CFD prediction of unsteady wicket gate-runner interaction in Francis turbines: A new standard hydraulic design procedure

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Nomenclature

φ	*Phase angle*	*[°]*	f_r	*Runner rotation/frequency*	*[Hz]*
n_{bl}	*Number of blades*	*[-]*	f_{wg}	*Wicket gate passing frequency*	*[Hz]*
n_{wg}	*Number of wicket gates*	*[-]*	f_{bl}	*Blade passing frequency*	*[Hz]*
$T_{z.bl}$	*Unsteady/dynamic blade torque*	*[Nm]*	A_s	*Area at sliding interface*	*[m²]*

1. Introduction

In hydropower generation the push towards lower cost and more flexible output leads to the demand for more compact machines and a larger required operating range for new and upgraded turbines. As a result hydrodynamic effects in turbines such as the draft tube rope at part load and the wicket gate-runner blade interaction become important issues in the design of these machines.

Particularly in high head Francis runners the presence of vibrations from the rotor-stator interaction between wicket gates and runner blades is a known phenomenon. In the past, some of the vibrations resulting from this interaction have been attributed to a phase resonance phenomenon in the radial passages of the turbines e.g. [1], [2]. According to these publications strong machine vibrations occur because pressure waves generated by the interaction between individual blades and wicket gates propagate circumferentially in the radial spaces. If these waves are in phase, they amplify and can cause problematic levels of vibration. Doerfler [2] recognizes that in some cases where site observations report strong vibrations no correlation can be made with phase resonance. In those cases he supposes the wicket gate-runner blade interaction to be directly responsible. More recently the hydrodynamic wicket gate-runner blade interaction has been found to be directly responsible for blade cracking [3].

Extensive work on rotor-stator interaction has been done in the field of acoustics. In thermal turbo machines determining which modes form and propagate, therefore contributing to the acoustic field, is of primary concern, e.g. [4]. For the structural response of hydraulic turbines to rotor-stator interaction the modes and mode shapes are as important as in machine acoustics, which has been studied extensively by Tanaka [5]. More recently [6] have reported on a method to predict the forced response of a hydraulic turbine due to rotor-stator interaction. Arndt et al [7] have investigated the interaction between a vaned diffuser and the impeller of a hydraulic pump. They performed unsteady pressure measurements and discovered large pressure amplitudes, which decreased significantly with an increasing radial gap. Dring et al [8] have performed an extensive experimental investigation of the rotor-stator interaction in a gas turbine stage and discovered large pressure amplitudes in both the rotor and the stator channels.

Steady state Computational Fluid Dynamics (CFD) analyses have been an integral and well validated part of the hydraulic design procedure at GE Energy Hydro for many years now [9]. However, only recent developments in CFD software [11] and affordable computer technology – notably the availability of Linux clusters for parallel computing – have made the simulation of the unsteady flow field possible, opening the door for the prediction and understanding of unsteady flow phenomena, e.g. [10].

In hydraulic turbines there are two aspects to rotor-stator interaction: the shape of the unsteady flow field and the response of the structure to the forcing by the unsteady flow field [6]. The present paper presents a method of simulating the unsteady flow field resulting from rotor-stator interaction. The method is validated by means of unsteady pressure measurements performed on a model scale runner.

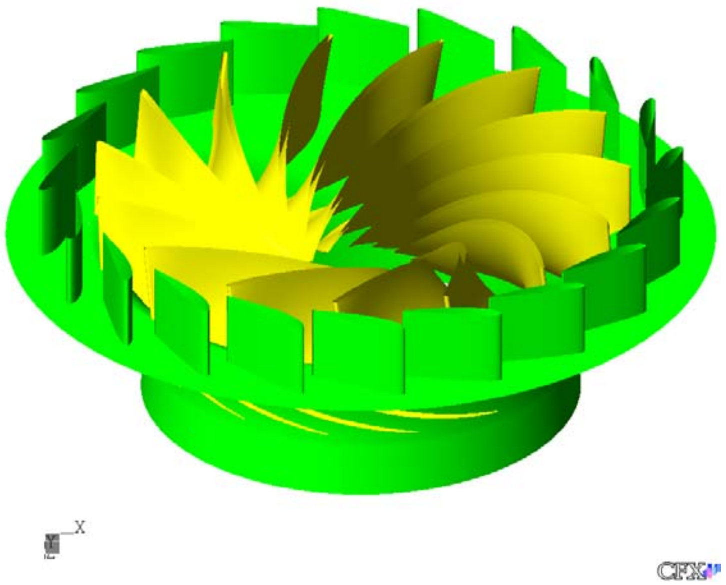

Figure 1: 3D view of the investigated runner and wicket gate geometry

2. Hydrodynamic rotor-stator interaction in Francis turbines

While rotor-stator interaction of wicket gates and runner blades occurs in all hydraulic turbines, only in medium to high head machines, such as the one depicted in **Figure 1**, do the pressure fluctuations become significantly large with respect to stress levels. This is due to the fact that in these machines the velocity at the wicket gate outlet is sufficiently high and the radial gap between the blade rows tends to be small. The hydraulic effect leading to the unsteadiness is primarily a potential flow interaction between the non-uniform flow distribution at the outlet of the wicket gates and the rotating runner blades passing through this flow. Since the flow field in the radial space between the wicket gates and runner is non-uniform circumferentially, both the static pressure and the velocity (velocity magnitude and flow angle) vary circumferentially. This results in three effects on the runner:

- As each runner channel passes through the flow field, it is subject to a varying static inflow pressure. If a phase difference between adjacent channels is present, this results in a varying pressure difference between those channels, i.e. an unsteady load on the blade.
- As each blade passes through the flow field, it is subject to a varying incidence angle. This in itself creates an unsteady load on the blade.
- As each blade passes through the flow field, the magnitude of the velocity passing over the blade varies. This also creates an unsteady load on the blade.

Figure 2 illustrates these contributing factors. The phase shift between two adjacent runner channels is calculated by:

$$\varphi = 360^\circ \left(\frac{n_{wg}}{n_{bl}} - 1 \right) \tag{1}$$

A useful way of representing the effect of unsteadiness on the blades of a runner is the unsteady or dynamic blade torque $T_{z,bl}$. This is the torque an individual blade contributes to the shaft torque as a function of time. On the runner, i.e. in a rotating frame of reference, pressure and any pressure derived fluctuations such as the dynamic blade torque occur at the wicket gate passing frequency $f_{wg}=n_{wg} \cdot f_r$. On the stationary components upstream of the runner the pressure fluctuations occur at the blade passing frequency $f_{bl}=n_{bl} \cdot f_r$. The pressure fluctuations that form in the radial space between wicket gates and runner propagate in all directions including upstream and downstream.

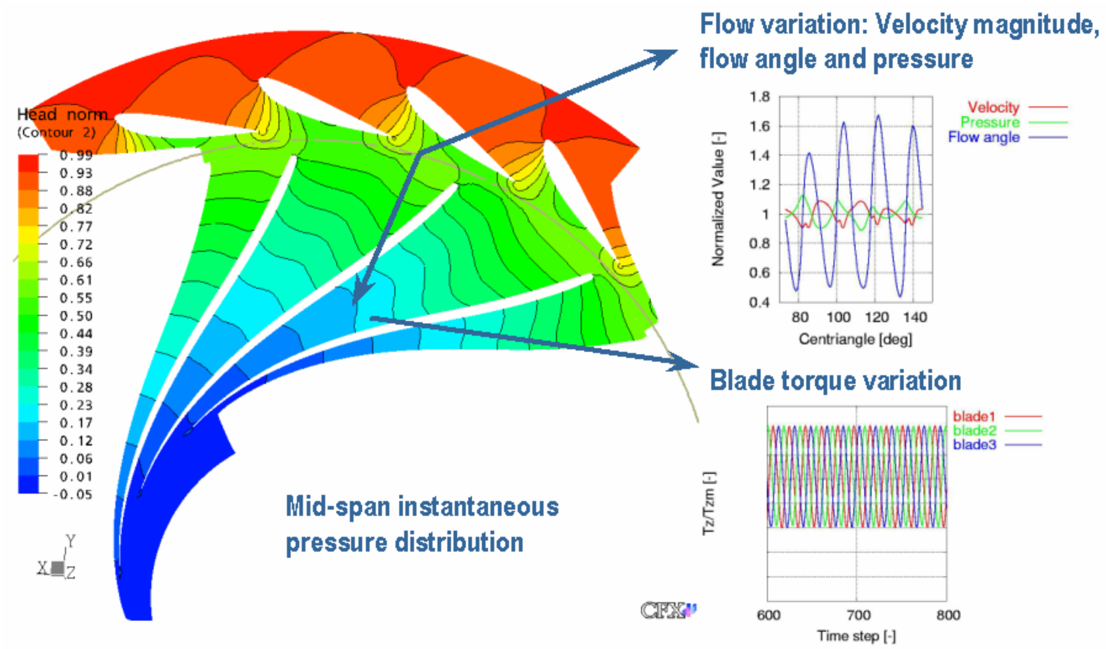

Figure 2: Rotor stator interaction between blades and wicket gates of a Francis turbine

3. CFD methodology and results

General

At GE Energy Hydro we use the commercial CFD software CFX-5 for our unsteady CFD analyses. The transient rotor stator interface implementation of CFX allows for the runner mesh to rotate relative to the wicket gates during calculation thus permitting the simulation of the 3D unsteady flow that results from the rotation of the runner.

Unlike steady state calculations with stage or frozen rotor interfaces, unsteady rotor-stator calculations theoretically require an identical periodicity for the steady and the rotating domain, i.e. the ratio of the overlapping areas at the sliding interface has to be $A_{s.stat}/A_{s.rot}=1$. Seeing that there are no Francis turbines with identical numbers of wicket gates and blades, it is never possible to do an unsteady rotor-stator analysis with a single wicket gate and a single runner channel. Some n_{wg}-n_{bl} combinations allow for a reduced circumferential section of the turbine to be analyzed while maintaining an identical periodicity on both the wicket gate and the runner mesh, see **Figure 2**. We have found that the range of $A_{s.stat}/A_{s.rot}=0.99 \ldots 1.01$ is acceptable and gives dynamic torque values that are very close to the ones obtained with a full runner calculation. For example in the case of 24 wicket gates and 17 blades 7 wicket gate channels and 5 runner channels can be calculated with $A_{s.stat}/A_{s.rot}=0.99167$. This significantly reduces the mesh size and therefore the calculation time. Unsteady CFD calculations require an initial solution. The best way of obtaining such an initial solution is a frozen rotor calculation on the same mesh. A frozen rotor solution is better suited than a stage solution because it accounts for the relative position of the blades and wicket gates. A smoother start up of the unsteady calculation is the result.

As mentioned before, in Francis turbine rotor-stator interaction viscous effects are secondary. Therefore it is sufficient to perform the CFD calculations with the k-ε turbulence model, and y+-values do not have to be controlled as tightly as in loss calculations. The dynamic torque relative to the mean torque allows the assessment of unsteadiness from the rotor-stator interaction. At GE Energy Hydro we generally perform the analyses at prototype scale. This allows us to use the time dependent pressure distributions directly for our stress analyses. As a standard for all our CFX calculations we use the CFX high-resolution differencing scheme.

Mesh and time step dependence

Since the wicket gate-blade interaction is primarily a potential flow field interaction, it is possible to predict the effect with relatively coarse meshes. However, **Figure 3** shows clearly that there is a dependence of the predicted dynamic torque on the mesh size. There is still a 4% reduction of the dynamic blade torque from the 1387k-node mesh to the 2641k-node mesh. A further refinement of the mesh is needed to be able to conclude that a plateau has been reached. Since the total mesh size depends on how many runner and wicket gate channels have to be included in the analysis, very large total meshes can result, if large meshes per channel are used.

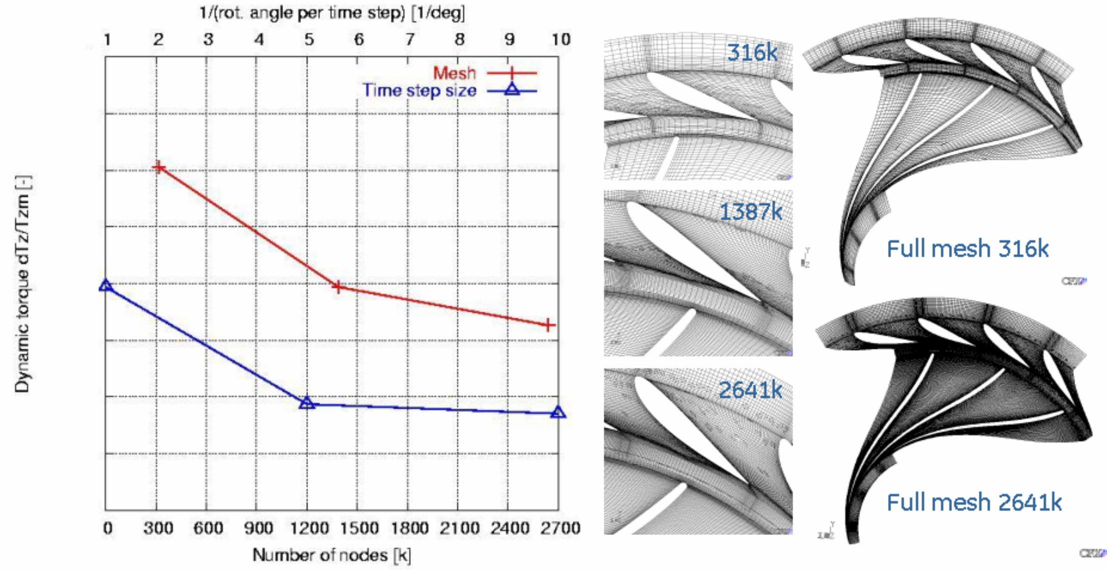

Figure 3: Mesh size and time step size dependence of dynamic blade torque

The time step size of the unsteady calculation has to be chosen according to the period of the dynamic effect to be resolved. In the case of wicket gate-blade interaction this period depends on the number of wicket gates. Considering that a minimum of 10-20 time steps are necessary to resolve a period this leads to about 1° rotation per time step. As we can see in **Figure 3**, the predicted dynamic torque is as strongly dependent on the time step size as it is on the mesh size. At 0.2° rotation per time step the dynamic torque has reached a plateau and a further reduction of the time step size does not warrant the increase in computational effort.

Results

The time dependant pressure can be monitored at selected locations during an unsteady calculation. Examples of pressure signals monitored in such a way are shown in **Figure 4**. It is clear that relatively large pressure fluctuations are possible. As mentioned above, the best indicator for the strength of the rotor-stator interaction is the dynamic blade torque. Also in **Figure 4** we can see the evolution of the blade torque from the start up of the unsteady calculation to the switch to a smaller time step. Several periods are required before a fluctuation with a steady mean value is obtained both at start up and after the switch to the smaller time step. A Fourier transform of the time signals can be used to identify frequencies and amplitudes.

With CFXAMP, a GE proprietary program, a 3D Fourier transform for a given frequency can be performed on the CFD results. In this way pressure amplitudes and phases can be determined in the entire CFD domain. Example plots of these values are given in **Figure 5**. We can see that the amplitude is the highest at the leading edge and declines sharply along the blade towards the trailing edge. The phase is virtually constant within each runner channel and assumes a value according to **Eq. (1)**. This representation of the data permits the identification of local pressure amplitude maximums.

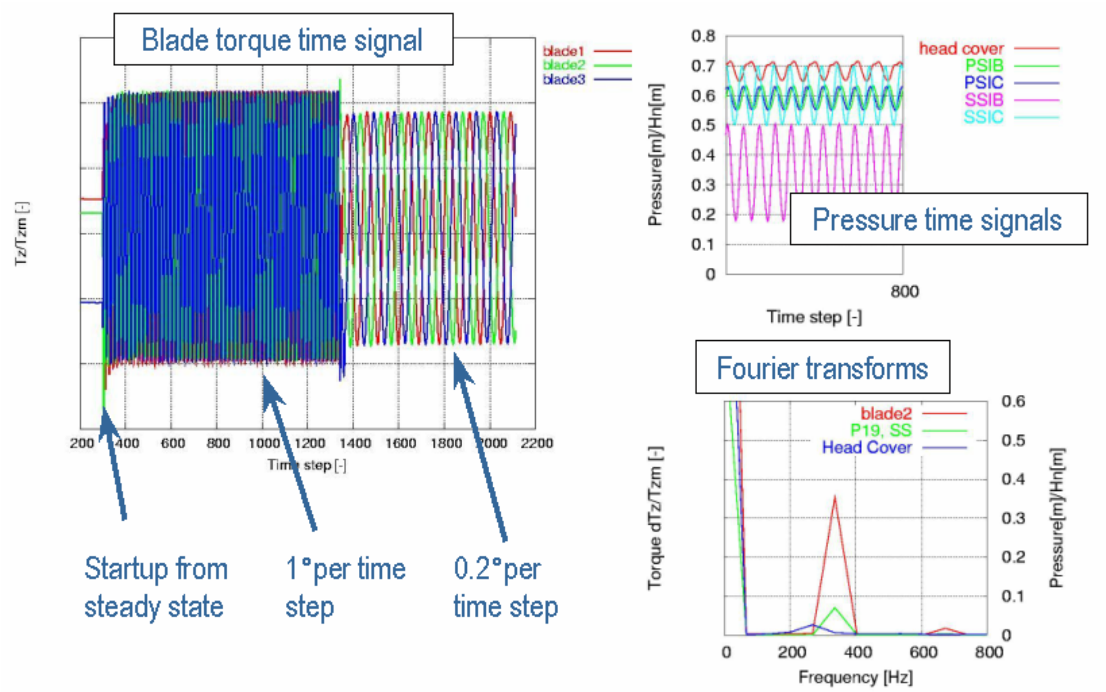

Figure 4: Local monitoring: Blade torque start up and switch of time step size, pressure signals. Fourier transform of blade torque and two pressure signals.

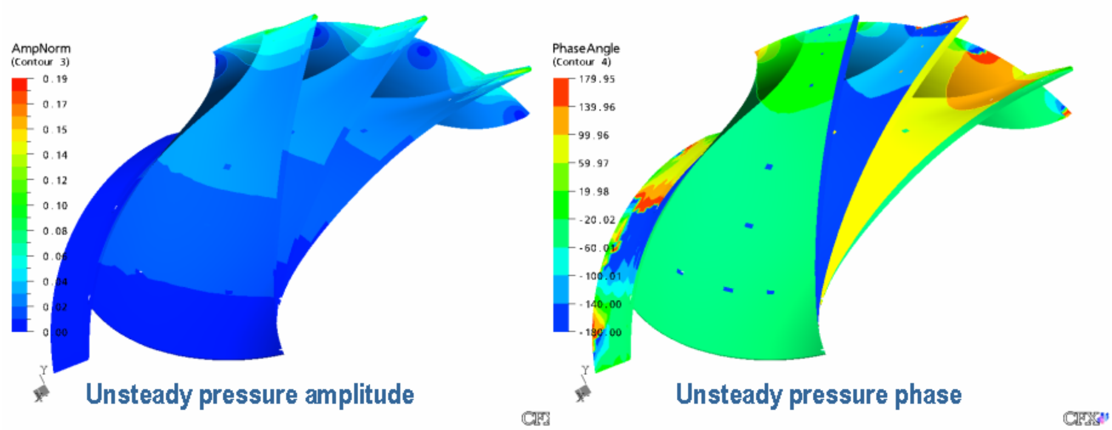

Figure 5: 3D pressure amplitude and phase computed with CFXAMP

4. Experimental technique

The mounting procedure of miniature pressure sensors

The validation of the numerical model is carried out through a collaborative research program between GE Energy Hydro and École Polytechnique Fédérale de Lausanne. Both static and fluctuating parts of the pressure are measured in two hydraulic channels with the help of 32 miniature piezo resistive pressure sensors, spread on the pressure sides of blades 1 and 8 and suction sides of blades 2 and 9, see **Figure 6** and **Figure 9**.

Each sensor has a cylindrical shape with 3 mm diameter and 1 mm height. It is fitted inside a chamber previously drilled in the blade and connects to the outer flow through a small cylindrical pipe of 1 mm diameter and 0.5 mm height. The sensing element is glued into the chamber and protected by filling the connecting pipe with a plastic compound having the same density as water. The 5 wires of the sensor are then led into the annular chamber and connected to the signal conditioning electronics located in the runner crown through a cable path previously drilled in the blade. Such a mounting procedure allows a significant improvement of the measurement quality (precision, hysteresis) and makes it possible to flush mount a pressure sensor in a highly curved area of a blade having a thickness as low as 2 mm without any geometry alteration (see reference [12]). Obviously, this is of major importance with regards to the topic of the present study since any modification in the hydraulic profile would have an unpredictable influence on the fluctuating pressure field. Once the pressure sensors are mounted, the runner is assembled and the cables are introduced into the annular chamber through specific holes drilled in the base of the instrumented blades. **Figure 6** illustrates the runner during its assembly.

Figure 6: Views of the instrumented runner during its mounting

The onboard electronics

The onboard electronics for signal conditioning is made of 32 preamplifiers and anti-aliasing filters and is fitted in a box close to the runner as shown on **Figure 7**. The conditioning electronics is connected to eight acquisition boards located on the top of the turbine shaft. Each of these boards has four channels input and four 12-bits A/D converters. The maximum sampling frequency is 20 kHz and the memory depth allows the storage of 64k-samples per channel. A host computer is used to monitor this 32-channel acquisition system through an ArcNet based communication network. A 4-channel slip ring located at the top of the turbine shaft ensures the power supply as well as data transmission. Transfer rates of digitized signals as high as 1.5 Mbits/s are thereby reached.

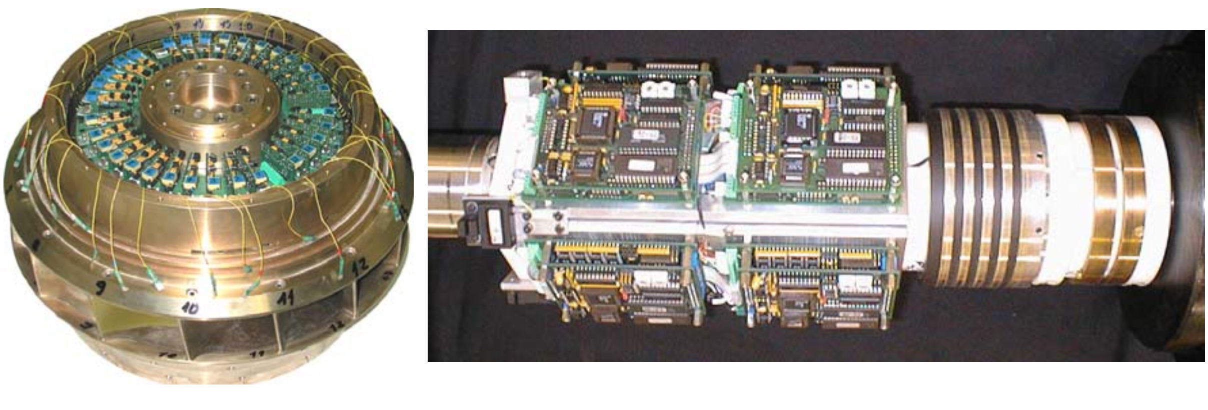

Figure 7: Left: Instrumented runner with onboard conditioning electronics in its crown
Right: Instrumented shaft with 8 acquisition boards and multi-channel slip ring

The synchronization of the data sampling over active boards is achieved through a master-slave scheme. After the slave boards are all armed, the master board is armed in its turn. Once the master is triggered by the onboard tachometer signal, it outputs a TTL signal to trigger onboard slave modules.

Thus, all active modules are synchronously triggered within a 5 µs time frame. The trigger output from the master module is also transmitted to the static part of the turbine through transmitter and receiver diodes. It is then possible to synchronize the onboard acquisition with other acquisition boards located outside the runner. Specific acquisition software is used to fully control the digitizers located in both static and rotating parts. It ensures data transfer and storage.

Calibration

In order to perform the static calibration of pressure transducers, the instrumented runner is placed in a pressurized tank. Voltage outputs of pressure sensors are digitized then averaged and compared to the readings of a high precision reference pressure transducer. A typical result of such a calibration is given in **Figure 8** where an excellent linear response is observed. The measurement error, plotted in the same figure, is highly related to the mounting procedure of the transducer and found to be less than 0.5 % of the measurement range.

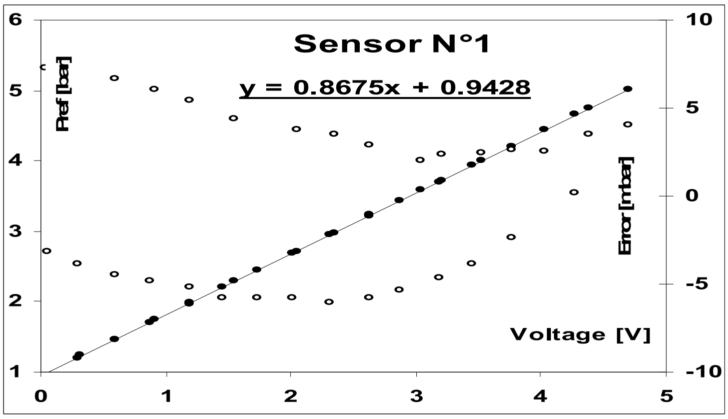

Figure 8: Typical curves resulting from static calibration of embedded pressure sensors

The dynamic calibration of similar sensors has been performed with the use of a spark-generated bubble as a source of high frequency excitation. Such a device allows the deposition of high-density energy in water, which leads to the formation of a plasma, and to an explosive growth and collapse of a vapor cavity. The resulting shock waves are used to excite the pressure sensors in a wide frequency range. A linear response of pressure sensors is thus obtained for frequencies up to 25 kHz (see reference [11]).

5. Comparison of CFD and experimental results

The comparison of the experimental and computational results was performed at the locations of the pressure probes as shown in **Figure 9**. For the comparison, phase-averaged data of the experimental time signals have been used. These, as well as the equivalent computational data, were treated by a Fourier transform to obtain the amplitudes of the harmonic components. The solution from the 1387k-mesh at 0.1° per time step was used for the comparison.

The results in **Figure 9** show excellent agreement in terms of the general shape of the curves. Larger differences occur near the leading edge where the amplitude gradients are very high. Therefore small deviations in the geometric location of the monitoring points have large effects. In this area, due to the very large gradients, an insufficient mesh resolution may also be the cause for the deviation. The average relative deviation for all points is in the order of 10% and drops to values as low as 3% in some locations. This seems large but potential causes for the deviation near the leading edge have been given above. Near the trailing edge values approach zero so that small absolute deviations results in relatively large relative errors. Measurements of unsteady pressures at selected locations of prototype runners indicate comparable levels of correlation with the CFD results.

The dynamic blade torque cannot be measured directly, but it is directly derived from the dynamic pressures and is therefore subject to the same errors. For the purpose of rotor-stator calculations, i.e. the prediction of potential problems due to dynamic pressure fluctuations and their prevention in final machine designs, the degree of accuracy is very satisfactory. Calculations performed with smaller size meshes such as the 316k-node mesh in **Figure 3**, achieve accuracy of approximately 20% and less.

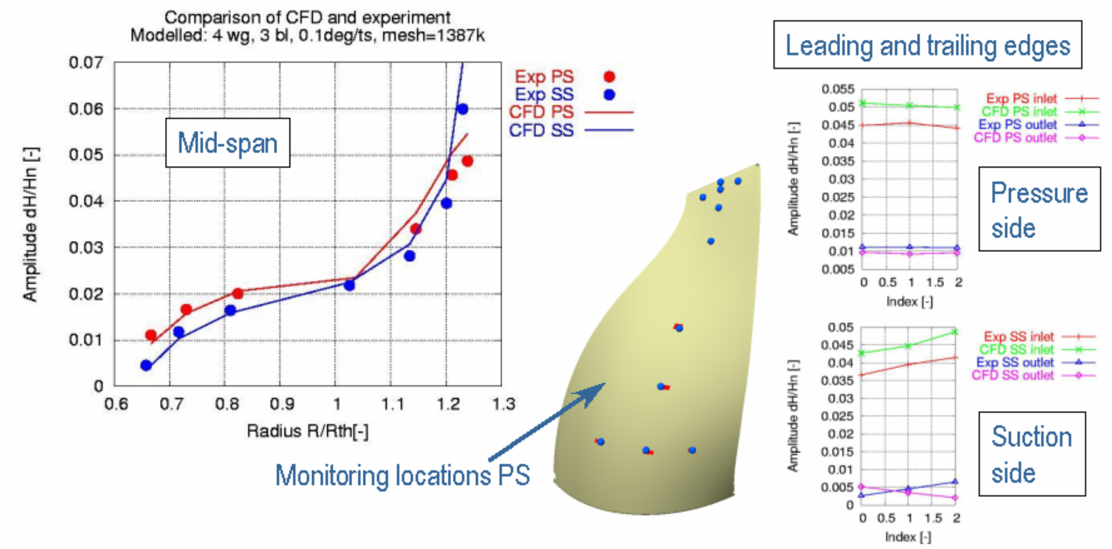

Figure 9: Comparison of experimental and computational results on pressure (PS) and suction (SS) sides

6. Procedure adopted for hydraulic design

Based on experience gained with rotor-stator CFD calculations for Francis runners, GE Energy Hydro has found empirical correlations to estimate the dynamic blade torque in a machine. If this estimate indicates potentially high levels of dynamic torque, a CFD rotor-stator calculation is performed with a standard mesh equivalent to the 316k-node mesh in **Figure 3**. This provides us with a conservative estimate and allows for a fast analysis and fast feedback to the hydraulic design engineer. If a high dynamic blade torque is confirmed by CFD, either the design is changed in order to lower the dynamic torque, or stress analyses are performed using advanced forced response technology [9]. The latter is used to determine accurate stress levels due the dynamic forcing from the wicket gate-blade interaction. Non-linear damping and resonance are taken into account in this methodology.

The procedure used for the CFD analyses of rotor-stator interaction at GE Energy Hydro has been streamlined to such an extent that fast and reliable results are achieved. Analyses speed is achieved using standard size meshes, the minimum number of wicket gate and runner channels required, automatic mesh generation and extensive scripting of the setup procedure. The accuracy and thus the reliability of the results have been validated with high quality unsteady pressure measurements. Due to the efficiency of the procedure and the availability of parallel computation capacities, unsteady CFD rotor-stator analyses are now routinely employed in the design of Francis runners.

7. Conclusion

GE Energy Hydro has developed a new procedure for the CFD analysis of the unsteady rotor-stator interaction between the wicket gates and runner blades of Francis turbines. The results from the CFD calculations have been successfully validated by means of high quality unsteady pressure measurements on a model turbine performed at École Polytechnique Fédérale de Lausanne. Due to the efficiency of the procedure and the available computational capacities, unsteady rotor-stator analyses are now routinely employed in the design of Francis runners.

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