Quantifying the Effect of Kaplan-Type Runner Blade Gaps on Fish-related Flow Conditions

Pedro Romero-Gomez(1), Alison Colotelo(2) and Simon Weissenberger(1)
(1) ANDRITZ Hydro, Lunzerstrae 78, 4030 Linz, Austria
(2) Pacific Northwest National Laboratory, 902 Battelle Blvd, Richland, WA 99354, USA
E-mail: simon.weissenberger@andritz.com

Abstract. In double-regulated Kaplan turbines, gaps form between the rotating blade and the hub as well as the discharge ring due to both the local geometry and the operational positioning of the runner blade. It is conventionally assumed that minimizing such gaps may decrease the likelihood of grinding and exposure of fish to high shear flows. Together with blunt leading edges of runner blades, minimum gap runners (MGR) are featured as the most effective measure to ameliorate the risks of mortal injury for fish passing through the turbine. However, the merits of such premise need yet to be quantified and made publicly accessible in the technical literature. This, in turn, will facilitate the evaluation of candidate MGR geometries in future designs.

In this work, we propose a metric to quantify the influence of the gaps features on the mortality risks for fish passing the turbine runner. Two main outcomes derived from the present research: (1) It informs the minimum set of hydraulic variables that may be considered as test conditions for laboratory experimentation on fish subjected to near gap-like flows and (2) it formulates a metric to be integrated into the modelling strategies currently in use for the biological performance assessment of new turbine geometries. This physics-based examination of the flow phenomena was conducted in two Kaplan-type units of very distinct features (one is a very low-head, mid-size unit of 1.8 MW-rating capacity and another is a large-size unit of 90 MW-rating capacity).

1. Background
The compromise between hydropower generation and its environmental impacts has spurred research on the identification and quantification of detrimental circumstances causing high mortality of aquatic biota as well as on the mitigation strategies to maintain healthy river ecosystems. Bypass systems have historically been the preferred technical solution to route downstream migratory fish away from turbines in operation but they are not 100% efficient, which in turn makes fish turbine passage inevitable. A comprehensive review of various field studies of fish mortality through Kaplan turbines revealed that survival rates can be in the order of 91% and 74% for juvenile salmonids (e.g., Oncorhyncus sp.) and eels (Anguilla sp.), respectively [1]. Such empirical evidence prompted investigations on environmentally enhanced turbines—that is, on identifying and implementing geometric features of the runner and ancillary components to enhance the turbine-related survival of migratory fish [2, 3]. After their implementation, the effectiveness of such enhanced features have been put to test by means of both empirical and modelling techniques [4, 5]. Turbine design teams in industry have gradually adopted and integrated “fish friendliness” concepts into their design practices.
For instance, reducing the number of runner blades to a minimum possible can lower the likelihood of collision, thereby reducing the overall collision-related mortality. As another example, minimizing gaps between the runner blades and shroud as well as between runner blades and hub is often cited as one of the most effective strategies to reduce grinding of passing fish and their exposure to high shear flows. While the notion is well-reasoned and widely adopted in industry, its effectiveness still needs to be put under scrutiny. This study proposes a method for evaluating “fish-related” flow features derived from gaps and applies the method to two very distinctive Kaplan turbine geometries. This study is relevant because minimizing gaps introduce considerably efforts for turbine design as well as new challenges for manufacturing and installing MGRs, yet their effectiveness in reducing fish mortality remains unclear.

For Kaplan turbines, gaps form due to the construction and operational positioning of the runner blade and their size plays a role in the expected hydraulic performance of the machine (figure 1). Gaps are also known as “tip clearances” in the literature and have long been a subject of empirical and numerical studies. Gaps cause cavitation as the flow moves nearly instantly from a very high pressure region (over the blade) to the suction side of the blade [6]. In the process, absolute pressures may drop below saturated vapor pressure, thereby forming vapor-filled bubbles. Once subjected to a high-pressure environment again, these bubbles collapse and yield an intense shock wave. This in turn causes surface pitting, the prevention of which is a continuous source of testing and investigation. For fish, the gap cavitation may induce an overall drop in absolute pressure that makes low pressure-sensitive (physostomous) fish species vulnerable. Physoclistous species (largemouth bass, Micropterus salmoides, and blue guill, Lepomis macrochirus) are even more sensitive to rapid changes in pressure than salmonid species are [7]. In addition, gaps promote the appearance of flow vortices that increase pressure pulsations and high shear regions which may propagate further downstream from the gaps [8]. While pressure pulsations are detrimental for the mechanical stability of the unit, high shear regions may increase likelihood of abrasion and other injuries on passing fish [9]. Furthermore, the possibility of grinding if a passing fish gets “trapped” in gaps is also present. While the notions about the role of gap flows on fish bodily damage are rational and compelling, a quantitative analysis is still lacking. This study aims to fill such knowledge gap

![Figure 1](image1.png)

**Figure 1.** Gaps are formed due to the geometric construction of a runner blade and its operational positioning. Minimizing gaps is assumed to be an effective measure to reduce mortality of fish due to grinding and exposure to high shear flows.
by quantitatively characterising “near gap” flow features using numerical simulations with CFD techniques. Previous CFD studies on gap flow features have examined the conditions arising from both the hub clearance [10] and shroud gaps [11] in relation to hydraulic performance; at the core of this study, we re-examined such flow features in view of their potential effect on risks of mortality to passing fish. In the outline of this work, we describe the two interrogated Kaplan units (sec. 2) and the numerical method to calculate flow fields (sec. 3.1). Next, the evaluation method to characterise “gap” conditions is introduced (sec. 3.2) and the results presented (sec. 4). Finally, the potential implications of this study in the experimental design of labotatory tests with live fish is discussed (sec. 5) before some final remarks are offered (sec. 6).

2. Study cases and operating points

Two Kaplan units were examined in the present study. One of them can be categorised as a low-head, mid-size unit and the other one as a large-size machine (figure 2). Both Kaplan geometries were recently developed by an experienced turbine design team. The low-head unit was designed in the context of a rehabilitation project which required the incorporation of “fish friendly” geometric features in order to increase the likelihood of survival of eels and salmonids during turbine passage in the Rhine River corridor (Germany). The large hydro was developed as part of an internal research program which used the turbine model procured for the Ana Cua Hydroelectric Complex of the Parana River (Argentina/Paraguay) as a baseline.

With this selection of turbine geometries, we aimed to examine machines that yielded very distinct flow conditions near gaps. Because they are available in a wide range of sizes, Kaplan machines can actually give rise to a spectrum of flow conditions that is broader than the one derived from the present two cases. We assume that the analysis of a wider range of flow conditions can better inform experimentalists for the setup of laboratory apparatuses for simulating “gap-like” flows and determine the consequential mortality rates. We recognize that the present study may still miss other relevant variables or conditions that fish biologists later find to be imperative for an experimental design. Likewise, we are well aware that the logistical and financial challenges that such laboratory tests entail demand not only one but a series of preliminary studies of this kind. Our goal herein is in fact to initiate the information exchange between those whose expertise lies in the field of turbine hydraulics and fish experimentalists. The general characteristics of the two Kaplan machines are given in table 1.

Most model-based studies of fish-related properties of water turbines typically examine a few operating points that are deemed relevant for design or operation. The usual choice falls on a few points over a constant net head (e.g. rated head) within a range that reaches a low discharge at the lower 1% of peak efficiency, at peak efficiency, or at the upper 1% of peak efficiency. This study departs from such approach in that it investigates “near gap” flow conditions over the entire operating range of the interrogated turbines. This development, in turn, requires

Figure 2. The study was conducted in two Kaplan turbines of distinct characteristics, a three-bladed low-head (left) and a five-bladed large unit (right)
| Feature              | Low Head   | Large Hydro |
|---------------------|------------|-------------|
| Runner diameter     | 4.1 m      | 8.85 m      |
| Rotational speed    | 75 rpm     | 94.74 rpm   |
| Net head            | 2.82 (rated) | 18.90 (nominal) |
| Discharge           | 60 m$^3$/s (rated) | 506.8 (nominal) |
| Power output        | 1.8 MW (rated) | 88.50 MW (nominal) |
| Number of blades    | 3          | 5           |
| Number of guide vanes | 24       | 24          |

| Analysed operating range |
|--------------------------|
| Minimum head (*)         | 0.58       | 0.56        |
| Maximum head (*)         | 0.97       | 0.90        |
| Minimum discharge (*)    | 0.68       | 0.48        |
| Maximum discharge (*)    | 1.44       | 1.23        |

(*) Values are given with respect to the corresponding value at peak efficiency

Table 1. Geometric and operational characteristics of the interrogated turbine units

an intensive use of computer-based protocols for flow simulations. The limits of the operating ranges are also given in table 1.

3. Flow characterization in near gaps

3.1. Flow simulations

A computer-based tool chain constitutes the primary engine for the design team to iteratively propose and evaluate turbine geometries hydraulically. The workflow is graphically illustrated in figure 3. The turbine designers must specify the model boundary condition (pressure- or discharge-driven flow?) and prescribe the operating points to be analysed (e.g. runner blade angle, guide vane opening, rotational speed). The runner blade and guide vane geometries are an outcome from precursor software packages that compose hydraulic components of turbines based on more basic geometric features (e.g. base hydrofoil profiles, curvatures, gaps). Both the runner blade and guide vane geometries are given in a text-file format. CFD settings must be for the underlying physics models (e.g. fluid properties, turbulence model, treatment of pressure field) as well as for the numerical scheme (e.g. numerical treatment of pressure field, convection formulation for the momentum and turbulence equations, parallel scheme). Mesh settings (e.g. grid density, cell size distribution, gap meshing treatment) are necessary for the grid generation engine. Further information of the work flow for CFD simulations of turbine flows can be found in [12].

The tool chain takes all the aforementioned pieces of information and automatically generates a numerical grid of the computational domain. Then, the CFD model is set up with the commercial CFD software ANSYS CFX (ANSYS Inc., Canonsburg, PA, USA). Next, a high-performance computing protocol is generated and the simulation launched on local parallel computing clusters. After the simulation has reached convergence, the flow calculation stops and the tool change elaborates a detailed report of the hydraulic performance arising from the proposed turbine geometry (e.g. efficiency, head losses through components, torque).

More specific about the CFD flow simulations, we set up “stage” interfaces between the stationary (distributor) and the rotating (runner) regions of the turbine model, as well as between the runner and the draft tube. The runner region motion was modelled with a rotating reference frame applicable only to the runner. Both the stage interfacing and localized reference frame have historically been the modelling standard in turbomachinery applications. All solid boundaries were set as walls with zero velocity. We made use of the standard $\kappa$-$\varepsilon$ model together with a “scalable” wall treatment available in the commercial CFD software. Turbulent quantities were
advected with an “upwind” type scheme and solutions were carried out in steady state, which took between 300-500 iterations to achieve acceptable convergence. Ultimately, the most relevant calculation outcome for the present evaluation is a converged three-dimensional description of the flow fields, namely, the three velocity components, pressure and turbulence quantities (turbulent kinetic energy and its dissipation rate).

3.2. “Near-gap” flow passage examination

“Near gap” flows are fluid streams that pass the runner blades in proximity to the hub and shroud gaps. Due to the strong effect of gaps on hydraulic performance, field and laboratory studies have long examined the flow features deriving from gaps. The main assumption in this study is that two variables potentially play an important role in the consequential fish mortality: the pressure differential between the regions above and below the gap ($\Delta P$), as well as the passage velocity ($V_{gap}$). If we looked at absolute pressures at different circumferential bands of the turbine blades, we would find a large pressure differential in all radial locations. We assume, however, that only the pressure differential near the gaps will carry some fluid, and potentially fish too, from one side of the runner blade to the other. The greater the pressure differential is, the greater the “fish suction” potential through the gap is. Figure 4, for instance, shows the simulated absolute pressure at prototype scale on two streams bounding the gap region. Although we typically think that pressures above the blade are always greater than those below, they actually show a large range due to localized hydraulic conditions. For that reason, we took the time average of the pressure values from all the streams to obtain the differential pressure. The time-averaging was done over the portion of the streams shown within the “exposure region” in fig. 4, bounded by the lowest z-coordinate of the blade. For instance, in the example shown in fig. 4 (not corresponding to the machines described in table 1), the absolute pressure average was approximately 100 kPa and 79.5 kPa for the streams above and below the blade, respectively. Therefore, the pressure differential ($\Delta P$) for that test case was approximately 20.5 kPa. Likewise, the velocity magnitude (relative) was time-averaged over the same portion of the streams ($V_{gap}$). There was a slight tendency to find greater relative velocity values below the blade than above it but this was not deemed to be critical for characterising the passage environment. Therefore, the averaged velocity was calculated from all streams to determine a single “gap passage velocity” for each operating point.

To define the starting point for those streams considered to be near gaps, the distance between

---

**Figure 3.** The in-house tool chain for CFD simulations is highly tailored to the development of turbine technology.
the maximum and minimum radial coordinate of the blade was divided into 11 segments to generate “seeds” from which streams were released. We conducted a sensitivity test to determine the variation of both the pressure differential and gap passage velocity among the segments. For this purpose, the operating point near the peak efficiency of the large hydro unit was investigated. No singularity conditions were found near the gaps, i.e., no spikes in $\Delta P$ or $V_{\text{gap}}$ appeared with respect to pressure differential and velocities for releases in the interior part of the blade. The outer and inner segments are shown as bands in orange and blue on the left side of fig. 4, respectively. An array of points (in black in fig. 4) was then set up as the starting point of the streamlines. Such array was located just above the blade to ensure that streamlines for the most part remain near the gaps. The array of points was defined with respect to the runner blade angle, i.e., the array location was the same related to the blade position.

4. Results
To gain confidence in the evaluations of pressure differentials and passage velocities near gaps, the accuracy of the modelling approach for calculating the 3D flow fields should first be attested. In previous publications [12], we have validated the modelling approach described in section 3.1 by comparing CFD-based efficiencies of the large hydro unit against measurements from a physical model in a test rig at various net heads. Certain deviations were observed at low discharge and at low net heads. In such conditions, a vortex rope in the draft tube usually emerges and deteriorates the agreement between model outcomes and data due to the inability of the turbulence model to resolve complex vortical features. However, the overall accuracy of the CFD results was deemed acceptable to carry out the evaluation of “near gap” flow features.

The values of pressure differential and gap passage velocity near the shroud (fig. 5) and hub (fig. 6) are presented for the entire operating range of the machine. Plots of iso-lines indicate the magnitude of $\Delta P$ and $V_{\text{mag}}$ for both the low head (on the left) and the large hydro (on the right) unit as a function of the net head ($H^*$) and discharge ($Q^*$) normalized with respect to the corresponding values at best efficiency point.

At each evaluated operating condition, the magnitudes for near-shroud passages were always greater than those for near-hub events. On the pressure side of runner blades, surface absolute pressures tend to increase with radial distance; on the suction side this tendency is less evident. Given that surface pressures influence neighboring absolute pressures in the fluid volume, they yield in general greater $\Delta P$ values near the shroud. Relative velocities are strongly driven by tangential passage velocities, which are greater near the shroud due to the large radial distance. Qualitatively, we can therefore say that gap passages near the hub tend to be safer than near shroud passages for passing fish. The latter statement has not yet accounted for the gap size, which could favor grinding if gaps were relatively big. Another observation to be drawn from the modelling results is that the large hydro unit yields considerably larger magnitudes than the
low head turbine does, which in turn makes low head machines safer for fish passages at least from the standpoint of potential injuries associated to gaps. The latter should not be extended to other sources of fish bodily damage (e.g. collision on blades or low absolute pressures along the fish pathway).

Generally speaking, $\Delta P$ values are in the ranges of 16-50 kPa and 85-320 kPa for the low head and large hydro machines, respectively. If we keep discharge constant, $\Delta P$ always increases with $H^*$. If we keep the net head constant, $\Delta P$ mostly exhibits a maximum near the middle of the $Q^*$ range. In addition, $\Delta P$ shows a general tendency to increase with power output, although the two variables are only loosely correlated (low $R^2$). Greater power output is for the most part achieved with larger pressure differences between the upper and suction sides of the runner blade. This larger pressure difference is transferred to the $\Delta P$ values calculated in this study. In all instances, an increase in $\Delta P$ is concurrent with increments in $V_{gap}$ for passages near the hub (fig. 6).

$V_{gap}$ values are found in the ranges of 6.4-14.7 m/s and 14.5-37.0 m/s for the low head and large hydro machines, correspondingly. For the evaluations near the shroud, gap passage velocities do not exhibit a clear tendency with either the net head or the discharge. For the evaluations near the hub, gap passage velocities increase monotonically with discharge in both machines. While velocities in the large hydro unit are greater than in the low head machine, this difference is intensified in the shroud passages mainly due to the influence of the radial distance to the axis of rotation. One observation of the modelling outcomes for $V_{gap}$ is that the variability within each contour plot is relatively small. For instance, $V_{gap}$ is large for the
Figure 6. Absolute pressure differential ($\Delta P$) and gap passage velocity ($V_{gap}$) near the hub of both the low-head turbine and the large hydro unit, over the entire operating range of the machine

“near shroud” passages in the large hydro unit but varies only within the range 35-37 m/s. This means that the near-gap passage velocities are tightly linked to the machine features (diameter and rotational speed of the runner) and less influenced by the operating point.

5. Implications for fish passage and empirical studies of gap flows
Laboratory tests of fish subjected to simulated stressor conditions are an integral part of advancing fish friendly turbine technology. Once the parameters influencing fish bodily injury have been identified, fish experimentalists may conduct laboratory testing to correlate the magnitude of such parameters to the consequential fish mortality rate. Prior to constructing an apparatus, an experimental design phase should decide on the range of testing conditions that simulate “near gap” flows. First of all, the pressure suction effect would need to be tested in conjunction with the presence of a gap between the high- and low-pressure sides. In addition to the mechanical aspects that need to be observed under such conditions, a major challenge may arise in the attempt to achieve a pressure differential that matches the magnitudes presented in fig. 5 and fig. 6. Combining such suction effect with a high-velocity flow appears initially to be both an unrealistic and unnecessary task in view of the many uncertainties still present in our understanding of the actual effect of gap flows on fish mortality. Therefore, the gap passage velocity can instead be experimentally interpreted as an exposure time to the pressure differential. That is, lower velocities will imply longer exposure times than at greater velocities. A similar experimental scheme of representing the influence of flow velocities in terms of exposure time was to a good extent addressed in rapid decompression studies by varying the rate of
pressure change in experimental testing [13]. For juvenile Chinook salmon, it was found that the rate of pressure change slightly influenced the rate of mortality (the ratio of pressure change was the most important variable) but the validity of this influence on other fish species needs yet to be examined by specialized experimental teams testing the effect of hydraulic stressors on fish injury. This analysis would be particularly interesting for American eels as they were found to be relatively insensitive to rapid changes in pressure. Their ability to expel gas from the swim bladder may be to some extent related to the exposure time.

It is, however, worth noticing that most passage velocities extracted from our analysis of CFD-simulated flows indicate extremely short exposure times, e.g. 60-160 ms and 27-60 ms for the low-head and large hydro machines, respectively. Therefore, in practical scenarios, this short exposure time can likely be simulated as a single, sharp exposure of fish samples to “near gap” conditions. It is also important to remark that these preliminary notions of an experimental apparatus as well as the present numerical study left out the effects of shear flows that may be present near gaps and likewise contribute to the risks of mortal injury on passing fish.

While the present analysis focused on the magnitudes of “near gap” flow features, the examination of the frequency of fish encountering such detrimental conditions will complement this study. It is still uncertain what the tendency is for an incoming fish sample to end up passing near the gaps. Such passage frequency can preliminary be examined with simulations of particles released through runner blades in motion. The computational effort to conduct such simulations is considerable and deserves another independent study, results of which can reveal how incoming particles will ultimately be exposed to “near gap” flows.

6. Final remarks
Various assumptions about the injury mechanisms of fish passing turbines in operation need to be examined in two ways: firstly, by quantifying the magnitudes of the stressor assumed to lead to fish mortality and by experimentally determining the actual fish mortality rate. Both components can only be bridged by a thorough information exchange between a team with expertise in turbine hydraulics and a counterpart knowledgeable on the biological consequences of hydraulic stressors. This work sought to provide quantitative information about the extreme flow conditions that fish may encounter as they pass near gaps. Large Kaplan machines yield a potentially more detrimental “near gap” environment than small turbines do; on the other hand, the exposure time is considerably shorter in large machines. Further laboratory research is necessary to understand how these two effects ultimately play out on the fish mortality rate. Further modelling studies will focus on the use of particles to understand the mechanical forces that may bring bodies closer to the gap region, an effect that the current method—based on streamlines—cannot account for. This modelling advancement will be concurrent with transient simulations to determine frequency rates of encountering gaps. This frequency rates and the biological response of fish to gap exposure can ultimately assemble the pieces of information necessary for an integrative assessment of gap-related fish mortality risks in future turbine designs.

References
[1] Pracheil B M, DeRolph C R, Schramm M P and Bevelhimer M S 2016 Reviews in fish biology and fisheries 26 153–167
[2] Coutant C C and Whitney R R 2000 Transactions of the American Fisheries Society 129 351–380
[3] Čada G F 2001 Fisheries 26 14–23
[4] Hogan T W, Čada G F and Amaral S V 2014 Fisheries 39 164–172
[5] Richmond M C, Serkowski J A, Ebner L L, Sick M, Brown R S and Carlson T J 2014 Fisheries Research 154 152–164
[6] Motyczak L, Skotak A and Kubick R 2012 Kaplan turbine tip vortex cavitation—analysis and prevention IOP Conference Series: Earth and Environmental Science vol 15 (IOP Publishing) p 032060
[7] Pflugrath B D, Engbrecht K, Beirao B, McCann E L, Stephenson J R and Colotelo A H North American Journal of Fisheries Management, in Preparation

[8] Rivetti A, Lucino C, Liscia S, Muguerza D and Avellan F 2012 Pressure pulsation in kappan turbines: Prototype-cfd comparison IOP Conference Series: Earth and Environmental Science vol 15 (IOP Publishing) p 062035

[9] Neitzel D A, Dauble D D, Čada G, Richmond M C, Guensch G R, Mueller R P, Abernethy C S and Amidan B 2004 Transactions of the American Fisheries Society 133 447–454

[10] Wu H, Feng J, Wu G and Luo X 2012 Numerical investigation of hub clearance flow in a kappan turbine IOP Conference Series: Earth and Environmental Science vol 15 (IOP Publishing) p 072026

[11] Nennemann B and Vu T 2007 Kaplan turbine blade and discharge ring cavitation prediction using unsteady cfd 2nd IAHR International Meeting of the Workgroup on Cavitation and Dynamic Problems in Hydraulic Machinery and Systems

[12] Romero-Gomez P, Weissenberger S and Lang M 2019 Biological-hydraulic performance characteristics curves of kappan turbines HydroVision 2019, Portland OR

[13] Brown R S, Carlson T J, Gingerich A J, Stephenson J R, Pflugrath B D, Welch A E, Langeslay M J, Almann M L, Johnson R L, Skalski J R et al. 2012 Transactions of the American Fisheries Society 141 147–157