

ONBOARD MEASUREMENTS OF PRESSURE AND STRAIN FLUCTUATIONS IN A MODEL OF LOW HEAD FRANCIS TURBINE – PART 2 : MEASUREMENTS AND PRELIMINARY ANALYSIS RESULTS

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ABSTRACT

In this second part of a 2 papers series, we present the analysis of “onboard” measurements on the runner of a Low Head Francis model and comparison with the same type of measurements on a similar prototype, as well as a comparison with numerical calculations. The details of the experimental procedures for onboard measurement of pressure and strain fluctuation in the model runner are presented in the first part (Ref. 1).

Load fluctuations on the blade of a Francis runner depend largely on the different hydrodynamic phenomena that may be present. For instance, the partial load rope vortex, blade trailing edge Karman vortices and any other cavitating occurrences at off-design operating points. Almost all of these phenomena can lead to fatigue failures, which translates into the interest of better understanding their behavior and interaction with the structures. The present paper shows how onboard measurements can be a valuable source of information on these phenomena and how it brings solutions to the evaluation of the fatigue life of turbine components. The good comparisons between the prototype and model results illustrate that the phenomena can be observed at both scales, although the transposition of dynamic component of pressure or stresses, is not yet feasible. The also very good correlation between measurements and numerical calculations of the flow and of the structural response of the runner to the static loading, enables us to progress to the next step, that is, dynamic analysis and fatigue life prediction with reliable data. Finally, we will show also that such measurements can be very useful in the development of numerical calculation tools used for performance optimization.

RÉSUMÉ

Dans cette deuxième partie d'un article à 2 volets, nous présentons l'analyse de mesures embarquées sur une roue modèle de turbine Francis de basse chute, ainsi que la comparaison avec des mesures similaires sur prototype et avec les calculs numériques. Le détail de la procédure expérimentale pour la mesure embarquée de la pression et des contraintes est présenté dans la première partie (Ref. 1).

Les fluctuations de charge sur l'aubage d'une roue Francis dépendent principalement des phénomènes hydrodynamiques au point de fonctionnement: torche de faible charge, tourbillons de Karman, et autre cavitations présentes aux points éloignés de l'optimum. Presque tous ces phénomènes peuvent mener à la rupture par fatigue, ce qui motive donc une meilleure compréhension de leur mécanisme et de leur interaction avec la structure. Cet article montre comment ce type de mesure permet de mieux comprendre les phénomènes et de résoudre les difficultés liées à l'évaluation de la durée de vie des composantes de la turbine, notamment la roue. Une bonne comparaison des résultats mesurés entre prototypes et modèles nous montre qu'on peut observer le phénomène en question dans les deux cas de la même façon, bien que la transposition des composantes dynamiques de la pression et des contraintes ne soient pas encore possible. L'excellente corrélation entre les résultats et les calculs numériques au point de vue du chargement statique, nous permet d'aller de l'avant dans l'étude des aspects dynamiques avec des données dont la fiabilité est validée. Enfin, on montrera aussi que ce type de mesures sera très utile pour le développement de nos outils de calcul numériques, utilisés eux-mêmes pour l'optimisation des performances.

NOMENCLATURE

p^*_{11}	Unit pressure	[bar]	S_{11}	Unit stress	[MPa]
Q	Flowrate	[m ³ /s]	H	Hydraulic head	[m]
Q_{opt}	Nominal flowrate	[m ³ /s]	H_{opt}	Nominal hydraulic head	[m]

INTRODUCTION

Operation at partial load of single regulated Francis turbine may lead to hydraulic dynamic phenomena in the runner. When units are operated extensively far from best efficiency point, these dynamic loads on the runner can conduce to damages on material. During past years, some fatigue problems were detected on some of our low head runners, mainly at blade to crown and blade to band junctions. The first identified cause was precisely the large amount of operating hours at partial load. A conservative and temporary solution was to use the turbine closer to nominal conditions.

In order to have a deeper comprehension of this phenomenon and to improve the design of Francis turbines for a safer partial load operation, ALSTOM decided in year 1999 to launch extensive investigations on this kind of low head turbines. The campaign started with prototype measurements on two low head Francis turbines in Brazil and lead to precise knowledge on the phenomenon for real life service conditions.

After field measurements, the program was continued in year 2001 with model test. Although prototype measurement allows to explore phenomenon in realistic conditions, the use of a model permits a larger exploration of the operation parameters, such as the net head, the effect of rotation speed or the submergence (tailwater level). It leads to better comprehension of physical phenomena identified as the source of the so called “partial load” dynamic phenomena.

EXPERIMENTAL MEASUREMENTS

From the prototype...

ALSTOM first performed extensive measurements on prototype Francis runners. As seen hereafter, the ground idea of the campaign will be to have similar type of measurements on both prototype and model runners. The first test on prototype leads to chose a low head type of turbine plant with a specific speed over 300 (rpm – kW – m).

Prototype measurements discussed hereafter were made in a Brazilian powerplant. The runner outlet diameter is close to 5 meters and the nominal output equal to 50 MW under 31 m head. Details of measurements on prototype turbine can be found in Ref. 2. In brief, the runner was instrumented with 51 strain gages on blade to crown and blade to band junction, 8 accelerometers (crown and band) and 3 pressure sensors. All rotating signals pass through the shaft up to the top of the rotor. A conditioning device with pre-amplifier, filtering and digital conversion delivers a signal at 1000 Hz sampling rate to a fixed recording system through a slip ring. On the entire turbine, an amount of 62 rotating signals and 9 fixed signals were simultaneously recorded. Main results and conclusions of this measurement campaign are detailed in Ref. 3.

...to the model

After this first step and in order to observe the turbine behavior in various operating conditions, ALSTOM launched a model test campaign. The same hydraulic design as the

prototype was used to fabricate a model runner. The test work was conducted in co-operation with the research institute EPFL-LMH. The model was specially designed and manufactured to suit rotating measurements on the runner. This campaign benefited from the large experience of the EPFL on model measurements Ref. 4.

The main objective of the model test was to determine the fluctuating pressure field in the flow. For this reason, many channels are dedicated to pressure sensors and the analogies are focussed on hydraulic laws.

Moreover, technical grounds don't allow direct transposition of the dynamic mechanical behavior between prototype and model. The response of the structure to fluctuating pressure is very different for both runners given the different boundary conditions and the different damping factors. For these reasons, we decided to record only some strain signals from the blades and two acceleration signals from the band.

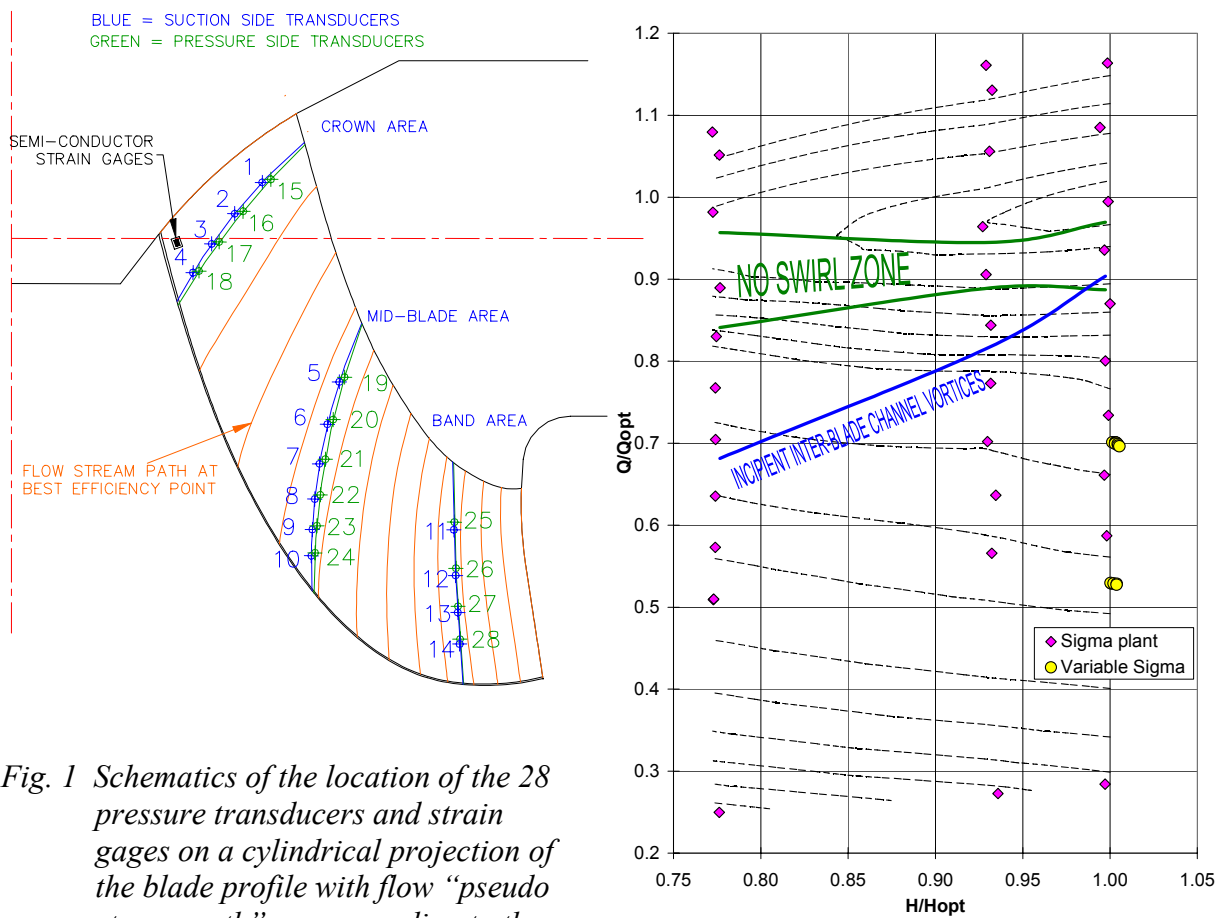


Fig. 1 Schematics of the location of the 28 pressure transducers and strain gages on a cylindrical projection of the blade profile with flow “pseudo stream path” corresponding to the rows of aligned pressure transducers.

Fig. 2 (on the right) model test hill chart with onboard pressure, strain and acceleration measurement points at plant sigma and variable sigma values.

In brief, 28 piezo-resistive pressure transducers are located on suction and pressure side of two facing blades. They are directly manufactured in the blade as such to avoid any geometrical modification of hydraulic profile which could be an eventual source of flow disturbance. Up-to 6 strain gages are located at the trailing edge of the blades near the junction with the crown. For these we developed in ALSTOM's lab the application of miniature semi-conductor strain gages for this type of application with excellent results. Also two accelerometers were located in the runner band. All the details of the model

instrumentation, static and dynamic calibration and the acquisition protocol are given in Ref. 1.

The transducers and gages showed really good reliability and enabled us to perform measurements over the whole hill chart operating range of the turbine as shown on Fig. 2. At the end of the measurement campaign only one of the 28 transducers became unusable.

In brief three series of measurement were performed:

- Series A : Q/Q_{opt} varying from 0.2 to 1.1 with H/H_{opt} equal to 0.77, 0.94 or 1.00
- Series B : Σ varying from 0.18 to 0.95 with H/H_{opt} equal to 1.00

Series A was made for Σ equal to plant value. Series B allows to determine the influence of Σ on dynamic phenomena.

ANALYSIS OF TESTS

Evolution of mean values

For the model test, each side of the blade is instrumented with 14 pressure sensors. The analysis of the static component of pressures gives interesting information about the loading of the blade for various operating conditions and may be compared with a flow calculation of any type for validation. For instance, at best efficiency point the comparison of the measured pressure with an inviscid flow calculation will validate the measurements, since the calculation is quite reliable at such operating points. In the near future same type of comparisons will be made at off-design operating points to validate our flow calculation tools, since the measurements are now believed of constant reliability, independently of the operating point. Thus for any operating conditions, the torque generated by the runner can be deduced by a simple integration on the full blade from the pressure difference measured between both side of it. This is a useful addition to our toolbox for the analysis of the mechanical capacity of the runner design under different operating conditions.

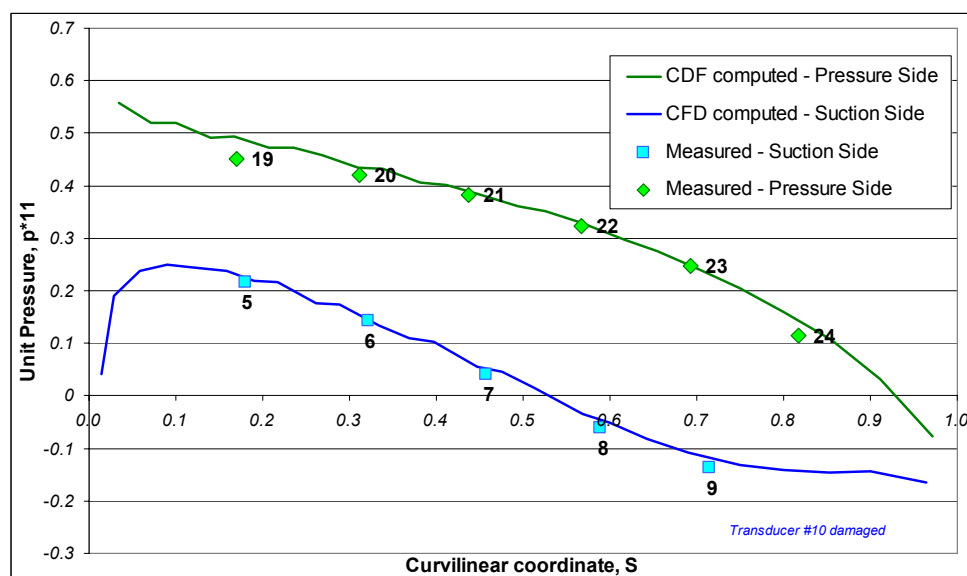


Fig. 3 Comparison of measured pressure with inviscid flow calculation at best efficiency point for a "pseudo stream path" located at mid-blade span.

One of the main parameters governing the overall pressure level, and thus the hydraulic load on the blade is the discharge of the turbine. A problematic aspect of the transposition of stress

measurements from model to prototype is the difference of geometry and of the boundary condition of the blade attachment to the crown and band. Consequently on Fig. 4 the comparison of the mean stress on the blade at the location of blade to crown junction near the trailing edge, between model and prototype, can be considered as very good for the whole range of discharge (or load). This allows us to confirm the quality of both prototype and model measurements and gives us a valuable information for further understanding of the transposition of the dynamic component of stress.

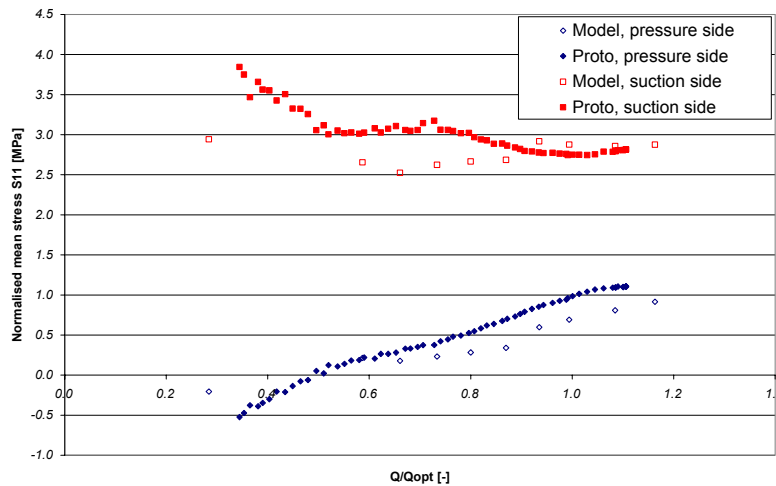


Fig. 4 Mean stress at the blade junction with the crown near the trailing edge for the whole range of prototype discharge. Comparison between model and prototype.

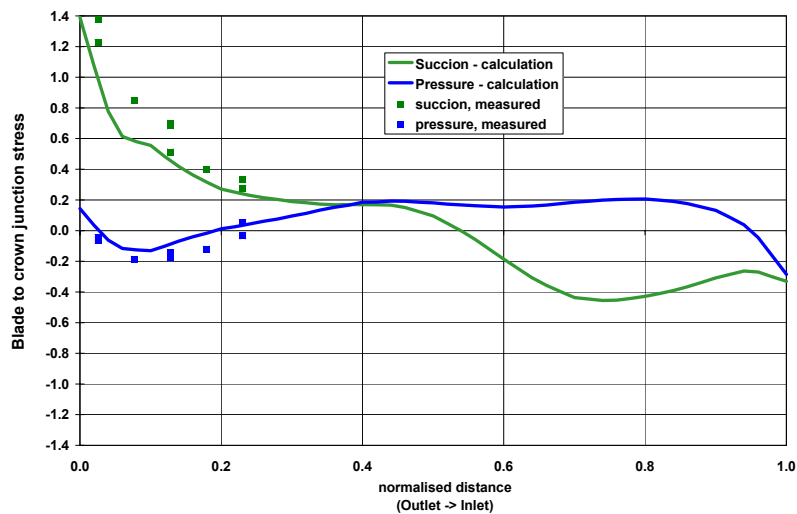


Fig. 5 stress along blade to crown junction at optimal operating point. Calculation and measurement

It is also clear that the stress level in the runner can be accurately calculated by finite elements model. The comparison between stress calculation and strain measurements on prototype shows very good correlation of results, as seen on Fig. 5.

These observations lead to conclude that the measurements made both on prototype and on model are valid and accurate, and show good comparison with our calculation tools for fluid and for mechanical calculations. Based on these conclusions we enable ourselves to pursuit the analysis of the results of these measurements for the dynamic aspects of it, which are pressure and strain fluctuations and their consequences on fatigue life of the runner.

Analysis of pressure fluctuations

Contrary to static pressure in the runner varying linearly with the load, measurement of pressure fluctuation shows strongly non linear behavior of dynamic phenomena with the operation point. As seen on the prototype turbine, dynamic amplitudes of pressure and stress increase quickly when going from full load to partial load and decrease again at low load (see Ref. 2 and Ref. 3 for more details). As a reminder, the global RMS stress level measured on blades was multiplied by about 5 to 6 times between 100 % load and the worse operation range close to 60-70 % load. Spectrum analysis also shown high frequency excitation at about 8 to 12 times the rotor rotation speed appearing at partial load.

A complete treatment of recorded data on prototype led to a good evaluation of fatigue life of the runner. The chosen method was “rainflow” analysis of the time stress signal given by strain gages located on most critical area. Fig. 6 shows relative life time of the prototype runner calculated from measured stress for different turbine output. Fatigue properties of materials are issued from corrosion-fatigue tests on cast steel (Ref. 5) and specific tests provided by ALSTOM for similar welds on stainless steel.

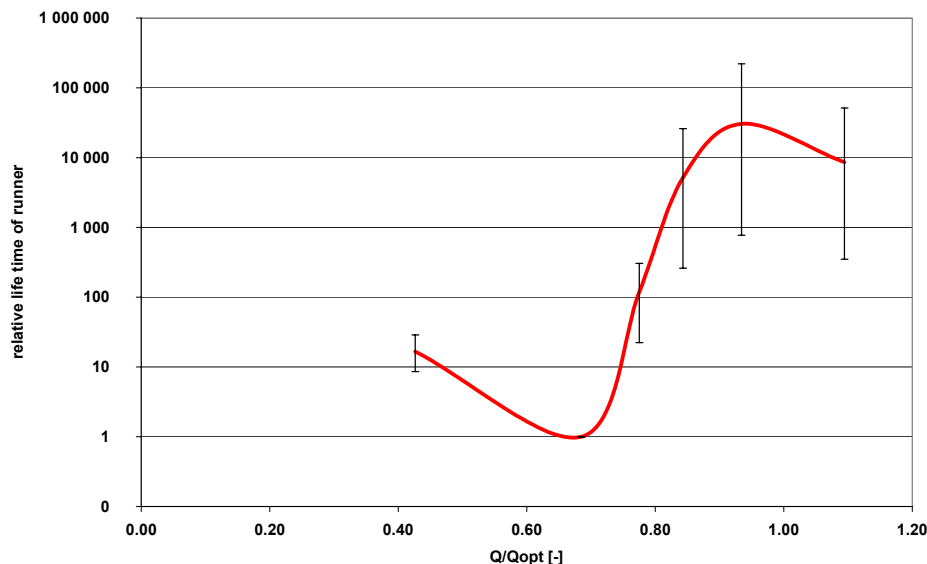


Fig. 6 calculated mean life time of the runner from load history analysis. Error bars include uncertainty from calculation methods and natural dispersion of material properties

The next step corresponding to the instrumentation of a model runner allows to go further in comprehension of phenomena observed on prototype runner. As seen above, dynamic pressure in the runner strongly depends on the operation point. Fig. 7 shows an example of pressure signal measured on blade at best efficiency point and at partial load. Note the high frequency dynamic pulsation appearing at partial load. FFT analysis of the pressure signal from blade gives the frequency range corresponding to damaging operation. The waterfall diagram on Fig. 8 gives a global view of the dynamic pulsation on blades for a range of discharges. Low frequencies pulsation (around 1 times the rotation frequency) are visible in the whole load range and linked to runner rotation and partial load rope, when the high frequencies pulsation (8 to 12 times the runner rotation) appears only at partial load. Stress fluctuations are also directly correlated with these pulsations in the runner. Also note the water flow on Fig. 7 at partial load where one can see typical cavitation phenomena.

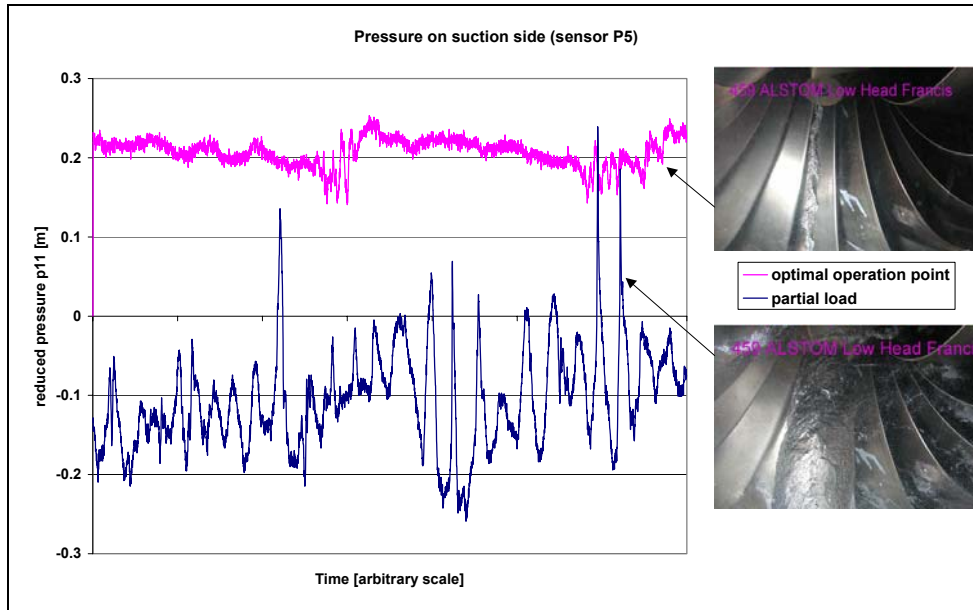


Fig. 7 comparison of pressure signal on suction side for optimal point ($Q/Q_{opt}=1$) and for partial load (Q/Q_{opt} in range from 0.60 to 0.75)

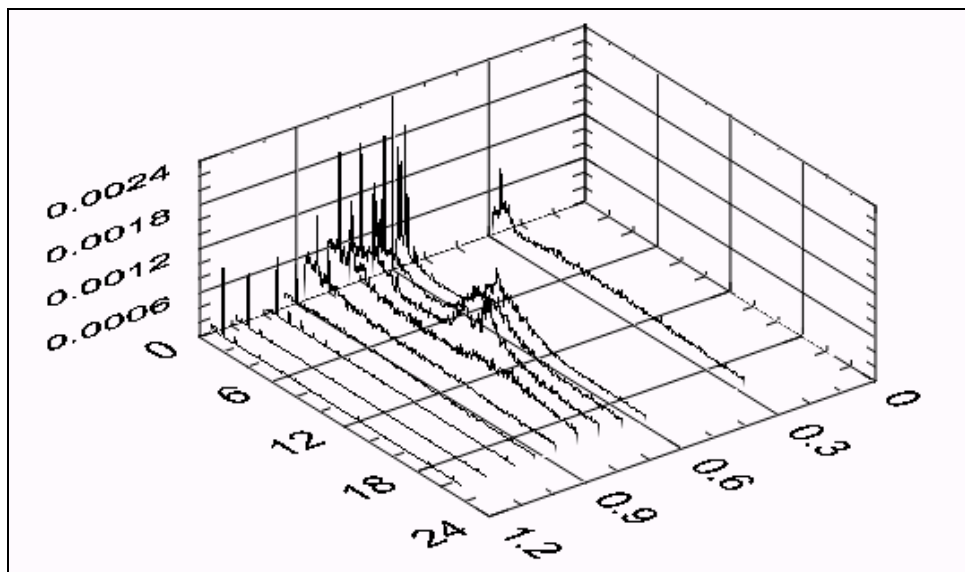


Fig. 8 Waterfall spectra of pressure on suction side (sensor P6) showing amplitude of pressure fluctuations against order (f/f_0) and relative discharge.

CONCLUSION AND PERSPECTIVES

After discovering of some fatigue failures on blades junctions of low head Francis runners, ALSTOM engaged an extensive investigation campaign. The first step concerned two different powerplants in operation in Brazil and mainly consisted of rotating strain measurement on runner blades and other structural parts. Exhaustive list of real life operations has been analyzed on site, from start up to runaway, stabilized output or SCO. This first campaign has quickly allowed to exclude some sources of troubleshooting such as natural vibrations of structures, mechanical or hydraulic unbalance, runner and guide vane interaction or Karman vortices. On the same way, it has led to define a critical range of

operation. Further fatigue calculation from recorded load history also confirmed the observed life time of runners.

As a first consequence, ALSTOM could define reactive solutions to return to normal plants operation conditions for owners. We can quote for example an optimization of the load dispatching on the different units avoiding running at critical partial load and leading to a more efficient use of runners (in term of better hydraulic efficiency but also of longer life time of structures). An other solution was the implementation of an ALSTOM new design of runner with “intermediate band” (Ref. 3), particularly well suited in case of necessary partial load operation.

A deeper comprehension of origin phenomena was possible thanks to a second measurement campaign on model. It has been led by ALSTOM and EPFL-LMH on model runner similar to Brazilian prototype powerplant. Although complete analysis of tests are still in progress, many conclusions can be already drawn. The similar global behavior of model and prototype runner concerning these low load phenomena is now proved, although particular cares must be taken concerning correct transposition of data both for amplitudes and frequencies. The high dynamic stress level observed on prototype blade is now identified as a consequence of particularly high pressure fluctuations in water flow passing between blades. Cavitation phenomena (as shown in Fig. 7) are pointed as a most basic source of flow disturbing. In addition, model test leads to better understanding of physical laws and governing parameters, such as the influence of the discharge rate, the cavitation number or the specific speed. At last, another very useful aspect of such model test campaign is the possibility to use the results to improve the calculation tools commonly used for design and for mechanical assessment of the runners.

ALSTOM is now using this large experience in partial load operation of Francis turbines for a better original design of next coming projects. A right analysis of model tests can now lead to concrete improvement concerning mechanical and hydraulic design of the runner, and also concerning the global use of the plant in operation inducing profitable gains of efficiency and for life time of structural parts.

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