Simulation of steady and unsteady flows through a small-size Kaplan turbine

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Abstract
The increasing demand for electric energy has pushed to rely on the microhydropower as one of the most cost-effective technologies to improve the grid stability in rural localities. This article investigates the steady and unsteady flows in a complete small Kaplan turbine by means of the solver ANSYS-CFX. Examination of the basic design parameters in terms of blade setting, vane opening, and interdistance has revealed the best operating parameters and the potentiality of performance improvement. Moreover, the Fast Fourier Transform of static pressure fluctuations recorded at different monitor points allowed analyzing the flow unsteadiness arising from the stator-rotor interactions (RSI). The prevailing harmonics coinciding with the modes of RSI were determined and compared with the analytically predicted main diametrical modes and frequencies of the pressure waves. The found results in this category of low-head small Kaplan turbine may serve in developing similar hydroturbines suitable for the microhydropower.

KEYWORDS
ANSYS-CFX solver, hydraulic performance, low-head, pressure fluctuations, small Kaplan turbine, stator-rotor interactions, steady and unsteady flows

1 INTRODUCTION

Hydropower remains the most important renewable source for electrical power generation worldwide, providing one-fifth of the world's electricity supply.1 Hydropower is able to back-up the integration of renewable energy and contribute to the continuity of energy supply and grid stabilization owing to immediate response to varying loads and demands.3 Kaplan turbines operate efficiently on sites with low heads and relatively high flow rates. The vane opening and runner blade setting are adjusted separately to provide a wide flow range and a flat efficiency curve.4 Most of hydropower potentials are located within the proximity of off-grid remote areas, where the small Kaplan turbines are more suitable.5 They may run off-river without a significant water storage,6 but their hydraulic efficiency is not high compared to large Kaplan turbines added to water feeding not stable.

Kaplan turbines have a rather large vaneless space inducing a complex dissipation of vanes' wakes and vortices, causing large pressure pulsations,7 added to the flow rotation and secondary flows near the hub and shroud, which are difficult to capture.8 The energy transfer is accompanied by pressure oscillations caused by vortices, cavitations, and complex...
unsteady flow phenomena inducing excessive blade vibration. Zhang et al. reviewed the vortex formation as one of typical structures of unsteady flows in hydroturbines leading to various kinds of instabilities (large pressure fluctuations), as well as the vortex rope in the draft tube. Another specific feature of Kaplan turbines is the stator-rotor interactions (RSI) that are known to induce pressure fluctuations on the runner blades, which provoke additional fatigue of the machine.

Given the limited amount of information that an experimental investigation could provide the computational fluid dynamics (CFD) techniques have become powerful tools that may substitute to the experiments to get the detailed information about the strong effects from rotation, curvatures, leakage, and unsteadiness in different parts of a turbo-machine. Trivedi et al. reviewed the CFD techniques used to simulate the hydraulic turbines in steady and transient operating conditions, including a discussion on different turbulence models. Their conclusion is that the two-equation eddy-viscosity models are more robust and not computationally expensive and become common in the steady and unsteady simulations. Liu et al. by using the code FLUENT and RNG $k$-$\varepsilon$ turbulence model simulated unsteady flows in the whole passages of model and prototype of Kaplan turbine. The predicted pressure fluctuations at different points agreed well with the test data in terms of frequency and amplitude. Also, Wu et al. used the code FLUENT and several turbulence models to simulate unsteady flows throughout the whole passages of a prototype Kaplan turbine. By analyzing the pressure fluctuations in different locations, upstream and downstream of the runner, they showed that the first peak is a low frequency given by upstream propagation of vortex rope while the second peak corresponds to the runner rotating frequency. Rivetti et al. investigated numerically the pressure pulsations in a large Kaplan turbine focusing on the tip gap flow and the pressure field at the discharge ring where two zones of low pressure were identified. The first is induced by a local acceleration of the flow entering into the gap and the second corresponds to the tip gap vortex. Ko and Kurosawa presented flow simulations in a Kaplan turbine and compared between the Reynolds stress turbulence model (RSM) and the large eddy simulation (LES) model. They concluded that RSM model damped the blade tip vortex and the pressure fluctuations at runner outlet were underestimated. Javadi and Nilsson simulated the unsteady turbulent flows through the passages of U9 Kaplan turbine first by considering the code OpenFOAM with RNG $k$-$\varepsilon$ as a turbulence model, and later used the Unsteady Reynolds Average Navier Stokes (URANS) and the hybrid URANS-LES. They concluded that both captured the tip vortex, hub vortices, blades’ wakes, and on-axis structure, and also showed that the dynamics of tip runner vortices is affected by the rotation and shroud boundary layer. Based on flow simulations in the same Kaplan turbine model, Iovâne discussed the advantages and limitations of RANS and URANS modeling, while several turbulence models were used. Most of the researches focused on the pressure fluctuations caused by RSI and the torque fluctuation and vibration induced by the asymmetric pressure loadings were of great concern. Basically, there are two types of RSI: the viscous-wake interaction resulting from downstream blades cutting through upstream vanes’ wakes and the potential-flow interaction induced by the movement of blades along vanes. The modulation between the potential flow perturbation and that caused by vanes is to induce pressure waves propagating towards the runner blade at the same time that RSI generate large pressure oscillations. Wang and Tsukamoto and Zhang and Tsukamoto predicted the RSI pressure pulsations in pumps with a reasonable accuracy. Gagnon and Deschênes simulated in a propeller turbine the interactions between vanes and blades and revealed the pressure and torque fluctuations at part-load and over-load and the frequencies and peaks corresponding to the potential interaction. Rodriguez et al. presented an original interpretation for the characteristics of RSI of a pump-turbine which were clearly observed in the frequency domain and the harmonics of blade passing frequency (BPF) were shown to have a particular relationship among the amplitudes. The phenomena of RSI in Francis turbines were the subject of various experimental and numerical studies. Trivedi et al. carried out experiments in a high head Francis turbine where the pressure-time measurements were performed at several operating conditions added to unsteady simulations. The pressure-time signals showed that the complex phenomena of RSI caused torque oscillation away from best efficiency point (BEP) and the highest amplitudes were captured by sensors mounted on the blade. Amiri et al. investigated experimentally the sources of periodic loads exerted on a Kaplan turbine runner when operating at BEP and off-design. The torque measurements revealed an asymmetric flow at the outlet of distributor vanes. Li et al. analyzed RSI in a pump-turbine based on unsteady simulations under six vane openings, and found that at the best vane opening the frequencies are mainly the BPF and the harmonics. Zhang et al. investigated the pressure fluctuations in a prototype of reversible pump-turbine and showed that at low load condition the pressure fluctuations are quite significant especially in vaneless space and spiral casing with BPF as the dominant frequency. Similarly in a pump-turbine, Li et al. showed that there are high-amplitude pressure fluctuations at partial operating points and the frequency characteristics were determined by using the time and frequency domain analysis methods. Recently, Chen et al. simulated the transient characteristics of a Kaplan turbine during load rejection, and revealed evolutions of inner flow patterns and
varying regularities in runner rotation speed, unit flow rate, unit torque, and static pressure which were consistent with the experimental data.

According to the literature, much of research about the phenomena of pressure pulsations were mainly dedicated to the Francis and Pump turbines but lesser for the Kaplan turbines and practically inexisten for the low-head (3-9 m) small Kaplan turbines. This article contribution tends to bridge this gap for this category of a small-size Kaplan turbine by providing insights about hydrodynamic characteristics and unsteady flow phenomena. Indeed, this Kaplan turbine has a moderate efficiency due to the scale effects and large axial and radial gaps which induced high-pressure fluctuations.

2 | TESTED KAPLAN MODEL

This model of small Kaplan turbine (Figure 1A) has a maximum output of 0.52 kW when operating at 2000 rpm and a head of 3.05 m (see Table 1). It consists of a distributor with four vanes of variable opening, a runner with four adjustable blades, and a straight conical draft tube. The vane and runner assembly are surrounded by a Perspex window for visibility. To test this turbine, a closed loop test-rig (Figure 1B) was used, consisting of PVC pipes, a fixed speed centrifugal pump, and a butterfly valve with a lever-type operating gear to regulate the flow. The blade adjustment is affected by means of a hand-wheel located on the bedplate and the vanes are adjusted by four locking rings. The generator is trunnion-mounted and the power measurement is obtained by means of the frame reaction. The torque is measured by means of Pony brake.

**FIGURE 1** A, Vanes and runner assembly. B, Test-rig

**TABLE 1** Main geometry parameters

| Component   | Parameter                      | Symbol | Value        |
|-------------|--------------------------------|--------|--------------|
| Runner      | Blades number                  | $N_r$  | 4            |
|             | Tip and hub diameters (mm)     | $d_t$, $d_h$ | 100, 50     |
|             | Tip clearance (mm)             | $\epsilon$ | 0.8         |
|             | Average chord (mm)             | $c_r$  | 52.2        |
|             | Blade setting angle (°)        | $\beta$  | 10 - 30     |
| Distributor | Vanes number                   | $N_s$  | 4            |
|             | Shroud diameter (inlet/outlet) (mm) | $d_s$ | 105.6, 101.6 |
|             | Average chord (mm)             | $c_v$  | 78.36        |
|             | Opening angle (°)              | $\alpha$ | 0 – 30      |
| Draft tube  | Shroud diameter (inlet/outlet) (mm) | $d_s$ | 101.6, 134.5 |
|             | Total length (mm)              | $l_d$  | 198         |
|             | Divergence half angle (°)      | $\delta$ | 4.75       |
reaction (Figure 1 B) via the force gauge indicated by mercury column. Other manometers measure the pressure difference of the Venturi meter and the absolute heads at inlet and discharge. The accuracy of mercury multimanometers is about 1 mm of upper and down meniscus and that of tachometer is ±0.5% of full-scale rpm.

3 | CFD MODEL

The 3D unsteady turbulent noncavitaing water flow is modeled by the simplified URANS equations with $k-\omega$-based SST and solved by means of the code ANSYS-CFX.\textsuperscript{32}

\[ \frac{\partial U_i}{\partial x_i} = 0, \]  
\[ \frac{\partial U_i}{\partial t} + U_j \frac{\partial U_i}{\partial x_j} = - \frac{\partial P}{\partial x_i} + g_i + \frac{\partial}{\partial x_j} \left( \nu \frac{\partial U_i}{\partial x_j} - \tau_{ij} \right) + S_i, \]  

where $U_i$ is average relative velocity component, $\tau_{ij}$ is the turbulent stress evaluated from the turbulence model, and $g_i$ is gravity force component. $S_i$ is a component of the sum of Coriolis and centripetal forces $\vec{S} = -2\Omega \times \vec{U} + \Omega \times (\Omega \times r)$, with $\vec{\Omega}$ rotational velocity vector (rad/s) and $\vec{U}$ relative flow velocity.

4 | GEOMETRY AND MESH GENERATION

The geometry is obtained by means of a coordinate measuring machine consisting of a high-speed scanning sensor, which records points based on optical triangulation of laser beam projected onto a surface and its reflection is collected and redirected to a camera. The scanning error is about 4 μm. The coordinates of points are stored and transferred to CAD software where the final geometry is obtained as shown by Figure 2. According to Figure 3 the vane opening is defined relatively to the axis of rotation, whereas the blade setting is with respect to the tangential direction.

The rotating domain (runner) and fixed domains (distributor vane and draft tube) are discretized separately with nonuniform hexahedral meshes generated for different vane opening and blade setting angles. An O-grid is used for the refinements around the vane and blade (Figure 4), and through the tip clearance there are 15 mesh lines. The automatic near-wall treatment allows a transition from the low-Reynolds number model in the viscous sublayer (fine meshing $y^+ < 2$) to the wall function ($20 < y^+ < 100$). The first layer of nodes from a wall is estimated by $\Delta y = y^+ \mu / \rho V_t$, where $V_t = V_\infty \sqrt{C_f / 2}$ and $C_f = 0.026 Re_c^{-1/7.33}$ for Reynolds number $Re_c = \rho V_\infty c/\mu$ based on vane/blade chord. For the nominal point of maximum power ($N = 2000$ rpm and $Q = 36.3$ L/s), $Re_c$ is equal to 532 963 and 604 317 in the vane and runner blade, respectively. The grid size dependency considered five grid sizes for runner blade while the grid sizes of vane and

**FIGURE 2** Kapan turbine geometry and main dimensions
draft duct were kept at optimal values. As shown in Figure 5, the hydraulic efficiency varied about 0.095% between the third and fourth grid and slightly (0.038%) between the fourth and fifth grid. The selected fourth grid has 237,960 nodes per a vane passage, 598,240 nodes per one runner blade passage, and 176,800 nodes per a sector of draft duct, reaching a total grid size of 4.052 million of nodes. Figure 6 presents the values of $y^+$ in the vane and runner blade which are acceptable for the automatic wall function treatment.

4.1 Boundary conditions

As boundary conditions, a mass flow rate is imposed at the inlet of distributor assuming the flow velocity uniform and normal, while a static pressure is prescribed at the outlet of draft tube. These are widely accepted boundary conditions for flow simulations in hydraulic turbines. The smooth no-slip wall is imposed to the runner hub and blade and vane surfaces, and a counter-rotating wall is set for surfaces composing the rotating domain which are not rotating. The
boundary conditions used the measured values of rotational speed, flow rate, net pressure heads for each operating point. For the conformity with the actual physical model, the losses caused by the bend and ducts lengths upstream vanes and downstream draft tube are considered in order to set the effective values in boundary conditions. The upstream ducting has an inside diameter of 134.5 and a bend of a radius of curvature of 154.5 mm. As the distance from the draft tube exit till the static pressure tapping is known the linear loss $h_{\text{loss},l} = f \frac{L \nu^2}{d g}$ is estimated based on the friction coefficient $f$ function of Reynolds number and roughness. The bend loss is evaluated by $h_{\text{loss},b} = \zeta_b \frac{V^2}{2g}$ where $\zeta_b = 0.25$ obtained by knowing the ratio between the radius of curvature and duct diameter and the relative roughness. Examination of the variation of efficiency with turbulence intensity showed a slight sensitivity with levels of 1%, 5%, and 10%, and thus an intensity level of 5% was selected. Mulu et al. adopted a similar value and concluded that the turbulence intensity has a slight effect.

### 4.2 Solver setting

The high-resolution scheme with a local time step was used to obtain the steady solution of the incompressible RANS equations. In each time step, the flow rate at the outlet of domain has to be enforced by that at inlet and converged as the number of time steps is sufficiently large. The pressure velocity coupling is achieved via the SIMPLE algorithm of Patankar and Spalding in conjunction with the momentum interpolation technique of Rhie and Chow to prevent pressure field oscillations. For the steady flow characterization, the frozen rotor model was used, where the coupling between the cells zones is done by keeping the absolute velocities which are just switched between the relative and absolute frames. On the other hand, the hydraulic performance computations used the stage interface, where the flow field data are averaged circumferentially at interfaces. For unsteady flows, URANS equations with $k-\omega$-based SST were solved by the high-resolution discretization scheme and the time marching scheme corresponded to the second-order backward Euler scheme. The frozen rotor steady-state solutions were taken for initialization. The data of flow interactions between the rotating and stationary domains are exchanged by means of the transient rotor-stator interface simulated at every time step to account for the frame of reference transfer and pitch change between vane exit and runner entry. The relative motion was simulated through the general grid interface (GGI) connection, which permits the nonmatching of nodes locations, element type, surface overlap among others. The transient simulations consist in physically advancing the flow in a real time, and since it is not possible to use the final time step's flow field alone to assess convergence in terms of RMS/MAX residuals and imbalances, some sort of averaging was considered over an appropriate timescale. The solver iterated till a number of 100 times per a step targeting a selected residual of $10^{-6}$ and for every time step 10 internal loops were selected. The minimum time step satisfied the components interactions based on the necessary time to cover the geometrical coincidence.

\[
\Delta t_{\text{min}} = \Delta t_{\text{round}} \frac{\text{GCD}(N_i, N_{3-i})}{N_i N_{3-i}} = 0.25 \Delta t_{\text{round}}.
\]  

However, to resolve the high frequencies, the time step corresponded to a rotation less or equal 1.5°, equivalent to a physical time step of 125 microseconds.
5 | STEADY FLOW RESULTS

The stage interface served to compute the hydraulic performance where the flow properties are circumferentially averaged. The torque $\tau = \int_S r \times (-p n) ds$ is obtained by integrating element forces over the runner surfaces. The hydraulic efficiency $\eta_h = \frac{P}{\rho g Q H}$ is computed in terms of head ($H$), power $P = C\omega$, and discharge ($Q$). The unit speed $\omega_{11} = \frac{\omega d}{\sqrt{H}}$ (rad/s), unit discharge $Q_{11} = \frac{Q}{d^2 \sqrt{H}}$ (m$^3$/s), and unit power $P_{11} = \frac{P}{d^2 H}$ (W) are defined according to References 14 and 41.

5.1 | Validation

The computed performances are compared with the measurements to validate the CFD model. As depicted in Figure 7, there is a fairly good agreement around BEP but it does not fully agree at high and low speeds of rotation. The computed performance for the variable rotational speed 500-3500 rpm corresponded to a constant head of 3.05 m, vane opening of 20°, and blade setting $\beta = 30^\circ$. As revealed, the performances present two peaks; the first corresponds to the maximum power ($N = 2000$ rpm, $Q = 36.3$ L/s) and the second to the maximum efficiency ($N = 1750$ rpm, $Q = 34.2$ L/s). The first point showed that the computed torque is equal to 2.462 Nm compared with that measured of 2.488 Nm within a relative error of 1.07%, whereas for the power the computed value is equal to 515.54 W compared with that measured of 521.06 W within an error of 1.06%, and finally the computed efficiency is equal to 52.58% compared with experimentally deduced value of 49.37% within an error of 6.51%. For the second point, the computed torque is equal to 2.676 Nm compared with that measured of 2.730 Nm within a relative error of 1.978%, whereas the computed power is equal to 491.70 W compared with that measured of 499.37 W within an error of 1.536%, and finally the computed efficiency is equal to 53.54% compared with experimentally deduced value of 50.03% within an error of 7.015%. To have a view on the global errors incurred, the values of root mean square error RMSE $= \sqrt{\frac{1}{n} \sum_{i=1}^{n} (X_{m,i} - X_{c,i})^2}$ are respectively equal to 0.181 Nm, 25.95 W, and 4.033%, for the torque, power, and efficiency. The hydraulic efficiency has the highest value of RMSE owing to interdependent errors incurred during the measurements of static pressure difference, head and rotational speed, in addition to the omission of the external losses in the actual CFD model. The measured hydraulic efficiency of the turbine is defined as the ratio of energetic output to input and evaluated by $\eta_h = \frac{P}{\rho g Q H} = 1 - \frac{\text{Loss}_{\text{Int}} + \text{Loss}_{\text{Ext}}}{\rho g Q H}$, which includes both internal losses $\text{Loss}_{\text{Int}}$ (which take place in the inner passages of the components) and external losses $\text{Loss}_{\text{Ext}}$ (which appear outside of the inner passages). In this closed test-rig, the net head $H$ represents the difference between the total pressures measured upstream the bend and downstream draft pipe and was considered in the actual CFD model. However, the external losses were not assessed, since the CFD model requires complicated geometries and grids to include the three components of
external losses such as external leakage loss ($\text{Loss}_{\text{external leakage}}$), disc friction loss ($\text{Loss}_{\text{disc friction}}$), and bearing friction loss ($\text{Loss}_{\text{mechanical}}$).

### 5.2 Computed hydraulic performances

The hydraulic performances are computed for the vane opening $\alpha = 20^\circ$ and blade setting $\beta = 30^\circ$, which are given in terms of unit power $P_{11}$ and hydraulic efficiency $\eta_h$. Figure 8A plots the evolution of $P_{11}$ vs $\Omega_{11}$ for fixed values $Q_{11}$, which depicts a variation following a parabola. After reaching peaked values, $P_{11}$ drops rapidly due to increased losses by flow incidence, wakes, secondary flows, and leakage flow. Hydraulic efficiency $\eta_h$ plotted against $\Omega_{11}$ (Figure 8B) for given values $Q_{11}$ has the same trend but the values of peak are shifted. By varying simultaneously, the unit speed and unit flow rate, the velocity triangles approach the same configuration as that of nominal speed. The peaked values of $P_{11}$ are given for $\Omega_{11}$ in between 11 and 17 rad/s ($N_{11} = 105-162.3$ rpm), while the maximum yield corresponds to $\Omega_{11} = 15$ rad/s ($N_{11} = 143.2$ rpm) and $Q_{11} = 2.1$ m$^3$/s. On the other hand, the local peak values of efficiency lay for $\Omega_{11}$ in between 10 and 14 rad/s ($N_{11} = 95.5-133.7$ rpm), while that of maximum yield corresponds to $\Omega_{11} = 11.5$ rad/s ($N_{11} = 114.6$ rpm) and $Q_{11} = 1.95$ m$^3$/s.

### 5.3 Flow structure

The flow solution at the nominal point ($N = 2000$ rpm and $Q = 36.3$ L/s) is based on the frozen interface. Besides boundary layers and wakes structures, the flow patterns in the blade endwall regions exhibit an imbalance between the normal pressure gradient and the centripetal acceleration to cause a flow deviation from pressure side to suction side. For the blade setting $\beta = 20^\circ$, the position of the stagnation line moves toward the pressure side (Figure 9A), and thus, the effective flow angle is varied, added to the recirculation zones appearing over the fore of blade suction side and near the hub from trailing edge. The secondary flows are revealed by the streamlines deflected from hub and tip toward the mid of blade suction side, added to the clear passage vortex formed at the hub corner from blade suction side, which strengthens with vane opening. Over the blade pressure side, the streamlines diverge outward to reach the blade tip. For higher blade setting $\beta = 30^\circ$, the stagnation line (Figure 9B) shifts from the blade suction side and the deflection of streamlines towards the blade leading edge is less pronounced and the same for flow recirculation in hub corners. The pressure difference between

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**Figure 8**  A, Unit power vs unit speed. B, Hydraulic efficiency vs unit speed
FIGURE 9 Flow structures for vane opening of 20° and: (A) $\beta = 20^\circ$; (B) $\beta = 30^\circ$

pressure side and suction side induces a leakage flow in the clearance between the blade tip and casing and creates a jet that separates from the suction side leading to a reduction in the local static pressure close to the blade tip from suction side. The leakage flow leads to vortex shedding from the blade and swirling into tip leakage vortex presenting a longwise wavy shape at the clearance. For larger vane opening, the radial pressure gradient increases in size progressively and the center of associated vortex moves away.

The pressure coefficient $C_p = (p - p_{ref})/\frac{1}{2} \rho W_{ref}^2$ is a good indicator of the hydrodynamic loading. $p_{ref}$ and $W_{ref}$ are respectively static pressure and relative velocity upstream the blade. Figure 10 plots $C_p$ around the blade at midspan,
showing a strong dependence on the blade setting and vane opening. For a blade setting from 20° to 30°, the area equivalent to pressure loading becomes larger and similarly for the power. This variation can be explained by the displacement of the stagnation point leading to a rapid flow deceleration over the fore of blade pressure side. At the trailing edge, the flow direction reverses due separations.

Figure 11 illustrates the influence of blade setting and vane opening on the hydraulic performances at constant rotational speed \( N = 2000 \text{ rpm} \) and flow rate \( Q = 36.3 \text{ L/s} \). By changing the vane opening from 20° to 30°, the tangential velocity and the produced work increase, but above this limit there is flow separation and the hydraulic efficiency drops. On the other hand, the increase in blade setting from 20° to 30° increases \( P_{11} \) and \( \eta_h \) about 25% and 15%, respectively. The blade setting \( \beta = 30° \) and vane opening 20-30° seem offering the best operating conditions.

The interdistance is a parameter influencing the vane-rotor interactions and the hydraulic performances. The steady flow simulations were carried out for interdistance varied in between 20 and 42mm, while the turbine operated at the nominal conditions. Figure 12 plots the contours of turbulent kinetic energy at midspan, revealing that for a short distance the wake impinges on the leading edge and overlaps the blade while the turbulent kinetic energy keeps high owing to low turbulent diffusion. For a larger interdistance, the wake is attenuated before reaching the blade and, therefore, the turbulent kinetic energy is dissipated by diffusion. The wake impinges the blade pressure side farther from leading edge. Figure 13 shows that larger interdistance induces a drop in power and hydraulic efficiency about 3.5% and 1.67%, respectively. Below an interdistance of 30mm, the effect of interdistance is mainly due to vane-blade potential interaction and the vane wake and the secondary flows convected upstream blade. Above 30mm, the drop in performance is due to the process of diffusion. An interdistance of 20mm seems producing better performances, but to confirm the final value smaller interdistances should be simulated.
6 | UNSTEADY FLOW ANALYSIS

The transient relative motion on each side of GGI connection was simulated and the interface position updated at each
time step. The high-resolution advection scheme and second-order backward Euler transient scheme were used. For every
time step, 10 internal coefficient loops were selected and the pressure was monitored at the same spatial locations. In this
section, the expected dominant modes and frequencies and the sequence of interactions were determined analytically,
followed by a characterization of the pressure fluctuations computed and recorded at several monitor points and lines.

6.1 | Expected modes of interactions

The components interactions are important sources of pressure fluctuations and vibrations in hydromachinery. It is, there-
fore, useful to know a priori the modal frequencies to identify the resonance leading to operation instability and fatigue
of blades. For a number of blades \( N_r \) and vanes \( N_s \), after an entire revolution each blade receives \( N_s \) times the same fluid
force.\(^{42}\) The blade takes \( 2\pi/\Omega \) for completing an entire revolution and receives a repetitive force with a period \( T_s = 2\pi/\Omega N_s \),
therefore, the resulting force over the blades related to RSI could be described by the Fourier series,\(^{26}\) as follows:

\[
f_n = \sum_i F_i, \sin(iN_s\Omega t - \phi_i). \tag{4}
\]

During an entire revolution, each of \( N_r \) blades interacts with all \( N_s \) vanes so that there are a total of \( N_r N_s \) interac-
tions separated by an angle \( \Delta \theta_{RSI} = (1/N_s - 1/N_r) \) and a time \( \Delta t_{RSI} = 2\pi/(N_r N_s \Omega) \). The phase difference between two
consecutive interactions is constant for the same harmonic.\(^{43}\) For a given frequency \( f = \Omega/2\pi \) occurring after an angular
rotation equivalent to \( \Delta t_{RSI} \), the order of interaction depends on the first position of blade.\(^{26,27,44}\) The nonuniform flow
field at the exit of vanes caused by wakes and blade loading generates a periodic flow pattern.\(^{45}\) The induced pressure
wave rotates at the same runner speed and thus the characteristic pressure oscillation has a frequency \( f_s = m \Omega N_r \) at every
stationary point and \( f_r = n \Omega N_s \) at every rotating point. The common action of the rotating and stationary fields gives
rise to \( N_r N_s \) pressure impulses per a revolution.\(^{42}\) The flow field leaving the distributor vanes (Figure 14) is characterized
by a velocity defect and the pressure field attached to the runner blades induces distortions to the incoming flow field.

The pressure field combining both the vanes and blades pressure fields is characterized by a modulation process representing the interactions which by considering \( \theta_s = \theta_r + \Omega t \) may be expressed according to Franke et al.\(^{44}\) by:

\[
p_{m,n}(\theta_s, t) = \frac{A_{mn}}{2} \cos [mN_r \Omega t - (mN_r - nN_s)\theta_s + \phi_n - \phi_m] + \frac{A_{mn}}{2} \cos [mN_r \Omega t - (mN_r + nN_s)\theta_s - \phi_n - \phi_m].
\]  

(5)

where \( A_{mn} \) is a combined pressure amplitude due to interaction of each harmonic. This pressure field presents two diametrical pressure modes \( k_1 \) and \( k_2 \), which indicate the number of high-pressure and low-pressure regions for a frequency component in the circumferential direction\(^{46}\):

\[
k_1 = mN_r + nN_s \quad \text{and} \quad k_2 = mN_r - nN_s.
\]  

(6)

**TABLE 2**  Expected diametrical mode numbers and frequencies

| Mode | \( m \) | \( n \) | \( k_1 \) | \( k_2 \) | \( \Omega_1/\Omega \) | \( \Omega_2/\Omega \) | \( f_s/f \) | \( f_s \) | \( \phi_1/\Omega \) | \( \phi_2/\Omega \) | \( f_r/f \) | \( f_r \) |
|------|------|------|------|------|------|------|------|------|------|------|------|------|
| 1    | 2    | 12   | -4   |      | 0.333| -1   | 4    | 133  | 0.667| -2   | 8    | 267  |
| 1    | 3    | 16   | -8   |      | 0.25 | -0.5 | 4    | 133  | 0.75 | -1.5 | 12   | 400  |
| 2    | 1    | 12   | 4    |      | 0.666| 2    | 8    | 267  | 0.333| 1    | 4    | 133  |
| 2    | 3    | 20   | -4   |      | 0.4  | -2   | 8    | 267  | 0.6  | -3   | 12   | 400  |
| 3    | 1    | 16   | 8    |      | 0.75 | 1.5  | 12   | 400  | 0.25 | 0.5  | 4    | 133  |
| 3    | 2    | 20   | 4    |      | 0.6  | 3    | 12   | 400  | 0.4  | 2    | 8    | 267  |
| 4    | 1    | 20   | 12   |      | 0.8  | 1.333| 16   | 533  | 0.2  | 0.333| 4    | 133  |
| 4    | 2    | 24   | 8    |      | 0.666| 2    | 16   | 533  | 0.333| 1    | 8    | 267  |
| 5    | 1    | 24   | 16   |      | 0.833| 1.25 | 20   | 667  | 0.167| 0.25 | 4    | 133  |
| 5    | 2    | 28   | 12   |      | 0.714| 1.666| 20   | 667  | 0.286| 0.667| 8    | 267  |
| 6    | 1    | 28   | 20   |      | 0.857| 1.2  | 24   | 800  | 0.143| 0.2  | 4    | 133  |
| 6    | 2    | 32   | 16   |      | 0.75 | 1.5  | 24   | 800  | 0.25 | 0.5  | 8    | 267  |
| 7    | 1    | 32   | 24   |      | 0.875| 1.166| 28   | 933  | 0.125| 0.167| 4    | 133  |
| 7    | 2    | 36   | 20   |      | 0.777| 1.4  | 28   | 933  | 0.222| 0.4  | 8    | 267  |
| 8    | 1    | 36   | 28   |      | 0.888| 1.143| 32   | 1067 | 0.111| 0.143| 4    | 133  |
| 8    | 2    | 40   | 24   |      | 0.8  | 1.333| 32   | 1067 | 0.2   | 0.333| 8    | 267  |
| 9    | 1    | 40   | 32   |      | 0.9  | 1.125| 36   | 1200 | 0.1   | 0.125| 4    | 133  |
| 9    | 2    | 44   | 28   |      | 0.818| 1.285| 36   | 1200 | 0.182| 0.286| 8    | 267  |
| 9    | 3    | 48   | 24   |      | 0.75 | 1.5  | 36   | 1200 | 0.25 | 0.5  | 12   | 400  |

---
TABLE 3  Principal spatial modes

| Mode | Behind TE of vane | Near LE of blade | Behind TE of blade |
|------|-------------------|------------------|-------------------|
|      | Span       | Span       | Span       |
|      | 10% 50% 90% | 10% 50% 90% | 10% 50% 90% |
| Mode | Ω_m (rpm) | Amplitude (Pa) | Ω_m (rpm) | Amplitude (Pa) | Ω_m (rpm) | Amplitude (Pa) |
| 1    | 2000 2000 2000 | 2072 2978 1120 | 2000 2000 2000 | 2000 2000 2000 | 2000 2000 2000 |
| 2    | 2000 2000 2000 | 6840 1843 3470 | 2000 2000 2000 | 9334 9471 1613 | 9471 1613 1613 |
| 3    | 2000 2000 2000 | 2807 1346 1486 | 2000 2000 2000 | 4906 537 88 555 |
| Others | <25000 | 100% | 100% | 100% |

The signs of the diametrical mode numbers $k_1$ and $k_2$ indicates that the diametrical mode is rotating in the same direction as the runner when is positive, while a negative value indicates a counter-rotation with a spinning speed $(\omega, \dot{\omega})$.42

$$\omega_1 = mN_r \Omega/k_1 \quad \text{and} \quad \omega_2 = mN_r \Omega/k_2.$$  \hspace{1cm} (7)

$$\omega'_1 = nN_s \Omega/k_1 \quad \text{and} \quad \omega'_2 = nN_s \Omega/k_2.$$  \hspace{1cm} (8)

As noted, the low amplitudes are expected for high $k$ values because of high harmonic number. As a result, $k_1$ is usually not relevant. The runner blades 1, 2, 3, and 4 are first excited by interference with vanes’ wakes. Runner blades are excited in phase and induce a vibration having a mode with two diametrical nodes. One blade interacts with another vane after $4\Delta t_{RSI}$, and the blade 1 interacts again with the vane 1 after 16 $\Delta t_{RSI}$, where $\Delta t_{RSI} = 2\pi/(N_rN_s\Omega) = 1.875 \text{ milliseconds}$. The runner provokes a pressure wave rotating at the same runner speed, inducing a characteristic pressure oscillation of frequency $f_s = m\Omega/2\pi N_r$ and $f_r = n\Omega/2\pi N_s$ at every stationary/rotating point. The pressure field presents two diametrical pressure modes $k_1$, $k_2$ indicating the number of high- and low-pressure regions for the frequency component in the circumferential direction estimated in Table 2. The highest diametrical positive mode of RSI occurs for $k_2 = 4$ at the successive expected frequencies of 267, 400, 533, 667, 800, 933, 1067, and 1200 Hz. The highest diametrical negative modes occur for $k_2 = -4$ for the successive frequencies of 133, 267, 400, 533, 667, 800, 933, 1067, and 1200 Hz. This sequence of expected pressure mode shapes and frequencies reveals the dominant modes but does not give accurately their amplitudes.

6.2 Analyses of unsteady pressure field

Unsteady flow simulations were first carried out at the nominal point (2000 rpm and 36.3 L/s), for two runner rounds and a step rotation of 1.5° equivalent to a time step of 125 microseconds. The temporal pressure fluctuations were recorded at
FIGURE 16  Spatial pressure fluctuations and fast Fourier transform, at rotational speed of 2000 rpm, recorded along a line after the vane trailing edge at different spans: (A) 10%, (B) 50%, (C) 90%
several monitor points (Figure 15) located after the vane trailing edge and the leading and trailing edges of the blade and at successively span 10, 50, 85, and 95%, whereas the spatial fluctuations of pressure were recorded after two rounds of runner along circumferential lines at span 10, 50, and 90% successively.

The results of fast Fourier transform (FFT) performed on the recorded static pressure signals are summarized in Table 3, revealing the main spatial modes. One runner revolution was divided into bins of identical size of $7.2^\circ$. The mode defined by harmonic 4 is clearly the dominant and its multiples which are related to the potential effect manifesting in the form of rotating pressure waves propagating upstream and downstream. Figures 16-18 show the spatial pressure fluctuations and the corresponding spectrums, which the amplitudes of harmonics and multiples depend on the recording locations. Behind the vanes, it is possible to discern (Figure 16) four main low-pressure zones corresponding to the four vanes. Near the hub, there are secondary peaks associated with the hub corner vortex. Figure 17 shows secondary peaks appearing near the shroud related to the vortex emanating from upstream vane. Downstream of runner blades (Figure 18), there are four low-pressure zones corresponding to the four blades. The rounded parts of low-pressure zones correspond to the loading on the blade suction side, while the spike secondary peaks are originated from the tip gap flow and associated tip gap vortex. Typically, the highest peak values are encountered near the leading edge and upper of blade. The flow unsteadiness is characterized by other weak harmonics occurring at span 90% related to the wake and tip-vortex interaction. The vanes’ wakes are the sources of unsteady phenomena added to the wake and tip-vortex interaction and the perturbation generated by the vortex migrating inboard and convected through the transition plane between the runner bulb and draft tube.

The next FFT analysis of spatial pressure fluctuations revealed strong dependence upon the rotational speed, as plotted in Figures 19-21, for the low-speed of 1500 rpm and high-speed of 2500 rpm. As noticed, the mechanisms of RSI originating from the interactions of coherent structures downstream and upstream of blades intensify with the rotational speed. The mode defined by harmonic 4 and its multiples are dominant and represent the potential effect. Also, there are discontinuities in the static pressure due to wakes segmentations that altered the distinguished amplitudes of harmonics. When probing near the shroud from blade trailing edge, the potential interaction seems having greater amplitude than at midspan as this could be attributed to the formation of secondary flows.

FFT analysis of the temporal pressure fluctuations obtained at the nominal point and recorded at several monitor points: after the trailing edge of vane, before the leading edge, and after the trailing edge of blade. The static pressure data were uniformly sampled allowing the results to be treated with the standard FFT. The Hanning window was applied on the fluctuating parts of static pressure and reduced from spectral leakage, and the bins number was set by default. A transitory phase occurred first; thereafter, the pressure fluctuations became practically periodic around an average. The propagated pressure wave upstream and downstream of the turbine parts is revealed by the peak of BPF = \( \frac{\Omega}{60}N_r = 133.3 \) Hz. As shown in Figure 22, the frequency of the highest RSI frequencies such as 266.7, 400, 533.3, 666.6, 799.9, and 933.2 Hz are the most evident. The upper region of vane is the most affected by RSI. Farther from the vane trailing edge, the peak representing the average value of the transitory amplitude decreases drastically. Being closer to the runner, the effects of RSI are more important but lesser downstream compared to the pulsating pressure itself. Near the hub and shroud corners of blade, the secondary flows tend to increase the pressure fluctuations locally and, therefore, the dominant peaks. At the top corner from the blade leading edge, the high unsteadiness is typically of the wake and tip-vortex structures. Also, there are other peaks of weak amplitudes related to the interaction between the convected wake and the shedding vortices at upper of blade referred to the vortices precessions correlated independently from the rotational speed. The blade tip-vortex interaction with the draft tube is revealed through a modulated frequency intermediate between two successive harmonics. Table 4 provides the main temporal modes where those of less amplitude are omitted. The expected fundamental frequencies are 133 and 267 Hz, whereas the CFD predicted values are 133.3 and 266.7 Hz, and so forth. One may conclude that only the main frequencies are predicted not the amplitudes. To illustrate the effect of varying operating point, other unsteady simulations were carried out at the points (1500 rpm and 32 L/s) and (2500 rpm and 40 L/s). For the low- and high-rotational speed, FFT analysis shows evidence of peaks related to the potential effect, which are characterized by BPF = 100 Hz for the low-speed of 1500 rpm and 166.67 Hz for the high-speed of 2500 rpm. The first peak appearing in the spectrum describes the transitory mode. The mechanisms of interactions of coherent structures downstream and upstream of runner blade seem to intensify at high-rotational speed. The region the most affected by the potential interaction is seen upward of runner blade. Near the shroud, the amplitudes of main frequencies are higher due to the secondary flows and wake and tip-vortex interaction typical to this low aspect-ratio runner blade. Furthermore, the wakes and secondary flows are shown to persist downstream of runner and play a significant role in flow unsteadiness. FFT analysis of temporal pressure fluctuations is relevant to avoid the natural frequency of vibration, which over time may cause high cycle of fatigue and blade failure.
FIGURE 17 Spatial pressure fluctuations and fast Fourier transform, at rotational speed of 2000 rpm, recorded along a line before the leading edge of the blade at different spans: (A) 10%, (B) 50%, (C) 90%
FIGURE 18  Spatial pressure fluctuations and fast Fourier transform, at rotational speed of 2000 rpm, recorded along a line after the blade trailing edge at different spans: (A) 10%, (B) 50%, (C) 90%
**Figure 19** Fast Fourier transform of spatial pressure fluctuations, at rotational speed of 1500 rpm (left) and 2500 rpm (right), recorded along a line after the vane trailing edge at span: (A) 50\%, (B) 90\%.

**Figure 20** Fast Fourier transform of spatial pressure fluctuations, at rotational speed of 1500 rpm (left) and 2500 rpm (right), recorded along lines before blade leading edge at span: (A) 50\%, (B) 90\%.
**Figure 21** Fast Fourier transform of spatial pressure fluctuations, at rotational speed of 1500 rpm (left) and 2500 rpm (right), recorded along lines after the blade trailing edge at span: (A) 50%, (B) 90%

**Figure 22** Fast Fourier transform of temporal pressure fluctuations, operating at nominal speed of 2000 rpm, recorded after blade TE at span: (A) 10%, (B) 50%, (C) 85%, (D) 95%
TABLE 4 Principal temporal modes

| Mode  | Span | Behind TE of vane | Near LE of blade | Behind TE of blade |
|-------|------|-------------------|------------------|-------------------|
|       |      | 10% 50% 85% 95%   | 10% 50% 85% 95%  | 10% 50% 85% 95%  |
| Mode 1| Frequency (Hz) | 16.67 16.67 16.67 16.67 | 16.67 16.67 16.67 16.67 | 16.67 16.67 16.67 16.67 |
|       | Amplitude (Pa) | 450 512.6 1034 1019 | 161.8 433 406.9 410.2 | 152.5 161 189.3 171.7 |
| Mode 2| Frequency (Hz) | 100 100 100 100 | 100 100 100 100 | 100 100 100 100 |
|       | Amplitude (Pa) | 86.97 88.57 96.66 102.1 | 81.18 86.93 123.5 127.4 | 339.1 153 |
| Mode 3| Frequency (Hz) | 133.3 133.3 133.3 133.3 | 133.3 133.3 133.3 133.3 | 133.3 100 133.3 133.3 |
|       | Amplitude (Pa) | 388.3 400.8 415.7 402.2 | 487.6 580.4 761.1 788.1 | 2776 492.1 2896 1366 |
| Mode 4| Frequency (Hz) | 266.7 266.7 266.7 266.7 | 266.7 266.7 266.7 266.7 | 266.7 133.3 266.7 133.3 |
|       | Amplitude (Pa) | 66.37 66.41 63.64 64.12 | 95.49 113.9 266.7 118 | 4122 1366 |
| Mode 6| Frequency (Hz) | — — — — | — — — — | 266.7 266.7 266.7 |
|       | Amplitude (Pa) | — — — — | — — — — | 1127 — 1893 489.9 |

The frequency analysis showed that the dominant frequencies in the acquired signals are the runner rotation and guide vane passing frequencies. The presence of the runner frequency and harmonics illustrates the flow complexity through the blade channels of this model of a Kaplan turbine. According to the measurements carried out by Trivedi et al.\textsuperscript{27} in the case of a Francis turbine, the static pressure exhibited a neat pattern at BEP and the occurrence of only one peak at the vane passing frequency explained by the small spacing between the vane and the runner that amplified RSI. However, in this model of a small Kaplan turbine, the larger spacing between the vane and the runner reduced RSI amplitude, and thus FFT shows two frequencies of comparable amplitudes. Indeed, the different harmonics of runner BPF with frequencies higher than the vane’s passing frequency may increase the possibility of resonance. Besides, the vanes’ wakes spreading out downstream are subject to more rotation, leading to further complexity of the flow. The wake region has a decreased velocity while in the free region of wake the velocity is higher; and this difference in velocities results in different flow incidences to the runner blades and therefore fluctuation in static pressure distribution. The vanes’ wakes are weaker close to the runner hub, hence the flow nonuniformity is lesser in this region compared with the runner shroud since the distance between the vane and runner blade is greater near the hub. This large distance may result in higher dissipation of wakes close to the hub lowering the fluctuations by reaching the runner blade and subsequently the amplitudes of pressure fluctuations acquired close to the shroud are stronger than those acquired near the hub.

7 | CONCLUSION

The characterization of hydraulic performances of this small model of a Kaplan turbine has revealed moderate hydraulic efficiency. This could be explained by the size effect, relatively large gaps, and high-rotational speed. The nominal operating point is shown to correspond to blade setting of 30° and vane opening of 20°. The variation in interdistance between 20 and 50 mm caused the power and efficiency to decrease about 3.5% and 1.7%, respectively. In order to forecast the pressure fluctuations in the complete passages of this Kaplan turbine model, URANS simulations were carried out at several operating points. FFT analysis of the spatial pressure fluctuations has revealed lobed structures of pressure waves, which are characterized mainly by the harmonic 4 and its multiples. The amplitudes of temporal pressure fluctuations due to blade passing events are shown to vary significantly with the monitor positions and operating conditions. For this small runner of low aspect-ratio blade, the modes of unsteadiness related to the wakes and secondary flows are significant, while the large spacing between vane and runner tends to reduce RSI amplitudes. Further details about RSI mechanisms could be revealed by using a refined grid and adopting LES turbulence model and a smaller time step in the flow simulations.
CONFLICT OF INTERESTS

The author(s) declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

AUTHOR CONTRIBUTIONS

Adel Ghenaiet contributed to the conceptualization, formal analysis, investigation, methodology, project administration, resources, software, supervision, validation, writing of the original draft, review, and editing. Mustapha Bakour contributed to the data curation, formal analysis, investigation, software, and visualization.

NOTATIONS

| Symbol | Description                                      |
|--------|--------------------------------------------------|
| BPF    | blade (or vane) passing frequency                |
| $C_p$  | pressure coefficient                             |
| $d$    | external diameter of runner (m)                  |
| $k_1, k_2$ | excited diametrical modes                      |
| $f$    | frequency (Hz)                                   |
| $H$    | head (m)                                         |
| $g$    | gravity constant (9.81 m/s²)                     |
| $N$    | vane/blade count                                 |
| $N_\text{r}$ | rotational speed (rpm)                         |
| $p$    | static pressure (Pa)                            |
| $P$    | power (W)                                        |
| $P_{11}$ | unit power (W)                                  |
| $Q$    | volume flow rate (L/s)                          |
| $Q_{11}$ | unit discharge (L/s)                            |

GREEK LETTERS

| Symbol | Description                                      |
|--------|--------------------------------------------------|
| $\alpha$ | vane opening angle (degree)                     |
| $\beta$ | blade setting angle (degree)                    |
| $\eta$  | hydraulic efficiency                             |
| $\Omega$ | angular velocity (rad/s)                        |
| $\omega_{11}$ | unit speed (rad/s)                             |
| $\omega$ | angular velocity (rad/s)                        |
| $\omega_{1}, \omega_{2}$ | mode spinning speed (rad/s)                   |
| $\rho$  | density (kg/m³)                                 |
| $\mu$   | dynamic viscosity (kg m⁻¹ s⁻¹)                  |
| $\nu$   | kinematic viscosity (m² s⁻¹)                    |
| $\theta$ | angular position (rad)                          |

SUBSCRIPTS

| Subscript | Description                                      |
|-----------|--------------------------------------------------|
| $r$       | rotating/runner                                  |
| $s$       | stationary/vane                                  |

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