Dynamic Runner Forces and Pressure Fluctuations on the Draft Tube Wall of a Model Pump-Turbine

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Abstract. When Francis-turbines and pump-turbines operate at off-design conditions, typically a vortex rope develops. The vortex rope causes pressure oscillations leading to fluctuations of the forces affecting the runner. The presence of dynamic runner forces over a long period of time might damage the bearings and possibly the runner. In this experimental investigation, the fluctuating part of the runner forces and the pressure oscillations on the draft tube wall were measured on a model pump-turbine with a simplified straight cone draft tube in different operating conditions. The investigation focuses on the correlation of the pressure fluctuations frequency measured at the draft tube wall with the frequency of the fluctuating forces on the runner. The comparison between pressure fluctuations and dynamic forces shows a significant correlation in all operating points. For the comparison of different components in the spatial directions of the forces, the pressure fluctuations were separated in a synchronous part and a rotating part for operating points with higher amplitudes. The rotating pressure fluctuations correlate with the radial forces especially in the operating points with a rotating vortex rope. At frequencies with higher amplitudes in the pressure fluctuations caused by the vortex rope movement, there are also higher amplitudes in the radial forces at the same frequencies.

1. Introduction
The draft tube is an important part of a hydraulic turbine. It converts kinetic energy at the runner outlet into pressure energy. In this way it increases the head to the runner. One difficulty in hydraulic turbines is the existence of an unsteady vortex rope in the draft tube, especially for Francis turbines running in part load. When a turbine operates at off-design conditions, the flow in the draft tube has not only a transport component, but also a circumferential component. At these off-design conditions a vortex rope in the draft tube can appear. The appearance of the vortex rope is related to the strength of the swirl at the runner outlet [1]. Flow separation at the hub and the swirl can lead to a wake region behind the hub in the center of the draft tube. If the swirl is strong enough, a rotating vortex rope can appear at the shear layer between the wake region of the runner hub and the region with higher transport component [2, 3]. The rotation of the vortex rope causes a rotating pressure oscillation. Also a synchronous pressure oscillation can occur, i.e. a pressure oscillation of the whole level in a respective cross-section of the draft tube [4]. The pressure oscillation may lead to fluctuations of the forces affecting the runner. The fluctuating forces affecting the turbine runner are recognizable as dynamic shaft forces. The dynamic shaft forces cause vibration and can lead to thrust and guide bearing problems [5, 6].
2. Experimental Setup and conversion of the results

The measurements were carried out at the closed-loop test rig of the Institute of Fluid Mechanics and Hydraulic Machinery (IHS). The relevant parts of this test rig are illustrated in figure 1. The test rig is equipped with two pumps for parallel or serial operation. The two pumps were driven by a speed-regulated direct current motor. The pumps deliver the water to the headwater vessel. The water flows from the vessel to the pump-turbine model. The pump-turbine is also connected to a speed-regulated direct current motor-generator. A straight acrylic glass cone is installed into the pump-turbine representing the draft tube; see figure 2. The cone has a diameter of 182 mm at the inlet and a diameter of 400 mm at the outlet. The cone angle relative to the center axis is ten degrees. Downstream the acrylic glass cone, a straight pipe with a nominal diameter of 400 mm and a length of approximately 2.4 m is installed. Downstream, the water flows into the tailwater vessel, which is built with an air chamber on the top of the vessel, in order to vary the pressure level of the test rig.

The pressure at the draft tube wall was measured with piezo-resistive transducers. Four pressure transducers are located in a cross section of the draft tube cone. The cross section is placed 99 mm downstream of the cone inlet. The positions of the pressure taps is depicted in figure 3. It was assumed that the transducers measured nearly the same pressure fluctuation amplitudes, thus, one transducer was selected for all presented results in this paper. The fluctuations of the measured pressure are normalized with the net head, equation 1. All presented frequencies are normalized with the rotational frequency of the runner (23.3 Hz). The accuracy of the measured pressure fluctuations is 0.17% based on the corresponding net head condition.

\[ \Delta \Psi = \frac{2 \cdot \bar{p}}{H \cdot \rho \cdot g} \]  

The dynamic forces and torques on the turbine runner are measured with a gauging shaft. A sectional drawing of the gauging shaft is depicted in figure 4. In the gauging shaft three three-component quartz force transducers are integrated in the vicinity of the runner. Nine signals are transmitted by slip rings from the rotating shaft to the fixed housing. The cables from the slip ring housing are connected with charge amplifiers, which transform the electric charge signals into electrical voltage. At the casing of the slip rings a shaft encoder is installed to measure the angle position of the shaft.

During the conversion of the signals, both forces and torques were transformed into a fixed frame of reference (Cartesian coordinates) by means of the angle position of the shaft. The steady-state
forces and torques were eliminated and the amplitude at the frequency of the runner speed (±0.75%) was cut off with a filter. This was done because these fluctuations were affecting the rotating shaft as steady state forces. With the used measurement device this steady state forces could only be measured with very high effort in an adequate accuracy. Additionally, the out-of-balance of the rotor generates forces at this frequency. This could lead to additional high amplitudes at the runner frequency which were not caused by the flow structure in the turbine. Afterwards, the results transformed to the frequency domain and corrected by frequency dependent calibration factors. Then the fluctuations of the components of the forces and torques were normalized by head and geometrical parameters of the runner according to equation (2) and equation (3). The x-axis points towards the penstock and the z-axis points in axial direction downwards.

\[ F_x = \frac{c_{Fx}}{\rho \cdot g \cdot H \cdot b \cdot D^2} \]
\[ F_y = \frac{c_{Fy}}{\rho \cdot g \cdot H \cdot b \cdot D^2} \]
\[ F_z = \frac{4 \cdot c_{Fz}}{\rho \cdot g \cdot H \cdot \pi \cdot D^2} \]
\[ M_x = \frac{c_{Mx}}{\rho \cdot g \cdot H \cdot b \cdot D^2} \]
\[ M_y = \frac{c_{My}}{\rho \cdot g \cdot H \cdot b \cdot D^2} \]
\[ M_z = \frac{c_{Mz}}{\rho \cdot g \cdot H \cdot \frac{\pi}{4} \cdot D^3} \]

In the dynamic calibration of the gauging shaft the first eigenfrequency was identified as approximately 6.5 times of the rotational frequency. The calibration procedure and results are described in [7]. To keep the accuracy of the forces and torques lower than 10% of the measured fluctuations all results were presented with a maximal frequency of approximately two times the rotational frequency.

3. Investigated operating points

Prior to the investigation of selected operating points the turbine characteristics were measured, in order to specify the operating points for the investigation. Figure 5 shows the hill chart in a \( Q_{11} \) against \( n_{11} \) diagram. The operating points, which are selected for the investigation, were marked with the numbers 1 to 4. Additionally the best efficiency point (BEP) was marked with a blue diamond. The selected operating points have nearly the same specific speed as the best efficiency point. The red line in figure 5 marks some additional investigated operating conditions.

Table 1 lists the operating conditions of selected operating points in relation to the BEP. Operating points 1 and 2 are at part load and operating points 3 and 4 are at overload conditions.
Table 1. Data of selected operating points.

|    | $Q_{11}/Q_{11,\text{BEP}}$ | $n_{11}/n_{11,\text{BEP}}$ | $P_{11}/P_{11,\text{BEP}}$ |
|----|-----------------------------|-----------------------------|-----------------------------|
| 1  | 72%                         | 101%                        | 69%                         |
| 2  | 83%                         | 100%                        | 82%                         |
| BEP| 100%                        | 100%                        | 100%                        |
| 3  | 111%                        | 100%                        | 110%                        |
| 4  | 135%                        | 100%                        | 129%                        |

Figure 5. Hill chart, specific discharge $Q_{11}$ against specific speed $n_{11}$.

To characterize those operating conditions the cavitating vortex-structure in the draft tube is visualized. The measurements of pressure fluctuations, forces and torques acting on the runner were performed at a high-pressure level i.e. at non-cavitating conditions. Figure 6 and figure 7 show partial load vortex ropes of the operating point no. 1 and the operating point no. 2.

Figure 6. Image of the visualised structure at operating point no. 1.  
Figure 7. Image of the visualised structure at operating point no. 2.

In figure 8 the straight overload vortex of operating point 3 is depicted. Figure 9 shows a helical vortex rope of operating point no. 4. In this high overload operating point the strength of the swirl is strong enough to develop a helical vortex rope. The curvature of the vortex rope is in the opposite direction than the curvature of the other ropes at part load conditions. The movement of the vortex rope was also observed with video sequences. The rotational direction of the vortex rope movement is against the rotational direction of the vortex ropes movement at the part load operating conditions. Skoták et al. [9] shows the same behavior at the variation of $n_{11}$ below and above the BEP.
4. Results

In this paragraph the results of the five operating points are presented by comparing the pressure fluctuations with the forces and torques. Additionally, the waterfall diagrams of the pressure fluctuations were correlated with the axial and radial forces and torques.

4.1. Operating point no. 1

The pressure fluctuations measured in operating point no. 1 are depicted in figure 10. The values of the pressure fluctuation coefficient are in a range between -0.02 and 0.01. The pressure decreases in intervals of approximately 0.15 s. The amplitudes of the pressure fluctuation coefficient show a strong peak at approximately 30% of the runner speed (see figure 11). There are also higher harmonics of this frequency in the pressure fluctuation coefficient visible.

Figure 10. Pressure fluctuations at operating point no. 1 in the time domain.

Figure 11. Pressure fluctuations at operating point no. 1 in the frequency domain.

Figure 12 shows the synchronous part and the rotating part of the pressure fluctuations. The amplitudes of the rotating part of the pressure fluctuations are much higher than the amplitude of the synchronous part. The amplitudes of the radial forces and tilting torques show a peak at approximately 30% (see figure 14) caused by the rotating part of the pressure fluctuations. A small increase by the components in axial direction could be seen in figure 15. This can be explained by the low synchronous part of the pressure fluctuations. Furthermore, small peaks at double runner speed are existent in all forces and the tilting torques. The peak at the frequency of the runner speed could not be seen in the forces and torques, because of the cut off filter at this frequency. Figure 13 shows the radial forces in x- and y- direction in the time domain. In addition to the measured signal a moving average of 20 values is shown. It can be seen, that the moving average signal in y-direction is nearly a sine wave with a shift of 90° to the signal in x-direction. This is caused by a strong rotating radial force. This radial force rotates with the rotating frequency of the vortex rope. The period of the sine wave is approximately 0.15 s.
4.2. Operating point no. 2

In figure 7 a very clear shape of a rotating vortex rope is shown in that operating point, but figure 16 does not show strong periodic fluctuations. The values of the pressure fluctuation coefficient are in a range between -0.005 and 0.005. This is approximately three times smaller than the range of the pressure fluctuations in operating point 1.

**Figure 16.** Pressure fluctuations in the time domain.

**Figure 17.** Pressure fluctuations in the frequency domain.

**Figure 18.** Forces and torques in the frequency domain.
In figure 17 no distinguish peak in the amplitudes of the pressure fluctuation coefficient is visible. This verifies the observation in the time domain. The maximum amplitude is more than ten times lower than the maximum amplitude in operating point no. 1. There are higher values at lower frequencies up to 60% of the runner speed. The highest value is at approximately 20% of the runner speed. Figure 18 shows the forces and torques in the frequency domain. In all force and torque signals peaks are visible at double runner speed. Also a small increase of the amplitudes for low frequencies can be seen, mainly in the radial forces for lower frequencies up to 65% of the runner speed. This is caused by the increase in the pressure fluctuations.

4.3. Best efficiency point (BEP)
At best efficiency point no vortex structure was observed. The pressure fluctuations measured in best efficiency point are depicted in figure 19. The values of the pressure fluctuation coefficient are also in a range between -0.005 and 0.005. This is in the range of operating point 2 and approximately three times smaller than the range of the pressure fluctuations in operating point 1. Figure 20 shows pressure fluctuations in the frequency domain. There are no strong peaks and the maximum amplitude is more than ten times lower than the maximum amplitude in operating point no. 1. Figure 21 shows the forces and torques in the frequency domain. There are peaks at double runner speed for the forces and the torques. Also an increase of the amplitudes to low frequencies can be seen.

4.4. Operating point no. 3
Figure 8 shows a central vortex shape at this overload operation point.
The pressure fluctuations measured in operating point no. 3 in the time domain are depicted in figure 22. The values of the pressure fluctuation coefficient are in a range between -0.01 and 0.005. This is approximately half range of the pressure fluctuations in operating point 1. It looks like a periodic signal with a higher frequency compared to the pressure fluctuations in operating point no. 1. Figure 23 shows the pressure fluctuations in the frequency domain. A peak in the amplitudes of the pressure fluctuation coefficient at the frequency of the runner speed is clearly visible. Also an increase of the amplitude to low frequencies can be seen. The amplitudes of forces and torques have peaks at twice the runner speed and an increase for very low frequencies (see figure 24). As mentioned before a peak at the frequency of the runner speed is not possible in the measured signals of the forces and torques, because of the cut off filter at this frequency.

4.5. Operating point no. 4
In this high overload operating point a helical vortex rope exists as shown in figure 9. The pressure fluctuations in the time domain for this operating point are presented in figure 25. The values of the pressure fluctuation coefficient are in a range between -0.03 and 0.04. This is approximately 2.5 times bigger than the range of the pressure fluctuations in operating point 1. The pressure fluctuations in the time domain does not appear to be periodic like the pressure fluctuations in operating point 1 or in operating point 3. In figure 26 the pressure fluctuations in the frequency domain are depicted. The amplitudes of the pressure fluctuation coefficient show a peak near the frequency of the runner speed and at approximately the double frequency of the runner speed. This double frequency is the first harmonic of the high amplitude near runner speed. It can also be observed, that there is an increase in the amplitude in the range between 20% and 70% of the runner speed. Figure 27 shows the synchronous part and the rotating part of the pressure fluctuations. The amplitude of the rotating part of the pressure fluctuations is higher than the amplitude of the synchronous part near the frequency of the runner speed and the first higher harmonic. The synchronous part shows higher amplitudes at frequencies in the range between 20% and 70% of the runner speed and a small increase at frequencies lower than 15% of the runner speed frequency. The rotating part shows a peak at approximately the frequency of the runner speed and also an increase at frequencies in the range between 20% and 70% of the frequency of the runner speed.

Figure 25. Pressure fluctuations in the time domain. Figure 26. Pressure fluctuations in the frequency domain. Figure 27. Rotating and synchronous part of the pressure fluctuations in the frequency domain.

Figure 28 shows the radial forces and tilting torques in the frequency domain. The amplitudes of the radial forces show a peak at approximately the runner speed frequency and an increase at frequencies lower than 70% of the frequency of the runner speed. The peak near the runner speed frequency correlates well with the signals of the pressure fluctuations. The increase at lower frequencies than 70% of the frequency of the runner speed can be observed in the rotating part of the
pressure fluctuations and in the radial force but the maximum amplitudes are not at the same frequency. There are also peaks at double runner speed for the forces and torques like in all other operating points. The axial force and torque in the frequency domain are shown in figure 29. The amplitude of the axial force shows higher values at frequencies lower than 70% of the runner speed frequency. The highest values are at a frequency of approximately 30% of the runner speed frequency. This fits well to the amplitudes of the synchronous part of the pressure fluctuations.

Figure 28. Radial forces and tilting torques in the frequency domain.  
Figure 29. Axial force and torque in the frequency domain.

4.6. Additional investigated operating conditions
In figure 30 waterfall diagrams of the pressure fluctuations, the dynamic radial forces in x-direction and the dynamic axial forces in z-direction are shown. In figure 31 waterfall diagrams of the torque in x-, y- and z-direction are depicted. The black lines represent the operating points presented above. The investigated operating points in this waterfall diagram have the same specific speed $n_{11}$ like the BEP. In the waterfall diagram of the pressure fluctuations higher amplitudes at part load and overload conditions can be seen. The waterfall diagrams of the radial forces and the tilting torques also show peaks at part load operation. The radial force fluctuations and the tilting torque fluctuations at high overload have not the high amplitudes at lower frequencies which can be seen in the pressure fluctuations. In the waterfall diagrams of the axial forces and the torques an increase in the amplitudes at high overload operation can be observed. This behavior could be also seen at the investigated operation points.

Figure 30. Waterfall diagrams of the pressure fluctuations, the dynamic radial forces and the dynamic axial forces.  
Figure 31. Waterfall diagrams of the dynamic torques in x-, y- and z-direction.
5. Conclusion
For five operating points with different flow conditions, the measurement results of the dynamic pressure at the draft tube wall as well as the dynamic forces and torques were presented. Additionally, the visualization of the vortex structure in the four off-design operating points was shown. In three operating points a rotating helical vortex rope exists where two points were characterized by dominant frequencies in the pressure fluctuations. For these two operating points the pressure fluctuations were separated in a synchronous part and a rotating part. If there are dominant frequencies in the rotating part of the pressure fluctuations, peaks were also observed in the components of the radial forces. In part load operation the amplitudes of the synchronous part of the pressure fluctuations are small. Near the best efficiency point pressure fluctuations and all forces and torques have low amplitudes, as expected. In the high overload operating points higher amplitudes at lower frequencies in the pressure fluctuations occur. These higher amplitudes can also be seen in the dynamic axial force and torque. In the observed overload point no. 4 these higher amplitudes at lower frequencies can also be seen in the synchronous part of the pressure fluctuations, as well as in the axial forces and the torques.

Nomenclature

| Symbol | Meaning |
|--------|---------|
| BEP    | Best efficiency point |
| D      | Diameter or the runner [m] |
| b      | Runner height (at runner inlet) [m] |
| f      | Frequency of the fluctuation [Hz] |
| fn     | Frequency of the runner speed [Hz] |
| g      | Acceleration due to gravity [m/s²] |
| H      | Head of the turbine [m] |
| n      | Runner speed [rpm] |
| p      | Density of water |
| p~      | Normalized pressure fluctuation [-] |
| ∆ψ     | Pressure fluctuation [Pa] |
| Q      | Turbine discharge [m³/s] |
| Q₁₁    | Specific discharge [m³/s] |
| Q₁₁,BEP | Specific discharge at BEP [m³/s] |
| n₁₁    | Specific runner speed [rpm] |
| n₁₁,BEP | Specific speed at BEP [rpm] |

References

[1] Goede E, Ruprecht A, Lippold F, 2004 *On the Influence of Runner Design on the Draft Tube Vortex*, in Proceding 13. Internationales Seminar Wasserkraftanlagen, Wien, November 2004, ISBN 3-9501937-0-7

[2] Nishi M, Matsunasa M, Okateno M, Uno M and Nishitani K 1988 *Measurement of Three-Dimensional Periodic Flow in a Conical Draft Tube at Surging Condition*, ASME Fed 69 81-88

[3] Helmrich T 2007 *Simulation instationärer Wirbelstrukturen in hydraulischen Maschinen*, Doctoral Thesis, Institute of Fluid Mechanics and Hydraulic Machinery, University of Stuttgart, ISBN 978-3-9807322-9-1

[4] Nishi M, Matsunaga M, Okamoto M, Takashu K 1990 *Wall Pressure Measurements as a Diagnosis of Draft Tube Surge*, 15th IAHR Symposium on Hydraulic Machinery and Cavitation, Belgrade, Yugoslavia, September 11-14, 1990.

[5] Dörfler P, Lohmberg A, Michler W, Sick M 2003 *Investigation of Pressure Pulsation and Runner Forces in a Single-Stage Reversible Pump Turbine Model*, IAHR 11th International Meeting of the Work Group on “The Behaviour of Hydraulic Machinery under Steady Oscillatory Conditions”, Stuttgart, October 8-10, 2003.

[6] Etter S, Gummer J H, Seidel U 2000 *Measurement of the Forces on the Shaft of a Model Hydraulic Turbine and their Application on the Prototype*, 20th IAHR Symposium on Hydraulic Machinery and System, Charlotte, North Carolina, USA, August 6-9, 2000.

[7] Kirschner O 2011 *Experimentelle Untersuchung des Wirbelzopfes im geraden Saugrohr einer Modell-Pump turbine*, Doctoral Thesis, Institute of Fluid Mechanics and Hydraulic Machinery, University of Stuttgart, ISBN 978-3-9812054-1-1

[8] Schneider K, Michler W, Faigle P, Brandao J 1992 *Measurements of the Dynamic Forces and Torques Acting on the Impeller of a Model Pump-Turbine*, 16th IAHR-Symposium on Hydraulic Machinery and Cavitation, Sao Paulo, Brazil, September 14-18, 1992.

[9] Skoták A, Mikulasek J, Troubil P 2001 *Unsteady Flow in a Draft Tube with Elbow Part A – Experimental Investigation*, IAHR 10th International Meeting of the Work Group on “The Behaviour of Hydraulic Machinery under Steady Oscillatory Conditions”, Trondheim, Norway, June 26-28, 2001.