MGV) to overcome difficulties with synchronization and to obtain stable speed after load rejection.

All authors agree that the so-called S-shape of the four quadrant characteristic of the pump turbines is responsible for the oscillations at no load operation. An example for such a four quadrant characteristic with S-shape near runaway is given in Figure 1. The vertical slopes of branches of the characteristics near runaway are directly linked to an exciting energy transfer from the flow to the oscillating system.

In the following a numerical study will be presented which focuses on the prediction of the characteristic near runaway and on the flow phenomena leading to the instability. To do so, tools were developed to analyze local and time-dependent flow, momentum and energy exchange in each of the runner and guide vane channels and in the vaneless spaces.

For validation of a model of a reversible pump-turbine with a known unstable behavior and well documented model test data was chosen. The four quadrant characteristic of this model turbine is given in Figure 1.

1. Numerical flow simulation

The flow near the no load operation of turbines becomes very complex in a sense that the flow is dominated by backflow regions and vortex formations in all parts of the turbine. Furthermore, partial pumping flows start to build up in some or all runner channels. Additionally, the flow becomes vigorously unsteady. Recirculation zones build up and disappear, vortical flows are swept away.
To predict such flows - at least qualitatively correct - grid generation must be carried out carefully. The grids used in this study were generated using only hexahedra elements. Grid generation was done with the commercial software ICEMCFD v11.0. Grid quality parameters are listed in Table 1. Table 2 gives the boundary conditions and the types of simulations which have been carried out.

![Image of Fig. 1. Measured four quadrant characteristics of the chosen pump-turbine model]

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<td>Outlet</td>
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Tab. 1. Grid quality parameters

Tab. 2. Boundary conditions and types of simulations
2. Validation

For validation the numerical flow simulations for operation near the runaway point experimental data from model test were used. The validation was carried out in two steps. In a first step stationary simulations were performed. The demand with respect to computational power is much lower for stationary simulations compared to unsteady, transient simulations. However, the expectations in the accuracy of the results of the stationary simulations are low, since the flow is certainly not stationary near runaway. In spite of this fact, the stationary simulated points follow well the measured points as demonstrated in Figure 3. General observations are that the stationary simulations predict well the slope of the characteristic near runaway and that the simulated points generally lie at lower $k_{u1}$ coefficients compared to the experiment.

On the other hand the transient simulations give results which are closer to the measured data and show a slightly overhanging characteristic near runaway – the typical S-shape.

3. Procedures to analyze fluxes

During mesh generation mesh-regions were defined for evaluation of local fluxes. This definition of mesh region which can be surfaces or volumes allows the analysis of local time variations of fluxes and balances, e.g. in each guide vane or rotor cannel. Figure 4a shows the control volumes defined for the guide vanes and the runner and Figure 4b shows surfaces defined in the vaneless space between runner and guide vanes and between guide vanes and wicket gate. In the following the fluxes through the entire, cylindrical surface A are analyzed.
The flow rate is the simplest example of the flux through a surface. In a three dimensional space the mass flow is defined by the following equation:

\[ Q(t) = \int_A [-\vec{n} \cdot \vec{c}] \, dA \]  

(1)

The normal vectors \( \vec{n} \) are orientated outwards with respect to the defined control volumes and velocity vectors \( \vec{c} \) are the absolute velocities on a surface fixed in space or the relative velocities on a rotating surface. With these vectors a sign of the flux is defined.

\[ Q(t) = \int_A [-\vec{n} \cdot \vec{c}] \, dA = Q_{in} + Q_{out} \quad \text{with:} \quad Q_{out} < 0, \ Q_{in} > 0 \]  

(2)

The absolute value of flow is:

\[ Q_{abs}(t) = \int_A [-\vec{n} \cdot \vec{c}] \, dA = Q_{in} - Q_{out} \]  

(3)

The flow rates of inflow \( Q_{in} \) and of outflow \( Q_{out} \) can be determined as follows:

\[ Q_{in}(t) = \frac{Q_{abs} + Q}{2} \quad Q_{out}(t) = \frac{Q - Q_{abs}}{2} \]  

(4)

The energy flux (e.g. total energy, kinetic energy) or the components of the momentum flux can be defined in the same manner. The total energy flux, also used in the following, is:

\[ \dot{E}_{tot}(t) = \int_A p_{total}(t)[-\vec{n} \cdot \vec{c}] \, dA = \dot{E}_{in} + \dot{E}_{out} \]  

(5)

\[ \dot{E}_{tot,abs}(t) = \int_A p_{total}(t)[-\vec{n} \cdot \vec{c}] \, dA = \dot{E}_{in} - \dot{E}_{out} \]  

(6)

\[ \dot{E}_{tot,kin}(t) = \frac{\dot{E}_{tot,abs} + \dot{E}_{tot}}{2} \quad \dot{E}_{tot,kin}(t) = \frac{\dot{E}_{tot} - \dot{E}_{tot,abs}}{2} \]  

(7)

Definition of power coefficients:

\[ K_{E_j}(t) = \frac{\dot{E}_j(t)}{\rho \cdot g \cdot H \cdot Q} \]  

(8)

4. Results

The process of energy dissipation for operating points near runaway involves in- and outflows from the runner. The high energy flow is entering the runner from the guide vanes and drives the runner up to speed where parts of the channel start to pump flow outwards. The equilibrium of energy input and dissipation by pumping results to zero torque at the shaft.
The discharge being pumped out of the runner has to reenter the runner. This increases the inflow into the runner above the flow rate given at the inlet to the turbine scroll. This process of pumping seems to be an unsteady process for the investigated model turbine for an operating point slightly above runaway.

Figure 5 shows a typical time history of flow rate fluctuations through the surface A in the vaneless space between guide vanes and runner over three revolutions of the runner. The normal vector is defined here as pointing radially outwards. The mass conservation is satisfied, as the evaluated flow rate $Q(t)$ through the surface A remains constant in time at the given value $Q$ at the inlet. Accordingly, fluctuations of the outflow $Q_{\text{out}}(t)$ and fluctuations of the inflow into the runner $Q_{\text{in}}(t)$ have to be in equilibrium.

![Figure 5. Time varying flow rates through the surface A of the vaneless space](image)

These flow rate fluctuations seem to be localized to the vaneless space between guide vanes and runner, since at the outlet of the turbine no fluctuations are observed as shown in Figure 6. Here, the normal vector is defined in direction of the draft tube. The backflow $Q_{\text{in}}(t)$ in direction of the runner amounts to about one third of the discharge $Q$ at the inlet to the turbine. All simulated flow rates are almost constant in time.

![Figure 6. Flow rate fluctuations at the outlet of the turbine](image)

The periodical flow rate fluctuations through the surface A in the vaneless space are linked to the torque fluctuation on the shaft. In phases in time where a partial pumping flow builds up in the runner channels, the torque increases simultaneously, Figure 7.
When analyzing the energy fluxes through the surface A of the vaneless space, we see that the power coefficients $K_{E_{tot\,in}}(t)$ and $K_{E_{tot\,out}}(t)$ fluctuate out of phase and at the same frequency as the mechanical torque fluctuations $K_{P_{mech}}(t)$. The resulting power coefficient $K_{E_{tot}}(t)$ fluctuates with a certain phase shift with the mechanical power indicating that inertia effects of the rotating water masses might be involved.

The question arises now how these flows lead to energy transfer to the vaneless space and how the in- and outflows look like in detail. Figure 9 clearly demonstrates the existence of enhanced vortices transporting fluid outwards. These vortices exit the runner channels in front of the leading edges of the runner vanes into the vaneless space. The vortex strength varies in time and space. For the chosen operating point, which is slightly above the runaway point, the variation in time is dominant, which results in the global flow rate fluctuation through the surface A. It can be assumed that with decreasing flow rate Q at the inlet to the turbine the effect of the spatial variation of the vortex formation will more and more dominate and that rotating stall will be observed for operating points below runaway, as it was experimentally observed for a pump turbine e.g. by Staubli [8].

The difference between the in- and out-energy fluxes through the surface A indicates that a large amount of the energy dissipation occurs in the vaneless space between guide vanes and runner for operating points near runaway.
5. Conclusions

The characteristics of the pump turbine close to runaway could be well predicted with transient flow simulations. Unstable flow fields were predicted for the simulations in the so-called S-shaped portion of the characteristic. This simulated instability shows time-varying in- and outflow from the runner into the vaneless space. For the investigated operating point, slightly above runaway, the band of the fluctuations corresponded to about 50 percent of the main inflow to the turbine.

The existence of unstable operation is confirmed by the model test where also instability was observed in this range of operation.
With detailed information available in the simulated flow field local flow effects could be analyzed. It could be concluded that local vortices forming in the runner channels close to the leading edge is the source for the unsteady in- and outflow from the runner into the vaneless space between guide vanes and runner. Therefore, the vortices and the induced outflow can be considered as the origin of the instability.

Most of the energy dissipation for operating points near runaway occurs in the vaneless space between guide vanes and runner.

Acknowledgement

This study was made possible by a grant of the Swiss Commission for Technology and Innovation (CTI) and swisselectric research. Industrial funding was provided by VA TECH HYDRO.

References


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