PIV measurements and CFD simulations of a hydrofoil at lock-in

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Abstract. As part of an ongoing investigation into the mitigation of vortex induced vibrations of hydrofoils, a combined experimental and numerical study of the fluid-structure interactions and wake of a hydrofoil at lock-in has been conducted at the Waterpower laboratory of the Norwegian University of Science and Technology. The hydrofoil has a blunt trailing edge and Von Karman vortex shedding induces a lock-in effect at a chord based Reynolds number of about 2.7·10⁶. The present paper presents the initial measurements of vortex shedding frequencies going through lock-in, along with CFD simulations at lock-off conditions as well as some empirical estimates of vortex shedding. Experimentally the hydrofoil wake was studied in detail using particle image velocimetry (PIV). Hydrofoil vibration frequencies were measured by both a strain gauge positioned near the trailing edge of the foil as well as by a laser doppler vibrometer (LD-V). Numerically the phenomena was simulated using ANSYS CFX. Several different turbulence models was tested, from the two-equation standard k−ε model to the scale adaptive SST-SAS model, with considerably different results. It is observed that the vibrations induced at lock-in considerably shifts and reduces the hydrofoil wake velocity deficit. Further, the CFD results suggest that the driving parameter influencing the shedding frequency is the cross flow separation distance at the trailing edge.

1. Introduction
In order to avoid unnecessary fatigue and risk of failure when designing hydraulic turbines, it is an established guideline that the shedding frequency of guide vanes, stay vanes, and runner blades should not overlap with the natural frequencies of the blades in the range of operation [1]. According to Blake et al. [2], the recognition of the link between the trailing edge geometry and vortex shedding became widespread in the 1960’s. However, as has been indicated by several researchers, the task of predicting the shedding frequency from a blunt trailing edge strut or hydrofoil can be difficult due to it being highly sensitive to both tip geometry as well the surrounding flow conditions affecting the boundary layers [3]. Nevertheless, for trailing edges with sharp edges and clearly defined separation points modern CFD methods have proven effective [4]. In modern installments of hydraulic turbines limits are constantly being pushed with respect to increasing the performance of components, while at the same time reducing material- and manufacturing costs. Accurate prediction of component behavior becomes more important because safety and expected lifespan must be kept at acceptable levels. For components which objective is to transfer forces to or from fluids, this continuous process might push the components into designs where the structure and fluid are mutually changing the
behavior of each other. This is when prediction of the behavior of the dynamics of the system becomes more involved [5]. In an effort to shed more light on the topic at hand, a combined numerical and experimental investigation is underway at the Norwegian University of Science and Technology (NTNU) for a hydrofoil under lock-in conditions. We present here some initial results.

2. Methods
2.1. Experimental Setup
2.1.1. PIV and frequency measurements setup  
The general layout of the experimental setup is illustrated in figure 1 a) and b). All measurements were done with the hydrofoil centerline angled parallel to the incoming flow field, i.e. 0 degrees angle of attack. The test section volumetric flow rate was measured by an ABB electromagnetic flow-meter located downstream of the test section. The maximum standard deviation in the mean flow during measurements was approximately 0.11 %, while the average temperature was 20.5 ± 1.5 Celsius, giving an approximate chord based Reynolds number uncertainty of 3.65 %.

The hydrofoil vibration frequencies and amplitudes were measured with a strain-gauge located near the trailing edge at approximately mid-span, as well as with a surface laser doppler vibrometer (LD-V) pointing at the trailing edge. Data acquisition was managed with National Instruments (NI) LabVIEW and NI data acquisition devices (DAQ’s). Sensor output voltages were sampled at 10 kHz, giving more than 10 samples per period for the frequencies investigated. For a more detailed explanation of the hydrofoil instrumentation, material properties and natural frequencies see Bergan et. al [6]. The frequency amplitude spectra later presented were obtained by performing a P.D. Welch power spectrum analysis in MATLAB with a Hanning window. As a means of smoothing the amplitude spectra the sensor signal were split into varying lengths relative to the approximate shedding frequencies measured, with a 50 % segment overlap.

The recording of the 2D PIV vector field in the wake of the hydrofoil was conducted with a high speed system provided by LaVision GmBH. Full resolution images of (1284 x 1024) pixels (px) were recorded in double frame mode at a sampling rate 2.4 kHz, with the camera...
oriented perpendicular to the illumination plane. The image sampling duration for each recorded image set was about 2 s. Table 1 gives a summary of the recording parameters. The recording parameters were balanced such that the average image size of a tracer particle was about 2.4 px and the estimated particle displacement between each frame was about 5-6 px. Vector fields were evaluated using a multipass method, stepping from a 96 px x 96 px interrogation area (IA) with a 50 % overlap to a 64 px x 64 px IA with a 75 % overlap. This gave an average number of illuminated particle image pairs within each area of roughly 10. According to the synthetic PIV image generation evaluation described by Raffel et al [7] these are image parameters that should give relatively low root mean square (RMS) random errors in the cross correlation evaluation of the vector fields. The degree of peak locking was investigated and found to be acceptably low. The image scaling calibration RMS error of the 3’rd order polynomial fit was about 0.38 px for the reported measurements.

Table 1: PIV recording parameters for the hydrofoil wake flow measurements

| Parameter                                           | Value                                      |
|-----------------------------------------------------|--------------------------------------------|
| Field of view (FOV) / Area of interest              | 21.1 mm x 16.9 mm / 1280 px x 1024 px (x-y) |
| Interrogation volume / Interrogation area            | 1.06 mm x 1.06 mm x 0.5 mm / 64 px x 64 px (x-y) |
| Experimental velocity range                         | (8 - 14) m/s                               |
| Observation distance & Lens F-number                | 215 mm & 5.4                               |
| Recording method & Camera sensor                    | Double frame/Double exposure & CMOS        |
| Exposure time & image acquisition rate              | 250 µs & 2.4 kHz                           |
| Image processing mode                               | cross-correlation                          |
| Mean tracer particle diameter \(d_p\)               | 13 µm                                      |
| Tracer particle density \(\delta_p\)                | 1.1 g/cm³                                   |
| Illumination source                                 | Nd:YFL dual cavity laser, 527 nm wavelength |

To compute the uncertainty, \(\Delta U\) in the time-averaged stream wise velocity \(U\) distributions later presented, the following estimate was applied [8];

\[
\Delta U = \frac{\sigma_U}{\sqrt{N_{eff}}},
\]

where \(\sigma_U\) denotes the standard deviation in \(U\) across all samples during a measurement series and \(\sqrt{N_{eff}}\) is the effective number of independent samples of \(U\). \(\sqrt{N_{eff}}\) involves the computation of the auto-correlation of the time-series of the instantaneous stream wise velocity vectors \(u(x, y, t)\) and approaches the total number of samples \(N\) in a signal in the case that the samples of \(u\) are completely independent. Finally, the error in \(U\) due to uncertainty in the the laser-plane span-wise position was investigated by measuring the hydrofoil wake at a parallel 10 mm offset plane. While the test clearly indicated 3D effects coming from the test section channel walls, the relative uncertainty in the positioning of the laser plane on the scale of 0.5 mm should have negligible impact on the time-averaged wake velocity distributions.

2.1.2. Hydrofoil Profile Geometry and Surface Roughness  To facilitate comparison with CFD simulations the hydrofoil surface roughness was measured with a profilometer at different chord-wise positions, giving a maximum arithmetic roughness average \(R_a\) of about 5.8 µm near the leading edge. Following Schlichting & Gersten [9], we approximate the technical roughness height by \(k_{tech} = 3.5R_a\). For a maximum chord based Reynolds number of approximately \(Re_C = 4.5\cdot10^6\) this gives for a fully turbulent boundary layer an estimated \(k^+\) value of approximately 11. Hence we assume that the surface roughness height exceeds the viscous sub-layer near the leading edge and may play an important effect in the development of the boundary layer.
Figure 2 shows the blade geometry. The hydrofoil surface position data was measured with a Leitz PMM-C 600 coordinate machine, capable of a repeat-ability range of less than 0.6 $\mu$m. These measurements were performed after the foil had been coated with a thin layer of matte black paint, to avoid unnecessary laser reflections in the PIV measurements. Figure 2(b) shows the measured trailing edge along with the numerical grid points along the blade used in the CFD simulations. A noticeable difference is only visible at the steepest part of the trailing edge, where the flow is assumed to be separated. Hence any considerable differences in between the results from the measurements and the simulations are assumed not to stem from differences in the profile geometry.

![Figure 2: a) Blade geometry. b) Measured trailing edge geometry plotted along the numerical grid wall cells.](image)

### 2.2. Numerical Setup

A numerical study was performed to investigate how well the shedding phenomena is predicted in the lock-off region. The numerical simulations was purely Computational Fluid Dynamics (CFD), i.e. no structural response. The simulations was performed in ANSYS CFX. The numerical domain and computational grid is illustrated in figure 3. The channel extends $>> 10D_h$ upstream of the blade to ensure that the flow entering the test section is fully developed [10]. The domain was also extended downstream to minimize the risk of back-flow and outlet conditions affecting the blade vortex dynamics. The inlet boundary condition was static pressure, and the outlet condition was a mass flow corresponding to the different flow velocities. The turbulence intensity at the inlet was tested in the range $I \in [0\% - 10\%]$, with no noticeable difference in the turbulence levels in the test section.

The mesh was created in ANSYS ICEM CFD, and contained about 13 million, all hexahedral elements. When refining the mesh, it was observed that the coarser mesh simulations under-predicted the shedding frequency. On the final mesh, for a typical flow velocity tested in this article, $U_{ref}=11$ m/s, the maximal $y^+$ value was 1 and $\leq 0.5$ at the leading and trailing edge, respectively. As an implicit numerical solver was used the time-step was chosen to be $8e-5$ s, giving a corresponding Courant number of 3. Shedding frequencies in the order of $f = 500$ Hz were expected, and the time-step used corresponded to about 25 points per period. For the Reynolds Averaged Navier-Stokes based simulations several different turbulence models have been tested in order to investigate the effects on the predicted shedding dynamics. The standard $k-\epsilon$ model [11, 12], was expected in this case likely to struggle to give an accurate result, due to the known problems with separation and streamline curvature. Another two-
equation model, the Wilcox $k - \omega$ model [13, 14], was also tested, along with the $k - \omega$ SST [15], a combination of the two. Additionally, the scale-adaptive SAS-SST model [16] was tested. Given an adequate computational grid the SAS model resolves the larger turbulent structures, at increased computational costs. Further, 2-dimensional simulations was performed on a simplified, shortened test section to investigate means of speeding up the simulations.

2.3. Empirical estimates for vortex shedding
It may be interesting to compare the measured and CFD predicted shedding frequencies with some empirical estimates. The first empirical approach utilized in the present study is the traditional Strouhal shedding frequency [17], $f_s$, here defined as

$$f_s = \frac{St \cdot U_\infty}{D}, \quad (2)$$

where $D$ in this case is approximated as the blade thickness at the trailing edge, $St$ denotes the Strouhal number and $U_\infty$ is the free stream velocity. For the chord based Reynolds number range encountered in this study, the Strouhal number was chosen to be $St = 0.22$, a commonly used value [18]. The thickness of the trailing edge, $D = 4.8$ mm, is measured at the point where the curved surface starts, see figure 2(b).

An empirical formula more specific for the Francis turbine and different trailing edge geometries is described in the paper by Brekke [1], where the frequency of the vortex shedding is approximated by:

$$f_s = 190 \frac{B}{100 (t + 0.56)} \frac{U_\infty}{D}, \quad (3)$$

Here $B$ is a constant linked to the trailing edge geometry, $U_\infty$ is the free stream velocity, and $t$ is the blade thickness in [mm]. The constant $B = 131$ is chosen from [1], and is related to a trailing edge geometry very close to the one tested here. Note however that in Brekke’s considerations, all blade geometries had parallel upper and lower surfaces. This is not the case for the blade geometry in this study which has tapered surfaces toward the trailing edge. Hence, strictly $t \neq D$. Nevertheless equation 3 is used here in its current form.

**Figure 3:** Numerical flow domain and mesh around blade
2.4. Wall effects
The hydrofoil tested in this study has a front section area to test section area blockage ratio of 8 %, and requires a correction for the measured Strouhal number, $St_{\text{meas}}$, due to added wall effects[19, 20]. Following the considerations of Ota et al. [20] for incompressible flow over a 2D-geometry experimental setup, we estimate the correction factor, $\epsilon$, by the following relation for the free stream Strouhal number $St$,

$$St = St_{\text{meas}}(1 - \frac{t}{h}).$$  

(4)

Here $t$ denotes the height of the hydrofoil (12mm) and $h$ the height of the measuring section (150mm), giving $t/h = 8\%$. The observed correction factor $\epsilon$ is here estimated by the assumed free stream Strouhal number of 0.22, and the measured Strouhal number outside the range of lock-in. Note that for the rough empirical estimates for the shedding frequencies later presented the free stream velocity is approximated as the average velocity across the undisturbed test section, such that $U_\infty = U_{\text{ref}}$, since, to the authors knowledge there is no reliable way to estimate the correction factor a-priori for the geometry tested in this study.

3. Results

Experimental results

Figure 4: Hydrofoil vibration frequencies and shedding frequencies measured by PIV, strain-gauges and LD-Vibrometer. The relative hydrofoil vibrational amplitude is plotted along the right y-axis.

Figure 4 presents the hydrofoil vibration frequencies and shedding frequencies measured by PIV, strain-gauges and LD-Vibrometer. There is a precise agreement between the measuring techniques in lock-in, with resonance starting to occur at around 11.1 m/s as indicated by the sharp rise in the vibrational amplitude. The first, standing peak found in the strain-gauge frequency amplitude spectrum, presented for some velocities outside lock-in in figure 5, is identified as the hydrofoil natural frequency [6]. The second, broad ranged traveling peak found in the frequency spectrum in the strain measurements can be identified as the shedding frequencies in lock-off conditions, as indicated by the agreement with PIV measurements in the wake. Since the shedding frequency is assumed to be inherently gaussian about it’s mean value, the size of the error bars given in figure 4 in the strain-gauge 2'nd peak was estimated by the half width at half maximum (HWHM) for the smoothed frequency distribution curves given in
In lock-in, the uncertainty in the hydrofoil vibrational frequency was estimated by the standard deviation in peak frequency from repeated measurements at 11.1 m/s and found to be approximately 1.6 Hz.

Figure 5: Amplitude frequency spectra for increasing reference velocities from strain-gauge voltage signal, showing the traveling shedding frequency peak (left peak) approaching the natural (standing) frequency peak of the hydrofoil. (a) $U_{\text{ref}} = 9.1$ m/s. (b) $U_{\text{ref}} = 9.6$ m/s. (c) $U_{\text{ref}} = 10.1$ m/s. (d) $U_{\text{ref}} = 10.6$ m/s.

The average observed Strouhal number, as found from frequency estimates by the PIV measurements given in figure 4 for reference velocities between 9.1 m/s and 10.6 m/s, is 0.274, with a standard deviation of about 1.3 %. The ratio between the assumed and the measured Strouhal frequency is $\approx 0.794$, and from equation 4 this gives an average correction factor $\epsilon \approx 2.57$.

Figure 6 gives the normalised time-averaged streamwise velocity distribution in the hydrofoil wake measured by PIV at two downstream positions $x_1 = 9.9D$ and $x_2 = 13.3D$, measured from the trailing edge tip. Sets of varying reference velocities are included, both for lock-off (6(a)-(b)) and lock-in conditions (6(c)-(d)). It is noted that the wake velocity distributions varies considerably more during lock-in.

**Numerical results**

Figure 7 shows the stream-wise velocity at two vertical lines, $x_1 = 9.9D$ and $x_2 = 13.3D$, downstream of the blade at $U_{\text{ref}} = 9.1$ m/s. The velocities are time averaged over 2 s and $\approx 100$ shedding periods in the experiment and simulations, respectively.

Next, the frequency of the shedding is compared in figure 8(a). The uncertainty in the fast Fourier transform of the simulated time-signal is estimated to be $\xi \approx 5$ Hz. We observe that a linear trend is found for all turbulence models, as is reported by most sources e.g. [17, 1], and thus it is assumed that the general vortex shedding phenomena is captured. For ease of comparison figure 8(b) shows the experimental and numerical results from the SAS simulations.
Figure 6: Time-averaged PIV measured velocity distributions normalised with respect to the mean channel velocity for different downstream positions. Height normalised by trailing edge thickness, with $y = 0$ set at the hydrofoil center line. In (a) and (c) $x = 9.9D$. In (b) and (d) $x = 13.3D$. Uncertainty error bars are only plotted for reference velocities of 9.1 m/s and 11.1 m/s, for clarity.

Figure 7: Experimental and numerical comparison of the time-averaged velocity profile downstream of the trailing edge. (a) $x = 9.9D$. (b) $x = 13.3D$.

together, along with empirical estimates for vortex shedding. Finally, figure 9 gives a comparison of the instantaneous velocity fields obtained with the different numerical schemes. The vertical black lines denotes one of the spatial locations of the PIV profile measurements ($x = 9.9D$)
4. Discussion

From Heskestad and Olberts [3] we note that according to their measurements, 3 geometries all similar in design to the foil under investigation here, gave a relative standard deviation in the shedding frequency Strouhal number of about 12%. This indicates that the trailing edge shedding frequency is quite sensitive to small changes in the geometry. Hence the error in the rough empirical estimates (equation 2 & equation 3) for the shedding frequencies of about 20% and the wall correction factor estimated to $\epsilon = 2.46$ is assumed to contain both wall effects as well as boundary layer separation effects specific to the hydrofoil and trailing edge geometry tested here. The level of the error is sobering, and illustrates the need for either model measurements or accurate case dependent numerical tools in the design phase of hydraulic turbine blade components, even in lock-off conditions.

Comparing experimental and numerical results in figure 7 we observe that the wake center is consistently shifted slightly below the hydrofoil center line, with about $0.2D$ in the experiments at lock-off conditions. It is believed that this is due to the asymmetry of the trailing edge, as the upper separation point is allowed to travel closer to the hydrofoil’s centerline due to the relatively gentle curvature, effectively shifting the wake profile. In the simulations points of zero wall shear stress was investigated in order to study the separation points impact on the wake and shedding frequencies obtained. When different flow velocities are compared, the separation occurs later the higher the flow velocity. Correspondingly, the “perceived” thickness of the trailing edge is also thinner at higher speeds. When comparing the results from different
turbulence models at a fixed reference velocity it was also found that a delayed separation point corresponded to an increase in the shedding frequency. This indicates that the differences in the turbulence models lie in the simulated boundary layer and subsequently the numerical prediction of the boundary layer separation points. As an example, comparing results in figure 7-9, the \( k - \epsilon \) model separated latest along the trailing edge, resulted in the highest shedding frequency and the most vertically shifted wake.

As an overall trend, the numerical simulations tended to underestimate the velocity deficit, except for the \( k - \epsilon \) model, where the deficit was overestimated. The wake is considerably different depending on which turbulence model is used. From figure 9 we can start to understand why the different velocity profiles in figure 7 look like they do. In the SAS model, the wake breaks down into smaller vortexes, and thus the mean flow in the wake is higher and flatter. This corresponds to the lower velocity deficit in figure 7. Similarly, it can seem like the \( k - \epsilon \) model have the strongest velocity deficit and widest wake. It is also observed that the oscillations in the \( k - \epsilon \) wake are very small. Compared to the PIV measurements, the SAS model performs qualitatively best (not shown), while interestingly, figure 7 indicates that the mean velocity distribution is farthest from the experimental values for this model.

Figure 8 (a) show the trends of the shedding frequency for different turbulence models. A linear best fit is performed and the slopes are used as comparison. The slope is varying with \( \approx 22\% \), from \( k - \epsilon \) to SAS. The SST and the SAS model perform very similarly. 2-dimensional simulations with the SST turbulence model is also included in figure 8 (a). Clearly, they provide some predictive value while also providing a significant speedup factor, in this case, of \( \approx 70 \times \). When comparing the numerical and experimental results in figure 8 (b), it is seen that there is a general under prediction of the shedding frequency by the numerical simulations. Based on the slope of the experimental results after a linear best fit the offset between the SST and SAS model compared to the experimental values are \( < 4\% \), while the numerically observed Strouhal number is about 10 \% lower than the comparatively obtained experimental Strouhal number of 0.274 in lock-off conditions. A closer investigation of the possible reasons for the relative offset between the CFD calculations and the experimentally obtained results are part of future work, along with simulations utilizing a fluid-structure interaction coupling for numerical investigation of the hydrofoil behaviour in lock-in.

5. Conclusion
The wake and shedding frequencies from a hydrofoil with a blunt, asymmetrical trailing edge has been investigated for free-stream velocities where turbulent von-karman vortexes incites a lock-in effect. At lock-in we observe larger stream-wise velocity fluctuations in the hydrofoil wake, than in lock-off conditions, likely due to wandering of the upper separation point at the trailing edge tip. Experimentally obtained shedding frequencies has been compared to numerical simulations as well as empirical estimates. The relative differences between simulations with different turbulence models clearly indicate the difficulties in the modelling of the separation points and subsequent wake characteristics crucial to estimating the risk of lock-in at the design phase for a hydraulic turbine blade component. The numerically obtained results for the trend in the shedding frequencies are in relative agreement with previous studies for similar trailing edge geometries [3], indicating that a delayed separation point leads to increased shedding frequencies.

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