Shear and vortex instabilities at deep part load of hydraulic turbines and their numerical prediction

B Nennemann¹, M Melot¹, C Monette¹, M Gauthier¹, S Afara¹, J Chamberland-Lauzon¹, and T Jurvansuu²

¹Andritz Hydro Canada Inc., Montreal, Canada
²Andritz Hydro Finland, Tampere, Finland

bernd.nennemann@andritz.com

Abstract. For all types of turbines, deep part load operation (DPL) poses a challenge. An example of a dynamic phenomenon due to vaneless space vortices occurring at DPL is presented for three turbine types: a diagonal, a propeller and a Francis turbine. The backflow region occurring at DPL is often considered to be a main factor in these phenomena. Our results confirm that these backflow regions play an important role, but other factors also seem to be significant in specific cases. In our example of a diagonal turbine, the intersection of the backflow region with the leading edge of the blades seems to generate particularly high pressure pulsations and vibrations. In the case of the propeller turbine, large vaneless space vortices are found in CFD, but vibrations on the prototype machine are well within acceptable levels. Inspecting the flow at an operating point before large vaneless space vortices occur, shows high shear levels near the inner head cover that generate the high-intensity vortices at an even lower load. In the Francis turbine, half the number of strong vaneless space vortices interact with equally strong inter-blade vortices with vorticity of opposite sign to result in high dynamic blade torque. CFD simulations are well capable of capturing these phenomena, allowing them to be considered in the mechanical design of the turbine components for safe operation.

1. Introduction

Hydraulic turbines are increasingly operated over the full operating range from zero to 100% power. Consequently, deep part load (DPL) operation is becoming an important topic for operators and manufacturers. The hydraulics are characterized by an imbalance between incoming angular momentum, flow rate and the extraction of energy in the form of shaft torque. For all turbines, such operating conditions pose a challenge. Kaplan turbines are less affected because they have an additional degree of control with their variable blade angle, but DPL conditions with coherent vortex structures can occur in sluicing (no-load operation to pass a given flow rate for environmental reasons) and during off-cam operation. Therefore, all types of hydraulic reaction turbines may operate at deep part load in a condition with a high imbalance of incoming and extracted hydraulic energy. These are often associated with high pressure pulsations and sometimes vibrations. If machines are to be operated over extended periods of time in these operating conditions, it is important to have a good understanding of their hydrodynamics, particularly of the origin and nature of the vortices that can lead to strong pressure pulsations and vibrations.

Seidel et al. [1] include part load below the vortex rope and above speed-no-load (SNL) in their analysis of causes for stresses on turbine runners. Magnoli et al. [2] and Yamamoto et al. [3] provide
studies on inter blade vortices (IBV). IBV occur below the part load vortex rope zone and persist into deep part load. These studies therefore provide good insight into these primary flow phenomena occurring inside the runner at deep part load. Pulpitel et al. [4] present a detailed experimental and analytical study of vortices occurring in the vaneless space of Kaplan turbines. They used air injection to visualize different types of vaneless space vortices, including columnar vortices, during model tests. The motivation of their study came from strong vibrations occurring on prototype Kaplan turbines under sluicing operation. Skotak [5] uses experiments and numerical modeling to show that vaneless space vortices in a Kaplan turbine originate at the interface between the backflow region and the main flow at deep part load and runaway. Houde et al. [6] investigate no-load conditions on the AxialT turbine through experimentally validated numerical analyses. They find a large recirculating region in the center of the machine extending from the inner head cover into the draft tube. Vortices attached to the inner head cover form in the radial transition between the main flow and the backflow regions. This occurs even in the absence of the runner indicating that the flow topology is largely independent of the runner.

Our study addresses recent observations of hydraulic phenomena at DPL with a focus on coherent vortex structures, which can induce high pressure pulsations and potentially shaft line vibrations. This is done by means of measurements and computational fluid dynamics (CFD) simulations. Three types of hydraulic turbines are investigated.

2. Theoretical considerations

Since most DPL phenomena mentioned in the literature are related to large backflow regions, we will present some simplified analytical considerations that can guide our understanding of their cause. Strscheletzky [8] developed two analytical models of flow with a swirling and a throughflow component in a cylinder. Hydrodynamic equilibrium imposes a radial division of the flow into a main flow region, with swirl and through flow, and a so-called “dead water zone” in the center of the cylinder. Strscheletzky provides two equations for the theoretical inner radius \( r_i \), which separates the two flow regions. His analytical derivation demonstrates mathematically what one can easily understand qualitatively, namely that centrifugal forces in swirling flow should move flow particles to the periphery of any enclosure. We only use his second equation for axially limited swirl in our comparisons since this gives better comparisons than his first equation of axially unlimited swirl. His equations are not given here.

We derived a different, quite simple estimate of the inner radius \( r_i \) based on the continuity equation of what we call the circumferential flow rate \( Q_{1u} \) in the radial space downstream of the guide vanes, assuming a free vortex circumferential velocity distribution \( c_u(r) \). The circumferential flow rate is defined as (nomenclature see figure 1)

\[
Q_{1u} = B_0 r_1 c_{1u} \int_{r_i}^{r_1} \frac{1}{r} dr = B_0 r_1 c_{1u} \ln \frac{r_1}{r_i} \tag{1}
\]

But the circumferential flow rate is also related to the guide vane opening area \( A_1(\alpha) \)

\[
Q_{1u} = c_{1u} k_c A_1(\alpha) \tag{2}
\]

The contraction factor \( k_c \) is the ratio of actual flow rate over theoretical flow rate from pressure difference and cross section area \( A_1(\alpha) \) at the guide vane opening \( \alpha \). Then we get an estimate of the inner radius from

![Figure 1. (a) Cross section of a hydraulic turbine (b) simplified control volume for formulation of analytical equations](image-url)
\[ r_i = r_1 \exp \left( -k_c A_1(\alpha)/(B_0 r_1) \right) \]  

### 3. Computational Fluid dynamics

Throughout our paper we show CFD results of computations in the DPL range. A standard setup is used for all computations. Details on the general setup can be found in [9]. CFX is still the flow solver used by Andritz for these types of computations in combination with the SAS turbulence model. Since publication [9] in 2014, mesh sizes have evolved, with 20M for the total number of mesh nodes now being standard. This type of CFD computation has proven reliable at providing pressure loads as input for mechanical analysis of turbine runners [7] as well as capturing the primary large-scale hydraulic features present in part load and no-load operation. It is therefore considered a suitable tool for the investigation of hydraulic features in these operating conditions.

### 4. Axial turbines

Pulpitel [4], Skotak [5] and Houde [6] all present studies of DPL and NL operation on Kaplan turbines, which clearly indicate that these operating conditions are relevant for axial turbines with respect to pressure pulsations and vibrations. Strong pressure pulsations and especially shaft line vibrations are also known to Andritz engineers in such challenging operating conditions often outside the normal operating range.

#### 4.1. Diagonal runner

During commissioning of one of Andritz’ diagonal runners, strong shaft line vibrations occurred near 30% of rated power. This was well outside the operating range of the machine, and therefore was of no concern for the project. Nevertheless, Andritz launched an investigation to uncover the root cause of these vibrations. Figure 2(a) shows how a slight drift in guide vane opening of about 0.5% results in an equally slight drift in power with a striking increase in shaft line vibration as measured by the turbine guide bearing proximeters. An image from the CFD simulation in figure 2(b) shows the three-dimensional topology of the flow. The vortex structures are highly dynamic and somewhat irregular as they move around the runner. The vortices paths are altered by the blade leading edges near the inner head cover, forcing their attachment point to a larger radius. This results in a change in rotation direction of the vortices about their own axes (not seen on the image, but identifiable in the corresponding movie).

Through conservation of angular momentum this changes the vortices’ vorticity. The phenomenon resembles Rossby waves, which are well known in atmospheric science.

**Figure 2.** (a)Time signals of high-vibration operating condition: Power, guide vane opening and turbine guide bearing proximeter. (b) Projected vorticity contour plot near hub and inner head cover as well as iso-surface of low-pressure regions in vortex cores from CFD simulation.
The power spectrum density of the turbine guide bearing proximeter’s signals (made complex by combining time signals 90° apart) in figure 3 shows the main frequency content of two different operating conditions. One has a positive sense of rotation and the other a negative. CFD and frequency analysis revealed that the former is associated with three vaneless space vortices while the latter is associated with four. The analysis presented by Pulpitel [4] explains the measured frequencies and their sense of rotation (propagation around the turbine axis), as well the fact that the shaft line is excited. This explanation can now be related to a well-known phenomenon in hydraulic turbines: rotor-stator interaction (RSI) [10]. A seven-bladed runner $Z_r = 7$ with 3 vortex structures $Z_v = 3$ gives $\pm k = mZ_r - nZ_v = 1 \cdot 7 - 2 \cdot 3 = 1$, i.e. $k = 1$ with a positive rotation exciting the forward whirling of the shaft line. For us $Z_r$ is the number of runner blades and $Z_v$ the number of vortex structures with which the runner interacts, rather than $Z_g$ the number of gates as in RSI. At the second operating condition shown in figure 3, 4 vortex structures are present and give $k = 1 \cdot 7 - 2 \cdot 4 = -1$, exciting the backward whirling of the shaft line.

As shown in figure 4(b) the intersection of the guide vane opening of the unstable condition (orange dashed line) and the runner leading edge radius on the inner head cover (green dashed line) coincide with the approximate inner radius of the main flow region (outer radius of the back flow region) determined by CFD. The range of inner radius calculated by equation (3) corresponds reasonably well with the one found by CFD around the critical guide vane opening, but the general trend deviates. Strscheletzky’s inner radius is far from the one found in CFD, indicating that his model, formulated for a purely cylindrical configuration, is not applicable to this geometry and flow.
4.2. Propeller runner

In the recent development of a propeller runner replacement, an in-depth CFD analysis of SNL and DPL operating conditions was performed. Measurements of shaft vibrations by means of turbine guide bearing displacements were available for the existing runner. The part load portion of a turbine guide bearing displacement sensor is shown in figure 5(a) and its power spectrum in figure 5(b). In this turbine, vibration levels at part load are within a reasonable range although the flow is highly unsteady and irregular.

![Image of turbine guide bearing displacement](a)

**Figure 5.** Propeller turbine guide bearing displacement (a) time signal with power ramp up over low power range (b) Power spectrum of SNL operation

The backflow region at SNL shown in figure 6(a) extends to well upstream of the blades. In figure 6(b) the theoretical inner radii of the backflow zone are shown for comparison. Since no highly unstable operation is present, SNL is indicated as a reference. Strzscheletzky’s theory is – as for the diagonal turbine – not applicable. The approximate inner radius from CFD (red curve) has a different tendency than the theoretical curve according to equation (3). At SNL the two are quite close.

![Image of backflow zone](a)

**Figure 6.** Propeller turbine (a) Backflow zone visualized by an iso-surface of 0.1m/s upward axial flow velocity at 26% guide vane opening (b) Theoretical inner radius $r_i$ of main flow compared to CFD, position of blade leading edge crown.

Figure 7 shows that at SNL complex and somewhat irregular vortex structures are present. Two main vortex structures can be discerned, forming upstream of the runner near the interface between the backflow region and the main flow. They continue along this interface through the runner into the downstream draft tube where they form two vortex rope filaments. In the upstream part (figure 7(a)) a circumferential vortex as reported in [4] is present.
The knowledge of the flow topology from CFD can help us in interpreting the power spectrum of the measured displacement in figure 5(b). Apart from the obvious runner rotation peak at $f_n$ there is a group of frequency peaks below runner rotation as well as one with strong amplitudes around $4f_n$. These two frequency groups can be explained as follows: the frequency peaks below $f_n$ can be linked to coherent vortex structures in and around the runner that rotate around the machine axis with the frequency that is measured. For these frequencies, the best candidates are the irregular strands of the draft tube vortex rope that we can identify in CFD (figure 7). Their frequency of rotation corresponds approximately to the measured displacement frequencies below $f_n$. Their eccentric pressure field will result in a radial force. The group of peaks around $4f_n$ can then be explained by an interaction of the 5-bladed runner with the vortex rope filaments. As explained above for the diagonal runner, this is equivalent to a rotor-stator interaction (RSI), and the theories by Tanaka [10] or Pulpitel [4] apply. The shaft line radial displacement only sees nodal diameter $k = \pm 1$ excitations with $\pm k = mZ_r - nZ_v$. The two factors $m$ and $n$ are arbitrary integers. In standard guide vane-runner blade interaction, the frequency on the stator is calculated by $f_s = (nZ_v \pm k)f_n$. In our case these formulae apply for an observer located on the vortex structures and for whom these vortex structures represent the equivalent of stationary guide vanes. We then calculate $f_{sv} = (nZ_v \pm k)(f_n - f_s)$ where the subscript $v$ stands for “vortex”. We can finally obtain the frequency as seen from a stationary reference frame $f_s = (nZ_v \pm k)(f_n - f_s) + f_s$. This is the vortex runner interaction frequency that a proximeter should see. Using as an example the second peak in figure 5(b) with $\sim 0.24f_n$, the frequency for interaction with the runner becomes $f_s = (nZ_v \pm k)(f_n - f_s) + f_s = (4 \cdot 1 + 1) \cdot (f_n - 0.24f_n) + 0.24f_n = 4.04f_n$. The next higher frequency $\sim 0.48f_n$ may well be from the partner vortex rope, i.e. $2f_0$ (in CFD we see two rope filaments of unequal size). RSI again projects this frequency into the $4f_n$ range.

The radial force on the runner calculated by CFD in figure 8 shows the same tendency. There are two peaks below runner rotation frequency associated with the twin vortex structures and ropes. These are projected to a frequency above runner rotation at just below $4f_n$. There is a slight frequency offset compared to the measurements, but the principal of RSI-projected vortex frequencies is well predicted by CFD.

The velocity profiles from steady state CFD upstream of the runner shown in figure 9 shed some light on the cause of the vortex formation. The circumferential velocity profile $c_u$ at 20° guide vane opening resembles a free vortex velocity profile with high values near the centre and consequently a high velocity gradient near the inner head cover wall. The meridional velocity component shown on the right-hand side of that figure exhibits a curious local meridional velocity overshoot very close to the inner head cover. In fact, the high meridional velocities are within the boundary layer of the circumferential velocity. The strong velocity gradients near the inner head cover generate flow element stretching in two directions and therefore result in three-dimensional vorticity generation. Eventually,
the velocity gradients become unstable and break down into large-scale vortices. At SNL (12° guide vane opening) the velocity profiles show that the vortices redistribute the flow such that the velocity gradients are reduced, and the average boundary layer thickness is increased.

**Figure 8.** Shaft radial force from CFD: left time signal and right power spectrum for two operating points, SNL and 15% power.

![Shaft radial force from CFD](image)

**Figure 9.** Normalized velocity profiles from unsteady CFD at two guide vane openings along the magenta line: circumferential component (left) and meridional component (right).

Looking at the flow quantity distributions in the vaneless space of an operating point above SNL in DPL in figure 10, reveals some interesting aspects of the flow. The free-vortex-like velocity distribution exists in the entire vaneless space. The high axial flow velocity is limited to a narrow band near the inside part of the inner head cover (at small radii). The pressure distribution is typical for a free-vortex-like velocity distribution with a pressure decrease towards the centre. This pressure distribution explains the high axial flow velocity near the inner head cover: a pressure equalization flow, i.e. secondary flow, driven by the pressure gradient along the inner head cover. As is typical for such flows, they form in the boundary layer (much like the well-known rotating teacup flow, which accumulates the tea leaves in the centre at the bottom of a teacup). The high flow velocities in circumferential and axial direction create the high shear (velocity curl) near the inner head cover.

**Figure 10.** Various instantaneous flow quantities on an r-z plane (grey plane figure 9) from unsteady CFD at a DPL operating condition with three vortices, $\alpha = 20^\circ$. 

![Various instantaneous flow quantities](image)
5. Francis turbines

Speed-no-load (SNL) has been studied as a reference operating condition on Francis turbines in the past few years, e.g. [1][7][9][11][12]. At SNL the flow is typically highly stochastic with little coherence detectable in the measurements. However, strain gauge measurements on Francis turbine prototypes show that the largest dynamic stresses are often at DPL rather than SNL, e.g. [1][7][12]. Some coherence can be identified in the stress measurements [7] and the comparatively higher stress levels have been attributed to inter blade vortices (IBV). To our knowledge, no detailed studies of the hydrodynamic causes of these higher stresses on Francis runners at DPL have been published up until now. In Andritz’ process for fatigue life analysis of Francis runners, DPL operation is included. The same process as at SNL is applied and the increased stresses at DPL are typically predicted.

In the CFD analysis for a recent runner design, a dynamic blade torque about two times higher than at SNL occurred at about 17% rated power, see figure 11(a). In the same figure the z-vorticity field (rotating reference frame) near mid-span of the distributor is shown for three operating conditions.

![Francis runner](image)

**Figure 11.** Francis runner (a) dynamic blade torque in range from SNL to 40% rated power. (b)-(d) instantaneous z-vorticity in rotating frame on plane at about mid-distributor height corresponding to the three operating conditions indicated by the red circles on the graph in (a). (b) SNL, (c) ~11% power, (d) 30% power. Warm colours positive vorticity and cold colours negative vorticity.

At SNL, figure 11(b), small distinct areas of high positive vorticity (red) due to IBV are discernable, but in general the flow inside the runner channels gives an impression of irregularity. Seven regions of high, relatively evenly distributed negative vorticity can be identified just to the inside of the guide vanes. These appear to have largely dissipated before reaching the runner channels.

At 30% power, figure 11(d), large IBV are present in most runner channels. In the vaneless space, five large zones of moderate negative vorticity (light blue) are present. Higher intensity negative vorticity is present at the entrance to some runner channels, unevenly distributed on the circumference. This negative vorticity seems to have relatively little effect on the strong IBV of positive vorticity inside the channels.
At about 11% power, near the maximum of dynamic blade torque, the vorticity distribution in figure 11(c) shows some interesting features. Six zones of strong negative vorticity (dark blue) are evenly distributed around the vaneless space. Since we have 13 runner blades, these zones of negative vorticity enter about every second runner channel. Having the opposite sign and approximately equal magnitude of the channel vortices, the vaneless space vortices disturb and partially dissolve the channel vortices. Since this happens in about every second runner channel, a blade is exposed to a different flow topology from its pressure to its suction side. This results temporarily in a high pressure differential and therefore high dynamic blade torque, occurring over the cycle of one runner channel passing one vaneless space vortex. The near 180° phase shift in the time signal of two neighboring blades is clearly visible in the dynamic blade torque time signals of figure 12. The spectrum shows the dominant frequency associated with this phenomenon.

![Figure 12](image)

**Figure 12.** Time signal and spectrum of dynamic blade torque from unsteady CFD

Figure 13(a) shows the large backflow zone that is present at the condition with the highest dynamic blade torque. The intersection of the runner leading edge radius on the runner crown with the guide vane opening of the largest dynamic blade torque (two dashed lines) almost coincides with Strscheletzyk’s inner radius (blue triangle). His theory seems to be applicable for this example. The approximate outer radius of the backflow region in CFD based on the main backflow area (CFD 1, red) lies below the intersection of the two dashed lines. If the “spill over” region (green circle in figure 13(a)) is considered, the approximate outer radius from CFD (CFD 2, green) is slightly above. This illustrates the subjectivity of determining a precise radius of the backflow region based on the CFD results. The range of inner radius calculated by equation (3) is fairly centred in the CFD inner radius range.

![Figure 13](image)

**Figure 13.** Francis runner (a) Backflow region at max dynamic blade torque based on iso-surface of upward 0.1m/s axial flow velocity (blades clipped) (b) Theoretical inner radius \( r_i \) of main flow compared to CFD, critical \( \alpha \), i.e. with highest dynamic blade torque.
6. Conclusions
Vaneless space vortices can result in highly dynamic conditions in all types of hydraulic turbines. In some cases, these vortices can be associated with large backflow regions extending from downstream to upstream of the runner. The shear layer between these backflow regions and the main flow are the source of vortex formation as seen on the diagonal runner. Inspecting the flow in the vaneless space of the propeller turbine at an operating condition before large vaneless space vortices occur, show strong shear layers near the inner head cover. Reducing the flow increases the shear until the flow becomes unstable and breaks down into distinct vortex regions. In our Francis turbine example, strong shear in the vaneless space generates distinct zones of high vorticity, which interfere with the vorticity of the channel vortices of opposite sign creating highly dynamic conditions. The inner radius of the main flow calculated based on continuity considerations compared to the runner leading edge at the crown or inner head cover, can give a rough indication of the guide vane opening range where highly dynamic conditions may be expected. However, more experience must be gathered with this parameter, before definitive conclusions can be drawn. CFD in conjunction with measurements has proven to be a useful tool in predicting many of the hydrodynamic aspects of deep part load operation. More interesting hydrodynamic phenomena are likely waiting to be discovered. Since CFD simulations are well capable of capturing these phenomena, they can be considered in the mechanical design process. Consequently, turbine components can be designed for safe operation even at DPL.

Acknowledgments
Many of our colleagues have contributed to this work through discussions, ideas, and otherwise. Our sincere thanks go out to them. Special thanks go to our PhD student Mélissa Fortin, who has significantly contributed to our understanding of flow in no-load conditions through her many ideas and the rediscovery of relevant papers from the past that had been hidden unused in library archives.

References
[1] Seidel U Mende C Hübnern B Weber W Otto A 2014 Dynamic loads in Francis runners and their impact on fatigue life 27th IAHR Symposium on Hydraulic Machinery and Systems
[2] Magnoli M V et al IOP Conf. Ser.: Earth Environ. Sci. 240 022044
[3] Yamamoto K et al 2017 J. Phys.: Conf. Ser. 813 012029
[4] Půlpitel L Skoták A and Koutnik J 1996 Vortices rotating in the vaneless space of a Kaplan turbine operating under off-cam high swirl flow conditions Hydraulic Machinery and Cavitation, E. Cabrera, V. Espert, and F. Martínez, eds., Springer Netherlands, pp. 925–934
[5] Skoták A 1996 Modelling of the swirl flow in a Kaplan turbine operating under off-cam conditions Aqua-Media International Lausanne Switzerland pp. 197–204
[6] Houde S et al 2018 Experimental and numerical investigations on the origins of rotating stall in a propeller turbine runner operating in no-load conditions ASME Journal of Fl. Eng.
[7] Chamberland-Lauzon J Monette C Nennemann B Melot M Birk S Ruchonnet N 2018 Francis design and prediction technology for flexible operation IAHR 29th Symposium on Hydraulic Machinery and Systems Kyoto Japan
[8] Strscheletzky, M., 1959, “Gleichgewichtsformen der rotationssymmetrischen Strömungen mit konstantem Drall in geraden, zylindrischen Rotationshohlräumen,” Voith-Forschung und Konstruktion, 5(3 S.1.1/1.19), p. 19
[9] Nennemann B et al 2014 Challenges in Dynamic Pressure and Stress Predictions at No-Load Operation in Hydraulic Turbines IOP Conf. Ser.: Earth Environ. Sci. 22 032055
[10] Tanaka H 2011 Vibration behaviour and dynamic stress of runners of very high head reversible pumpturbines International Journal of Fluid Machinery and Systems vol. 4
[11] Morissette J-F et al 2016 Stress Predictions in a Francis Turbine at No-Load Operating Regime 28th IAHR Symposium on Hydraulics Machinery and Systems Grenoble France
[12] Christine M et al 2016 Cost of Enlarged Operating Zone for an Existing Francis Runner 28th IAHR Symposium on Hydraulics Machinery and Systems Grenoble France