Load variation effects on the pressure fluctuations exerted on a Kaplan turbine runner

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Abstract. Introduction of intermittent electricity production systems like wind power and solar systems to electricity market together with the consumption-based electricity production resulted in numerous start/stops, load variations and off-design operation of water turbines. The hydropower systems suffer from the varying loads exerted on the stationary and rotating parts of the turbines during load variations which they are not designed for. On the other hand, investigations on part load operation of single regulated turbines, i.e., Francis and propeller, proved the formation of rotating vortex rope (RVR) in the draft tube. The RVR induces oscillating flow both in plunging and rotating modes which results in oscillating force with two different frequencies on the runner blades, bearings and other rotating parts of the turbine. The purpose of this study is to investigate the effect of transient operations on the pressure fluctuations on the runner and mechanism of the RVR formation/mitigation. Draft tube and runner blades of the Porjus U9 model, a Kaplan turbine, were equipped with pressure sensors. The model was run in off-cam mode during different load variation conditions to check the runner performance under unsteady condition. The results showed that the transients between the best efficiency point and the high load happens in a smooth way while transitions to/from the part load, where rotating vortex rope (RVR) forms in the draft tube induces high level of fluctuations with two frequencies on the runner; plunging and rotating mode of the RVR.

1. Introduction

Transient events are referred as harmful conditions for hydraulic turbines in the literature. They may lead to high pressure fluctuations on different parts of the turbine, including the rotating parts affecting the turbine lifetime [1]. Hence, transient events often account for most of the damages sustained by hydropower systems during their operation [2]. They affect the turbine lifetime both by accelerating crack propagation on the runner blades [3] and damaging the bearings. The exerted unsteady fluctuations during the transients are a challenge for the turbine designers and power plant owners since they are difficult to predict during the design stage. Furthermore, the transient characteristics of the turbines can be significantly different from one runner to another. Hydraulic turbines are subject to cyclic stresses, asymmetric forces on the runner, wear and tear during transient operations and each of them may affect the components lifetime [4]. From controlling point of view, the turbines behavior varies significantly with unpredictable loads, mainly due to the turbines complexity which are nonlinear and non-stationary multivariable systems. Consequently, the transient operation of the turbines poses challenges to the control community and the existing problems has not been completely solved yet [5]. At the same time, intermittent power generation has increased the average number of transient events that a power plant may experience during its lifetime period [6].
Most of the studies are focused on the steady state operation of the turbines and more studies on transient operation of the turbines are required as presented in the review paper prepared by Trivedi et al. [4]. Gagnon and Leonard [2] investigated runner blades deformation of two hydropower plants during load rejection. The study showed that in both cases the most fatigue damage occurs during the transition. Gagnon et al. [3] showed that the damage to a Francis runner during start-up event is significantly dependent on the start-up scheme. Houde et al., [1] directly measured the pressure fluctuations exerted on the runner blades of a propeller turbine during runaway and speed-no-load conditions. The results showed that in both cases the highest amplitudes were appeared in the signals during the transient events. The dominant frequencies were found to be in the sub-synchronous range, showing that the source of the fluctuations is associated with the draft tube instabilities. Trivedi et al. investigated the effect of load variation [6] and also start-up and shutdown sequences [7] on the pressure fluctuations exerted on the runner of a high head Francis turbine with the main focus on the rotor-stator interaction. The first paper results showed that the torque starts to oscillate with the start of the guide vane movement. An unsteady vortical flow also develops in the vaneless space during load variations resulting in large pressure difference between the guide vanes. Runner blades experience pressure fluctuations at their leading edge as well. The second paper showed that the pressure fluctuations on the runner were higher during shot down process compared to start-up. The fluctuations in the vaneless space were also higher during shot down. Simmons et al., [8] and Simmons [9] investigated the effect of start-up on the loads exerted on the bearings of a Kaplan turbine prototype. The results showed that the exerted load on the journal bearings of the turbine during start-up is higher compared with the load during steady state operation of the turbine. Jansson [10] used the same turbine to investigate the effect of start-up on the stresses exerted on the rotating parts of the turbine. The results showed presence of a wide band frequency in the torque and axial strain of the turbine main shaft during start-up process.

The hydraulic turbine used during the study is a 1:3.1 scaled model of the prototype investigated during start-up by Simmons et al, [8], Simmons [9] and Jansson [10]. The model has been the case study of different experimental and numerical studies during on-design and off-design operating point; see [11-17].

This paper deals with the effects of load acceptance and load rejection processes on the runner blades pressure of the above mentioned turbine. Different load variation schemes were investigated to estimate the exerted pressures on the runner and find the most harmful load variation processes for the turbine. The turbine was investigated in off-cam mode to trigger the rotating vortex rope (RVR) formation.

2. Experiments and experimental setup

2.1. Model specification and operating conditions

The model used during this study is a 1:3.1 scaled model of a Kaplan turbine which is located in the northern part of Sweden. The turbine consists of a penstock, spiral casing and an elbow type draft tube. The runner of the prototype has 6 blades and the spiral casing’s distributor consists of 20 guide vanes and 18 stay vanes. The model was investigated under a head of 7.5 m and at a rotational speed of 696.3 rpm. The guide vanes angle corresponding to the best efficiency point of the investigated propeller curve was found to be 26°. Guide vanes angles of 16° and 37.5° were selected as part load and high load, respectively. The turbine was investigated during all possible load variations between the three operating points. The operating parameters of the turbine before and after each load variation are presented in Table 1. Measurements were performed at Vattenfall Research and Development, Sweden. More information about the test rig can be found in [12].
Table 1. Parameters during load variations, subscripts 0 and 1 indicate the conditions before starting the load variation and after test stabilization, respectively. $t_{gen}$: time during load variation.

| Load acceptance | $\omega_{gv0}$ (deg) | $\omega_{gv1}$ (deg) | $Q_m0$ (m$^3$/s) | $Q_m1$ (m$^3$/s) | $H_{m0}$ (m) | $H_{m1}$ (m) | $P_0$ (kW) | $P_1$ (kW) | $t_{gen}$ (s) | $\omega_{gv}$ (deg/s) |
|-----------------|----------------------|----------------------|------------------|------------------|--------------|--------------|-----------|-----------|-------------|-----------------|
| Case 1          | 26.7                 | 37.5                 | 0.69             | 0.77             | 7.5          | 7.5          | 46.2      | 49.8      | 12.92       | 0.836           |
| Case 2          | 16.2                 | 26.7                 | 0.69             | 0.51             | 7.6          | 7.6          | 28.5      | 46.2      | 13.44       | 0.781           |
| Case 3          | 16                   | 37.5                 | 0.51             | 0.77             | 7.6          | 7.5          | 28.5      | 49.8      | 26.98       | 0.797           |
| Load rejection   |                      |                      |                  |                  |              |              |           |           |             |                 |
| Case 4          | 37.5                 | 26.6                 | 0.77             | 0.69             | 7.5          | 7.5          | 49.8      | 46.2      | 16.55       | 0.659           |
| Case 5          | 26.5                 | 16                   | 0.69             | 0.51             | 7.6          | 7.5          | 46.2      | 28.5      | 12.2        | 0.859           |
| Case 6          | 37.6                 | 16                   | 0.77             | 0.51             | 7.5          | 7.6          | 49.8      | 28.5      | 24.79       | 0.871           |
process of the RVR. The results showed that the main features of the flow is similar in each pair of load variations; high load to BEP and BEP to high load, high load to Part load and BEP to part load, part load to BEP and part load to high load. In the following sections, the acquired results from one of each pair are presented and discussed.

4.1. Load acceptance from BEP to high load

The load acceptance from the BEP to the high load, presented as case 1 in Table 1 starts with steady state operation of the turbine while the guide vanes angle is set to about 26.5°. After system stabilization, the guide vanes angle starts to increase linearly to 37.5°. The load variation process takes about 13 s as presented in Table 1. Since the test rig is closed loop, there is a delay in test rig head adjustments due to the head control system. Opening of the guide vanes in this case results in a sudden decrease in the turbine head in the beginning of the guide vanes movement. This is due to the fluid acceleration in the test rig circuit and pressure drop in the upper pressure tank. It takes around 130 s after the start of the load variation process before the turbine head is completely recovered.

Figure 2 illustrates the pressure variation on two pressure sensors located at the runner blades suction side and pressure side, close to the blade hub and trailing edge. The results show that during the load increase process, the pressure on the pressure side increases while it decreases on the suction side. This results in higher pressure difference on the blade, higher torque and higher output power. The results show that the transition happens in a smooth way and there is not any special phenomenon happening during the transient. Similar results were captured by the other blade pressure sensors as well as the sensors installed on the draft tube cone. The standard deviation of the signals increased on both blade sides during the guide vanes opening process due to the higher flow rate, higher Reynolds number and thus increased turbulence in high load compared to BEP.

![Figure 2.](image)

As discussed in the data analysis section, the fluctuating part of the signals is used to obtain the spectrogram. The spectrograms of four different sensors located on the pressure and suction sides of the runner blades close to the hub and tip are presented in Figure 3. For all cases, the runner frequency has the largest amplitude before the start of the load variation process. The other distinct frequencies in the spectrograms are the harmonics of the runner frequency. The amplitude of the runner frequency increases with the guide vanes opening because of the higher energy level at the high load compared to the BEP. The noise in the signal increases because of the increased turbulence level as mentioned before. The transition between both operating points happens without the occurrence of any special phenomenon; i.e., there is not any frequency appearing or mitigating during the load variation. The acquired results during the load variation from the high load to the BEP were similar; see Amiri et al, [18].
Figure 3. Spectrograms of the pressure sensors during load acceptance from the BEP to the high load operating point. The black curve represents the variation of the guide vanes angle. The scale of the power spectral density is logarithmic and slightly different for the figures.

4.2. Load rejection from high load to part load

Investigation of the load rejection process from the high load to the part load starts with steady state operation of the turbine at high load followed by a linear guide vanes angle change from 37.5º to 16º; see case 4 in Table 1. The turbine parameters are presented in Table 1 before and after the guide vane movement. The pressure signals from the sensors located on the blades suction and pressure sides close to the blade hub and the trailing edge are presented in Figure 4. Figures 4a and 4b present the raw and smoothed signals from the two sensors. The dashed red and green lines indicate the time of appearance of the plunging and rotating mode of the RVR on the sensors, respectively. The two modes and their appearance in the signal will be discussed in the following paragraphs. In the beginning of the load rejection process, the pressure on the pressure side starts to decrease and starts to increase on the suction side. The swirl at the runner inlet increases with the guide vanes closure; however, the runner blades are unable to extract all the energy from the flow. The combination of the increased swirl and decreased flow rate results in the formation of a RVR in the draft tube during the load variation. As the RVR forms in the draft tube, the pressure on the sensors close to the hub and trailing edge drops suddenly and then it starts to increase again. This effect was just found on the suction side for the sensors close to the hub. This results in a sudden pressure difference between the two sides of the runner; see Figure 4c. This may result in a sudden change in the turbine’s torque and an axial lift affecting the forces on the thrust bearing. Following the RVR formation, the pressure signals standard deviation increases suddenly, especially on the suction side sensors which are located in the vicinity of the RVR; see Figure 4d.
a) pressure side, sensor PS6

b) suction side, sensor SS6

c) PS6-SS6
d) standard deviation

Figure 4. Pressure development on the runner blade surfaces during load rejection from high load to part load together with the variation in the standard deviation of the results. Blue dot: instantaneous pressure, black line: smoothed pressure and green line: guide vanes angle; dashed red: start of the formation of the RVR plunging mode; dashed green: start of the formation of the RVR rotating mode

Spectrograms of the signals from different pressure sensors during the load variation process are presented in Figure 5. Before the start of the guide vanes movement, the only significant frequencies in the spectrograms are the runner frequencies and its harmonics. The amplitude of the runner frequencies start to decrease with the guide vane movement in all the sensors. Decreasing the guide vanes angle results in a vortex breakdown in the draft tube with the formation of the RVR. The RVR induces pressure fluctuations in the circumferential and axial directions; referred as rotating and plunging mode, respectively [14]. Figure 5 shows that the plunging mode appears in the signal around two seconds before the rotating mode appearance in the signal independent of the sensors location.

After the RVR formation, the RVR frequency increases with the guide vanes closure; see Amiri et al, [18]. The RVR amplitude in both the rotating and plunging modes increases as well. The rotating mode of the RVR is the dominant frequency at the end of the load variation process. Further closure of the guide vanes induces a wide band noise ranging from zero to around 500 Hz on the sensors located on the blades suction side close to the hub. This is explained by the increase in the dead zone diameter below the runner with the guide vanes closure. At a guide vane angle, the RVR radius is equal to the sensors located close to the hub and the RVR covers the sensors in this location. The wide band noise indicates a high level of fluctuations exerted to the runner by the RVR. The fluctuations may result in the resonance and eventual failure of some machine components if their natural frequencies lie within the frequency band.

Power spectral density (PSD) of the signals acquired from the sensors located on the draft tube cone near the RVR frequency is presented in Figure 6 to further investigate the time shift between the rotating and plunging modes. For the sensors located on the inner part of the draft tube, the PSD of the RVR frequency is close to zero up to $t=22.5$ s. Then, the amplitude of the RVR frequency on sensor I-4, installed close to the draft tube cone outlet, starts to increase. The amplitude sequentially increases from bottom to top for all sensors, indicating that the RVR formation starts from the bottom and travels up with the guide vanes closure. The swirl leaving the runner is responsible for the RVR formation. As the flux of angular momentum is conserved along the draft tube cone and the axial velocity decreases in the streamwise direction due to the flow expansion, the swirl increases from top to the bottom. Hence, the RVR formation is initiated at the bottom of the draft tube cone at around $t=22.5$ s. As the RVR location is far downstream of the runner at this moment, the rotating mode
cannot be captured by the sensors on the runner; perhaps due to the supercriticality of the rotating mode. Hence, only the plunging component is detected in the spectrograms. The RVR grows upstream with the guide vanes closing as presented in Figure 6. The RVR reaches to the position I-1 on top of the draft tube cone and the rotating component of the RVR appears on the runner pressure sensors signals at the same time. However, the PSD of the sensors installed on the outer part of the draft tube starts to grow up in a random way; that doesn’t start from the bottom to the top. It could be due to the effect of the bend resulting in a back pressure in the outer part of the draft tube cone. The load rejection results from the BEP to the part load operating condition showed similar features.

a) suction side-hub sensor (SS5)  
b) suction side-tip sensor (SS1)  
c) pressure side-hub sensor (PS6)  
d) pressure side-tip sensor (PS3)

Figure 5. Spectrogram of the pressure sensors during load rejection from high load to part load. The black curve represents the variation of the guide vanes angle. The scale of the pressure amplitude is logarithmic and slightly different for the figures.

Figure 6. RVR amplitude growth during load variation from high load to part load. Caption is in accordance with Figure 1.

4.3. Load acceptance from part load to BEP

The mitigation process of the RVR was investigated during load acceptance from the part load operating point to the BEP. The signals of the sensors located close to the hub and trailing edge of the blades are presented in Figure 7. The results show that the pressure on the pressure side increases and it decreases on the suction side during the load acceptance process. The larger pressure difference between the pressure and suction side results in a higher torque and thus a higher output power at the BEP compared to the part load operating point. The standard deviation of the pressure fluctuations decreased during the load acceptance process because of the RVR mitigation. The RVR mitigation
happens in a smooth way and the sudden change in the pressure of the sensors located on the suction side and close to the hub cannot be seen as during its formation.

c) Pressure side, sensor PS6

b) Suction side, sensor SS6

Figure 7. Pressure development on the runner blade surfaces during load acceptance from the part load operating point to the BEP. Blue dot: instantaneous pressure, black line: smoothed pressure, green line: guide vane angle

The spectrograms of the fluctuating part of the pressure signals are presented in Figure 8. The spectrograms are similar to the previous case, but in a reverse order. The runner frequency and its harmonics are present in the spectrograms during the steady state operation of the turbine at the part load operating point. However, the rotating and plunging frequencies of the RVR dominate the spectrogram. A wide band noise is present in the signals of the sensors located on the suction side and close to the hub. The RVR frequency decreases with the guide vanes opening and the wide band noise disappears. The RVR frequency of the plunging mode decreases, while it increases for the rotating mode with the guide vane opening; see Amiri [15]. The RVR amplitude decreases during the load acceptance process as well. The plunging mode of the RVR is more persistent than the rotating mode like in the previous case. The plunging mode mitigates around 3 s after the rotating mode disappearance.

a) Suction side-hub sensor (SS5)

b) Suction side-tip sensor (SS1)

c) Pressure side-hub sensor (PS6)

d) Pressure side-tip sensor (PS3)

Figure 8. Spectrogram of the pressure sensors during load acceptance from part load to BEP.

Variation in the power spectral density of the RVR frequency of the pressure sensors located on the draft tube cone is presented in Figure 9. The power spectral density decreases during the load variation process. The RVR amplitude approaches to zero in the sensors signal in a sequence from top to bottom. This indicates that the RVR mitigation may happen from the draft tube top to its bottom. The opening of the guide vanes results in a reduction of the RVR radius in the beginning of the process. Further opening results in a detachment of the RVR from the runner and its traveling downstream.
This is opposite to the RVR formation, which is in agreement with the discussion of the RVR formation concept in section 4.2. The mitigation of the PSD value on the outer part of the draft tube happens in a random order in this case as well. Similar results were captured during the load acceptance from the part load operating point to the high load operating point; see [18].

![Figure 9. Variation of the RVR frequency amplitude during load acceptance from the part load operating point to the BEP.](image)

### 5. Conclusion
The effect of the load variations on the pressure fluctuations exerted on the rotating parts and the draft tube of a Kaplan turbine model was investigated. Different load acceptance and load rejection processes were performed while the turbine operated under off-cam mode. The results showed that the transitions between the high load operating point and the BEP happen in a smooth manner. The distinctive phenomenon happening during a load rejection to the part load operating point was the RVR formation in the draft tube. Its formation starts with a sudden change in the pressure on the suction side of the blade, which may result in a change in the thrust bearings load and output torque. The standard deviation of the pressure signals increases suddenly after the RVR formation. The RVR results in induction of two dominant frequencies on the runner; the rotating and plunging mode frequencies. The rotating component showed to be the dominant frequency during the part load operation. The plunging mode frequency appeared on the signals a couple of seconds ahead of the rotating mode. This is explained by the formation of the RVR at the end of the draft tube cone where the swirl is the largest. The RVR travels upstream with the guide vanes closure. The amplitude of the RVR in the plunging and rotating mode increases with the guide vane closure. The RVR mitigation process showed to be similar but in the reverse order. The only difference was in the smooth disappearance of the RVR during the load acceptance process compared to the RVR formation. It can be concluded that the load rejection to the part load seems to be the most harmful load variation process for the turbines runner blades while the load variation in a RVR-free region is comparably safe.

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### Nomenclature

- $\gamma$: Non-dimensional frequency with respect to runner frequency
- $H_m$: Head
- $P$: Non-dimensional Pressure
- $P(W)$: Output power
- PSD: Power spectral density
- $Q_m(m^3/s)$: Flow rate through model
- $t$: Time
- $t_{\text{gen}}$: Load variation time
- $\alpha_{gv}$: Guide vanes angle
- $\sigma$: Standard deviation
- $\omega_{g}$: Average angular velocity of the guide vanes
7. References
[1] Houde S Fraser R Ciocan G Deschénes C 2012 Experimental study of the pressure fluctuations on propeller turbine runner blades: part 2, transient conditions Earth and Environmental Science 15
[2] Gagnon M Leonard F 2013 Transient response and life assessment: Case studies on the load rejection of two hydroelectric turbines [3] Gagnon M Tahan SA Bocher P Thibault D 2010 Impact of startup scheme on Francis runner life expectancy Earth and Environmental Science 12
[4] Trivedi C Gandhi BK Cervantes MJ 2012 Effect of transients on Francis turbine runner life: a review Journal of Hydraulic Research 51 121-132.
[5] Kishor N Saini RP Singh SP 2007 A review on hydropower plant models and control Renewable and Sustainable Energy Reviews 11 776-96.
[6] Chirag T Cervantes MJ Bhupendrakumar G Dahlhaug OG 2014 Pressure measurements on a high-head Francis turbine during load acceptance and rejection Journal of Hydraulic Research
[7] Trivedi C Cervantes MJ Gandhi BK Dahlhaug OG 2014 Experimental investigations of transient pressure variations in a high head model Francis turbine during start-up and shutdown Journal of Hydrodynamics
[8] Simmons GF Aidanpää J Cervantes MJ Glavatskih S 2013 Operational transients in the guide bearings of a 10 MW Kaplan turbine International journal on hydropower and dams 20 94-100.
[9] Gregory F Simmons 2013 Journal Bearing Design, Lubrication and Operation for Enhanced Performance PhD Thesis, Luleå Univ. of Tech.
[10] Jansson I 2013 Vibrant bodies of swirling flow: On the limits of mechanical power transformation PhD Thesis, Luleå Univ. of Tech.
[11] Mulu B Cervantes M. LDA 2010 Measurements in a Kaplan Spiral Casing Model 13th Symposium on Transport Phenomena and Dynamics of Rotating Machinery
[12] Mulu BG Jonsson PP Cervantes MJ 2012 Experimental investigation of a Kaplan draft tube – Part I: Best efficiency point Applied Energy 93 695-706.
[13] Jonsson PP Mulu BG Cervantes MJ 2012 Experimental investigation of a Kaplan draft tube – Part II: Off-design conditions Applied Energy 94 71-83.
[14] Amiri K Cervantes MJ Mulu B 2013(submitted) Experimental investigation of the hydraulic loads on a Kaplan turbine runner model and corresponding prototype J. of Hydraulic Research
[15] Amiri K 2014 An Experimental Investigation of flow in a Kaplan runner: steady-state and transient Licenciate Thesis, Luleå Univ. of Tech.
[16] Mulu B 2012 An experimental and Numerical Investigation of a Kaplan Turbine Model PhD Thesis, Luleå Univ. of Tech.
[17] Petit O Mulu BG Nilsson O Cervantes MJ 2010 Comparison of numerical and experimental results of the flow in the U9 Kaplan turbine model Earth and Environmental Science 12
[18] Amiri K Cervantes MJ Mulu BG Raisee M 2013(submitted) Unsteady pressure measurements on the runner of a Kaplan turbine during load acceptance and load rejection Journal of Hydraulic Research