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Numerical characterization of pressure instabilities in a vaned centrifugal pump under partload condition

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Abstract. This paper studies the hysteresis/saddle phenomena of the head-drop in a scaled model pump turbine using CFD methods. This lag was induced by complicated flow patterns, which influenced the reliability of rotating machine that was analysed by a commercial code with DES model for computing turbulence. Analyses were carried out on the pressure signals both in frequency and time-frequency domains at full and part load conditions. The results highlighted the remarkable interaction between the unsteady structures in diffuser and return.

1. Introduction

Nowadays new renewable energy sources such as wind and solar power constitute a larger piece of the national energy mixture. These intermittent energy sources have varying and, to some extent, unpredictable production. The pumped storage power plants can play an important role in stabilizing the electric power system when the electric production is excess or lacking. Consistently, many new pumped storage power plants in central Europe are recently initiated with an urgent need for the production stabilization.

A reversible pump turbine is widely applied, which is regarded as the most cost effective solution, even if there are also other technical arrangements such as the combinations Francis turbine/pump or Pelton turbine/pump. The reversible pump turbine can, depending on reservoir size, deliver long term energy storage, and is able to boost production (turbine) or consumption (pump) in peak power situations. However, pump-turbines often involve problematic S-shaped regions in their machine characteristics. While the pumped storage may solve some problems in the grid, pump-turbine operation and control can lead to other problems including severe self-excited oscillation in the hydromechanical system.

At off design conditions, the distributor and the draft tube do not work properly and give awkward boundary conditions to the impeller. In addition, there is a strong interaction between the runner and these parts. Flow features such as separation and recirculation occur severely in an unsteady manner. In particular, guide vanes may experience strong vibrations. High cycle fatigue stress may result in the propagation of cracks and the failure of shear pin or guide vanes stem. The damage root causes can range from the misalignment during shear pins assembly on the guide vane activation mechanism, which causes anomalous loading, to the strong excitation due to the Rotor-Stator Interaction (RSI).

One objective of this investigation is to understand the underlying physical mechanism of pump-mode instability of a two stages reversible-pump turbine.

Among the phenomena, the flow-dynamical unsteady stemming from the RSI is still an open ques-
tion and challenge fluid mechanical committee. During the past several decades, some experiments and numerical simulations have been carried out to study the effects of impeller/diffuser geometries and operating conditions on unsteady interactions by Gonzalez et al. [1], Hong and Kang [2], Guo and Maruta [3], Majidi [4], Rodriguez et al. [5], Pavesi et al. [6], Cavazzini et al. [7], and Feng et al. [8].

Besides, the analysis of the unsteady pressure signal is an effective method to understand the characteristics and origin of the interactions. Several techniques (auto and cross spectra, coherence function, wavelets, etc.) are frequently used to analysis the frequency and time-frequency signal, and to identify the flow structures and the propagation of the unsteadiness [9-12].

Even though previous studies advanced the understanding of the unsteady phenomena in pumps, a further investigation of the characteristics and flow mechanisms of the unsteadiness is valuable.

In the present paper, numerical study of this reversible-pump turbine was used to capture large-scale instabilities due to the dynamic interaction between rotor and stator. In order to capture the possible fluid-dynamical evolution of the instabilities and investigate their origin, a spectral analysis of the pressure pulsations obtained with ANSYS CFD 14.0 was compared at design and part load conditions.

2. Components Description and Computational Model
The analyses were carried out on a low-pressure stage of a two stages pump-turbine in the pump operating mode. The model consists of a radial, shrouded impeller, shown in figure 1, with seven 3D backward swept blades with a discharge angle of 26.5° referred to the tangent and a design specific speed \( n_s = 37.6 \text{ m}^{0.75} \text{s}^{-1} \) (dimensionless design specific speed \( \omega_s = 0.71 \)).

Refeeding channels were used to guide the flow that leaves the impeller to the inlet of the subsequent. The channels were made up of twenty two adjustable guide-diffuser vanes and eleven continuous vanes. The guide-diffuser allows continuous and independent adjustment of the vane angle and of relative azimuthally position with the return channel vanes. The radial gap between the impeller tip and the inlet edge of the stator vanes, with the configuration under test, was 5 mm, which is 2.5% of the impeller radius and the relative azimuthally position of the diffusers’ vanes was fixed rotating the system of 8 degrees from the face to face configuration.

The most noteworthy data of the impeller and diffuser vanes are reported in table 1.

The numerical analyses were carried out for some flowrates (\( Q/Q_{\text{Des}} = 0.4571, 0.5325, 0.630, 0.674, \) and 1.000). Only design flow rate and \( Q/Q_{\text{Des}} = 0.630 \) will be presented in this paper.

3. Computational Model
The commercial software package ANSYS 14.0 was used to discretize and perform the numerical simulations of the entire machine.

The draft tube was discretized by a structured mesh of about 339500 elements. No grid dependency study has been performed for the draft-tube mesh. Anyway, care was taken to have a \( y^+ \) value of approximately 30 at the first elements close to the wall.

An O grid for the runner investigation was performed. Preliminary tests were carried out to study the grid dependence and to guarantee an accurate and grid independent solution for the runner. The stage head \( H \) and the stage efficiency calculation with about 200 000 elements per passage had a grid independent solution. The adopted number of the elements was arbitrarily increased of the 85 percent to guarantee the capacity of numerical solution to capture the local pressure pulsation as well. The impeller computational domain used had a total 2.6 million cells with \( y^+ \) values below 30. O-type grids were adopted for both the diffuser and the return channel discretization with about 2.9 and 3.3 million cells, respectively. The leakage from the labyrinth seal was also considered and several H-blocks were built to describe the cavities.

On both blades and wall surfaces, the boundary layer was assumed fully turbulent. The detached eddy simulation (DES) model was chosen as turbulence model. The shear stress transport k-\( \omega \) model covered the boundary layer while the Smagorinsky-lilly model was applied in detached regions.
For the interface between stator/rotor blocks, the standard transient sliding interface approach was chosen. The mass flow rate with stochastic fluctuations of the velocities with 5% free stream turbulence intensity was described at the inlet and the average static pressure was fixed at the outlet.

The scheme adopted for the time discretization was a second-order implicit time stepping. The time step definition was based on the impeller rotation and it was of about one degree. So, the RMS courant number was CFL ≈ 0.15. A maximum number of five iterations were fixed for each time step, resulting in a mass residue of $10^{-6}$, momentum residues of $10^{-4}$, and turbulence kinetic energy and energy dissipation residues of $10^{-4}$.

In pump-mode the flow has to be turned from radial outwards to radial inward and again to the axial direction, while simultaneously ensuring the conversion of kinetic energy into static pressure rise.

Additional the strongly swirling flow has to be normalized to ensure a velocity field at the inlet of the next impeller which has no swirl, to enable the maximum pressure rise in the next runner stage.

The return channel system is probably the least understood component of a multistage system because the flow through the 180° U-turn bend, and its interaction with the down- and upstream vane rows, presents an extremely complex fluid dynamic problem. Unsteady pressure was measured in the diffuser and return channel, by 12 and 19 monitors set on mid span surface of one vane in diffuser and return channel, respectively. Figure 2 shows the distribution of the monitors in a channel. The pressure signals were acquired for 14 impeller revolutions, corresponding to about 1.4s after 20 revolutions required to achieve a quasi steady simulation convergence.

### 4. Signal processing

The pressure signals were analyzed both in the frequency domain and the time–frequency domains by MATLAB. The power spectral was used for the frequency analysis, whereas a time-frequency analysis was carried out by the wavelet transforms. For this analysis, the continuous wavelet transform \( W(s, n) \) of the discrete sampled pressure signal \( x_n \) was computed via the FFT-based fast convolution:

\[
W(s,n) = \sum_{t=n}^{N-1} X_k \left( \frac{2\pi s}{\delta t} \psi^*_0(s\omega_k) e^{i\omega_k n \delta t} \right)
\]

where \( s \) is the wavelet scale, \( n \) is the localized time index, \( k \) is frequency index, \( k \) is the discrete Fourier transform (DFT) of \( x_n \), \( N \) is the data series length, \( \delta t \) is the sample time interval, \( \sqrt{2\pi s/\delta t} \) is a normalization factor which could obtain unit energy at each scale. \( \psi^*_0(s\omega_k) \) is the complex conjugate of the Fourier transform of the scaled version of the “mother wavelet” \( \psi(t) \) and \( \omega_k \) is the angular frequency. The choice of the mother wavelet depends on several factors [3] and in this paper complex Morlet wavelets were used (\( 2\pi f_0 = 6 \) was chosen), since it could provide a good balance between time and frequency localization and it returned information about both amplitude and phase.

### 5. Results

The characteristics of the pump were evaluated in accordance with ISO standards (Figure 3). The

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**Table 1. Geometry characteristics and performance parameters of the tested pump-turbine.**

|                     | Impeller data |                  | Diffuser vanes data |                  | Return channel vanes data |
|---------------------|---------------|------------------|---------------------|-------------------|---------------------------|
| \( D_1 \) (mm)      | 400           | \( B_1 \) (mm)   | 40                  | \( n_b \)         | 7                         |
| \( n_b \)           | 7             | \( \beta_{2e} \) (°) | 26.5                | \( \phi_{Des} \)   | 0.125                      |
| \( D_2 \) (mm)      | 410           | \( B_2 \) (mm)   | 40                  | \( n_b \)         | 22                        |
| \( \alpha_{3c} \) (°) | 22            | \( \lambda \) (°) | 21.8                | 8                 |
| \( D_3 \) (mm)      | 516           | \( B_3 \) (mm)   | 40                  | \( n_b \)         | 11                        |
| \( n_b \)           | 11            | \( \alpha_{4c} \) (°) | 11                  | 30                |                           |
characteristic of the impeller becomes slightly unstable between $\phi/\phi_{Des} = 0.45$ to $0.70$. Below $\phi/\phi_{Des} = 0.40$ it rises due to the effect of fully developed inlet recirculation.

The numerically calculated impeller characteristic is also shown in figure 3 with the head oscillation band. Outside the head discontinuities on the lower branch of the saddle/hysteresis good numerical convergence was always obtained. Below the transition between the two head-branches numerical convergence was not obtained. This paper is mainly focused on the hysteresis/saddle phenomena and the results of $0.674 Q_{Des}$ was analysed more in deep.

5.1. The unsteady phenomena at design load

Stator-rotor interaction causes a pressure pulsation system. Figure 4 shows the velocity field near the hub, at the mid span and near the shroud in the impeller, diffuser and the first half of the return channel at design load. No unsteadiness develops inside the impeller and the diffuser whereas some swirls appear in the return channel due to return channel geometry. The power spectra are dominated by the blade passage frequency (BPF St=1), the impeller rotating frequency (1/7 BPF – St = 0.143) and their harmonics (Figure 5). In the return channel some more peaks in the low frequency range (St<0.3) can be identified (Figure 5). They appear to be related to local swirl but not to significant unsteady phenomena.

Figure 6 reports the wavelet diagram of the pressure signals acquired in the return channel (blade 2 point 10 – see figure 2 for the monitor point position). It confirms the foregoing statements and the pressure pulsation due to the blade and impeller passage were always present.

5.2. The unsteady phenomena at part loads

When the pump operated at part loads, unsteady structures started to appear. Figure 7 shows the flow fields inside the impeller, the diffuser and in the first part of the return channel at mid-span. Decreas-

![Figure 2](image_url) Distribution of monitor points.

![Figure 3](image_url) Experimental and numerical pump characteristic

![Figure 4](image_url) Flow field in the impeller, diffuser and the first half of the return channel at design flow rate
ing the flow rate, a reverse flow started to develop on the suction side of the impeller blades. This reverse flow appeared in the second half of the impeller channel and moved along the blade till it reached the trailing edge in correspondence of which it was absorbed by a wake zone.

As regards the diffuser, the flow field seemed to be much more perturbed in comparison with the design flow rate. Vortexes, which occurred near the trailing edge of the diffuser vanes, appeared and disappeared, partially or totally choking the flow rate coming out from the impeller with a consequent flow transfer to the adjacent channel.

**Figure 5.** Comparison of the power spectra of numerical pressure signals acquired in the diffuser and in the return channel at $Q_{\text{Des}}$.

**Figure 6.** Wavelet magnitude $|W_n|$ of the pressure signal (point 6) in the return channel vane at design load.

**Figure 7.** Flow field in the impeller, diffuser and the first half of the return channel for 0.674 $Q_{\text{Des}}$. (a) Near the hub (b) Mid span (c) Near the shroud.
The spectral analysis carried out on the numerical signals acquired for both the considered flow rates identified more frequencies in comparison with the design operating condition. Figure 8 reveals the power spectra of the pressure signals acquired in the diffuser and in the return channel for 

\[ 0.674 Q_{\text{Des}} \]

Besides the impeller rotating frequency, the blade passage frequency and their harmonics, the power spectra resulted to be characterized by a peak at \( St=0.33 \), also captured by the time-frequency analysis that highlighted its unsteady characteristics (Figure. 9).

To better visualize the zones of the machine in which these frequencies were more intense, a map highlighting the most important peaks of each monitor point was built (Figure 10). The interaction zone between the diffuser and the return channel, was interested by the development of several unsteady vortexes and the power spectra were dominated by three frequencies: the impeller rotating fre-
quency (St=0.143), the frequency associated to unsteady phenomena (St=0.33) and a third one (St=0.47), that resulted to be due to the non-linear interaction between the two previous ones (St=0.47=0.143+0.33). The intensity of the unsteady structure remains significant even along the blades of the return channel, as demonstrated by its domination in the power spectra of all the remaining monitor points (Figure 10).

6. Conclusions

The pressure instabilities in a vaned centrifugal pump were studied at design and part load conditions by numerical analysis. Pressure signals, acquired from several monitor points located on the diffuser and return channel vanes, were analysed by spectral analysis both in the frequency and time frequency domains in order to evaluate the evolution in time of the unsteady pressure pulsations.

The analysis of the pressure signals highlighted the presence at part loads of a pulsating phenomenon at a frequency of St= 0.33 both in the interaction zone between the diffuser and the return channel and along the blades of the return channel. As regards the flow field at part load conditions, an unsteady reverse flow was identified inside the impeller on the blade suction side, moving along the blade and disappearing in the wake zone near the impeller blade trailing edge. Moreover, a series of unsteady vortexes were identified in the interaction zone between the diffuser and the return channel as well as inside the return channel, partially or totally blocking the flow rate coming out from the diffuser channel. This blockage action determined a flow transfer in the circumferential direction.

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