

One-Dimensional Modeling of Rotor Stator Interaction in Francis Pump-Turbine

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Key words: Hydroacoustic modeling, rotor stator interaction, pump turbine

Abstract

A one-dimensional hydroacoustic model is set up to perform the numerical simulation of the rotor-stator interaction of a Francis pump-turbine scaled model. The numerical results enable to identify both rotating diametrical modes in the vaneless gap between the 20 guide vanes interacting with the 9 rotating impeller blades and the standing waves in the spiral casing. Moreover, the simulations enlighten the interaction between these rotating diametrical modes and the standing waves. Finally, the parametric study of the influence of the guide vane thickness, the wave speed and rotational speed emphasizes the predominant role of the eigen frequencies of the hydraulic system.

Introduction

Hydraulic pump-turbine operating under steady state conditions are subject to pressure fluctuations resulting from the interaction of the rotating parts and the stationary parts of the machine. This Rotor-Stator Interaction, RSI, are the consequence of the interaction between the rotating flow perturbations so-called potential flow perturbations caused by the impeller blades and the flow perturbation caused by the guide vanes. This interaction induces pressure waves propagating in the entire hydraulic machine. As a result the RSI phenomena may cause two different kinds of pressure fluctuations in the machine:

- diametrical pressure mode rotating in the vaneless gap between the guide vane and the impeller blades as described by Bolleter (Ref 1) and Tanaka et al. (Ref 2);
- standing waves in the spiral casing as described by Chen (Ref 3) and Dörfler (Ref 4).

The first phenomenon may cause hydromechanics resonance between the rotating diametrical pressure mode and the structure of impeller (Ref 5) or of the head cover (Ref 6) and may induce strong vibrations, noise, fissures or guide vanes bearing ruins. The second phenomenon may cause resonance with the power house structure that generates unacceptable vibrations and noise (Ref 2). The standing wave phenomenon may affect also the penstock (Ref 4), (Ref 7) which evidences the potential interaction of the hydraulic machine with the hydraulic circuit.

The prediction of such phenomena is a challenging task during the early stage of the design of a reversible pump turbine unit for a hydroelectric power plant. Some analytical models have been developed by Bolleter (Ref 1) and Tanaka and al. (Ref 2) for the diametrical mode shape and by Chen (Ref 3) and Dörfler (Ref 4) for the standing wave allowing for the prediction of the risk of occurrence of these phenomena. The prediction of the occurrence of the standing wave is based on the analysis of the traveling time of pressure waves propagating in a one-dimensional system modeling the pump turbine accordingly to its topology. Recently, Haban et al. [8] have developed more sophisticated one-dimensional models based on matrix transfer method that have shown their capability of predicting spiral casing standing wave patterns. By the use of such models Fischer et al. (Ref 6) have pointed out the link between the standing waves in the spiral casing and the penstocks and the diametrical pressure mode rotating in the vaneless gap by performing a forced response analysis in the frequency domain. However this approach requires the identification of the excitation pattern by the method described by Bolleter (Ref 1) and Tanaka et al. (Ref 2).

The simulation of the incompressible 3D unsteady flow of a vaneless centrifugal pump performed by Gonzalez et al. (Ref 9) using commercial CFD tool has shown the capability of CFD to predict accurately the unsteady convective flow field related to the RSI phenomenon at the blade passing frequency which is dominant close to the nominal operating point. However some discrepancies appeared for off-design operating conditions where acoustic behavior becomes more effective due to a blade-tongue interaction. Such an incompressible code cannot account for the propagating part of the flow which may lead to standing wave phenomenon. Therefore the separation of hydraulic and acoustic pressure fluctuations by means of least-squares residual method developed by Morgenroth (Ref 10) was used. A model accounting for both the jet-wake pattern of the convective flow and for the blade-tongue interaction has been set up by Parrondo et al. (Ref 11). Through an experimental identification of the parameters of acoustic model a good agreement has been found between the model and the measurements even for off-design operating conditions where the amplitude of the blade-tongue interaction was found to have 10 times the amplitudes of the wake jet part of the flow.

This paper aims to present the numerical simulation of the RSI phenomenon based on a one-dimensional hydroacoustic model. Therefore the case of 20 guide vanes and 9 impeller blades high head Francis pump turbine is investigated. First the RSI patterns of the pump turbine are described. Then the mathematical model implemented in the software SIMSEN is briefly presented. The determination of the pump turbine hydroacoustic parameters is described. The RSI excitation is modeled by a valve network driven accordingly to the spatial and temporal evolution of the flow distribution between the stationary and the rotating parts of the machine. The results obtained by the simulations in time domain evidences the RSI patterns of the pump turbine of interest. A parametric study is presented and influence of blade thickness, wave speed and the rotational speed is investigated.

Nomenclature

A = Cross Section Area, m^2	D = Pipe diameter, m
C = Absolute Mean Flow Speed, m/s, $C = Q/A$	E = Young Modulus, Pa
C_h = Hydraulic Capacitance, m^2 , $C_h = gA/a^2$	E = Machine Specific Energy, J/kg, $E = gH_1 - gH_2$

H = Head, m, $H = E/g$	ν = Machine Specific Speed, $\nu = \varphi^{1/2}/\psi^{3/4}$
L_h = Hydraulic Inductance, s^2/m^2 , $L_h = dx/gA$	λ = Local Loss Coefficient
Q = Flow rate, m^3/s $Q = C \cdot A$	ω_b = Impeller Angular Speed, rad/s
R_h = Hydraulic Resistance, s/m^2 , $R_h = \lambda dx Q /2gDA^2$	n = Harmonic Order
R_{ref} = Machine Reference Radius, m	m = Harmonic Order
h = Piezometric Head, m, $h = p/\varpi + Z$	z_o = Number of Guide Vanes, $z_o = 20$
a = Wave Speed, m/s	z_b = number of impeller blades, $z_b = 9$
f_b = Impeller Rotational Frequency, Hz	θ_s = Angle in the Stationary System
p = Pressure, Pa.	θ_r = Angle in the Rotating System
g = Gravity Acceleration, m/s^2	B_n = Amplitude for the n^{th} Harmonic, Pa.
ρ = Water Density, kg/m^3	B_m = Amplitude for the m^{th} Harmonic, Pa
φ = Discharge Coefficient, $\varphi = Q/\pi\omega R_{ref}^3$	ϕ_n = Phase for the n^{th} Harmonic
ψ = Specific Energy Coefficient, $\psi = 2E/\omega^2 R_{ref}^2$	ϕ_m = Phase for the m^{th} Harmonic

RSI Patterns in Francis Pump Turbine

The flow field leaving the guide vane of a Francis pump turbine in generating mode is characterized by the velocity defect caused by the guide vanes. The pressure field attached to the rotating impeller blade induces also incoming flow field distortions. No matter how complex are these two periodic flow fields they can be expressed as Fourier series. Then, both the stationary and rotating pressure fields can be expressed as:

$$p_s(\theta_s, t) = \sum_{n=1}^{\infty} B_n \cos(nZ_o\theta_s + \phi_n) \quad p_r(\theta_r, t) = \sum_{m=1}^{\infty} B_m \cos(mZ_b \cdot \theta_r + \phi_m) \quad (1) \text{ and } (2)$$

The resulting pressure field, combining the guide vanes and the impeller blade pressure field is characterized by a strong modulation process as illustrated Figure 1. The pressure in the area of the vaneless gap between the guide vanes and impeller blades can therefore be expressed as the product of both rotating and stationary fields of pressure leading to the summation of every p_{mn} component:

$$p_{mn}(\theta, t) = A_{mn} \cos(nZ_o \cdot \theta_s + \phi_n) \cdot \cos(mZ_b \cdot \theta_r + \phi_m) \text{ for } n = 1, 2, \dots, \infty \text{ and } m = 1, 2, \dots, \infty \quad (3)$$

Where A_{mn} is the combined pressure amplitude due to the interaction of each harmonics (Ref 2). By using the well known identity for circular functions, Eq.(3) can be expressed as:

$$p_{mn}(\theta_s, t) = \frac{A_{mn}}{2} \cos(nZ_o \cdot \theta_s - mZ_b \cdot \theta_r + \phi_n - \phi_m) + \frac{A_{mn}}{2} \cos(nZ_o \cdot \theta_s + mZ_b \cdot \theta_r + \phi_n + \phi_m) \quad (4)$$

Moreover, the impeller angle coordinate is related to the stationary system of reference as $\theta_r = \theta_s - \omega t$, then the pressure field in the stationary coordinates becomes:

$$p_{mn} = \frac{A_{mn}}{2} \cos(mZ_b \omega t - (mZ_b - nZ_o)\theta_s + \phi_n - \phi_m) + \frac{A_{mn}}{2} \cos(mZ_b \omega t - (mZ_b + nZ_o)\theta_s - \phi_n - \phi_m) \quad (5)$$

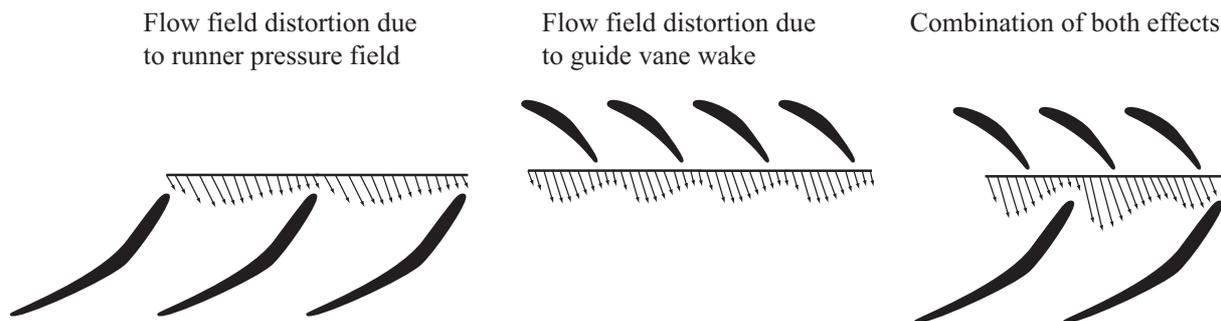


Figure 1 Modulation process between impeller blade flow field and guide vanes flow field.

This equation describes the RSI pressure field in the vaneless gap which is function of time and space (Ref 12). This pressure field represents 2 diametrical pressure modes having the following numbers of minima and maxima :

$$k_1 = m \cdot Z_b - n \cdot Z_o \text{ and } k_2 = m \cdot Z_b + n \cdot Z_o \quad (6)$$

rotating with the respective spinning speed in the stationary frame of reference.

$$\omega_1 = mZ_b\omega_b / k_1 \text{ and } \omega_2 = mZ_b\omega_b / k_2. \quad (7)$$

Furthermore, the sign of the diametrical mode numbers k_1 and k_2 indicates that the diametrical mode is rotating in the same direction as the impeller when positive and counter-rotating when negative. It is also important to notice that lower amplitudes are expected for higher k values, because of the high harmonic number. As a result, k_2 is usually not relevant. The pump turbine of interest is a scale model of pump turbine which has a specific speed of $\nu = 0.17$, 20 guide vanes, 9 impeller blades and an outlet diameter (turbine mode) of $R_{le} = 400$ mm. According to this rotor-stator blade arrangement the RSI patterns of this pump turbine are determined analytically using relations (5), (6) and (7), see Table 1.

Table 1 RSI patterns of the pump turbine ($Z_o = 20$; $Z_b = 9$).

n	m	k_1	k_2	Stationnary frame			Rotating frame		
				ω_1 / ω_b	ω_2 / ω_b	f / f_b	ω_2' / ω_b	ω_2'' / ω_b	f / f_b
		$mZ_b - nZ_o$	$mZ_b + nZ_o$	mZ_b / k_1	mZ_b / k_2	mZ_b	nZ_o / k_1	$nZ_o / -k_2$	nZ_o
1	2	-2	38	-9.0	0.5	18	-10.0	0.5	20
1	3	7	47	3.9	0.6	27	2.9	0.4	20
2	4	-4	76	-9.0	0.5	36	-10.0	0.5	40
2	5	5	85	9.0	0.5	45	8.0	0.5	40

Table 1 points out the 4 diametrical rotating modes shapes that may present high amplitudes, it is for $k_1 = -2, 7, 5, -4$ with the corresponding frequencies in the stationary frame $f / f_b = mZ_b = 18, 27, 36$ and 45. In the rotating frame of reference these frequencies reduced to $f / f_b = mZ_o = 20$ for the first 2 modes and $f / f_{bo} = 40$ the second 2 modes.

Hydroacoustic Modeling of Francis Pump Turbine

Hydroacoustic modeling

By neglecting the convective terms $C \partial/\partial x$ and assuming plane pressure wave and uniform velocity field in a cross section, the momentum and the continuity equations established for the pipe of a length dx , a cross section A and a wave speed a , Figure 2, reduces to the hyperbolic partial differential equations see (Ref 13) :

$$\frac{\partial h}{\partial t} + \frac{a^2}{gA} \cdot \frac{\partial Q}{\partial x} = 0, \quad \frac{\partial h}{\partial x} + \frac{1}{gA} \cdot \frac{\partial Q}{\partial t} + \frac{\lambda |Q|}{2gDA^2} \cdot Q = 0 \quad (8)$$

Where the h and Q variables are respectively the piezometric head and the discharge expressed as:

$$h = z + p/(\rho \cdot g) \quad Q = C \cdot A \quad (9)$$

The system of hyperbolic equations (8) is solved using Finite Difference Method considering a 1st order centered scheme discretization in space and a scheme of Lax for the discharge. This approach leads to a system of ordinary differential equations that can be represented as a T-shaped equivalent scheme as presented Figure 2. The RLC parameters of this equivalent scheme are given by:

$$R_h = \frac{\lambda \cdot |Q| \cdot dx}{2 \cdot g \cdot D \cdot A^2} \quad L_h = \frac{dx}{g \cdot A} \quad C_h = \frac{g \cdot A \cdot dx}{a^2} \quad (10)$$

Here λ is the friction coefficient. The hydraulic resistance R_h , the hydraulic inductance L_h , and the hydraulic capacitance C_h correspond respectively to losses, inertia and storage effects.

The model of pipe with a length L is made of a series of n_p elements based on the equivalent scheme of Figure 2 (Ref 14). The system of equations relative to this model is set-up using Kirchoff laws. Time integration of the full system is achieved by a Runge-Kutta 4th order procedure.

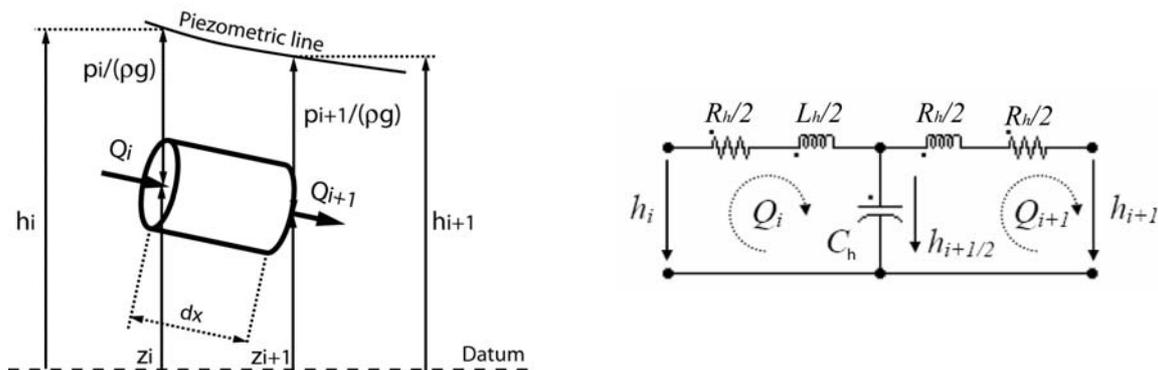


Figure 2 Equivalent scheme of a pipe.

All the procedures for building up, solving and analyzing the hydroacoustic model of the whole system are implemented within the in house code SIMSEN (Ref 15). Moreover, SIMSEN provides a complete object library including both electrotechnical, control and hydraulic components such as valve, surge tanks, turbines.

Pump turbine hydroacoustic model

The plan view of the 20 guide vanes and 9 blades Francis pump turbine scale model of interest is presented Figure 3. The hydroacoustic model of this pump turbine is made of a pipe network, Figure 3.

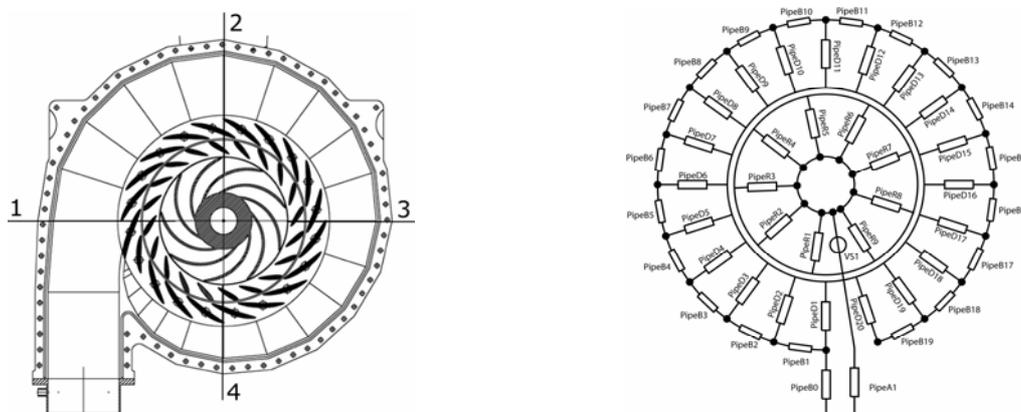


Figure 3 Plan view of the pump turbine(left) and hydroacoustic model of the pump turbine(right).

The hydroacoustic model is made of 20 pipes for the guide vanes (pipes D1 to D20), 9 pipes for the impeller (pipes R1 to R9) as well as 19 pipes for the spiral casing (pipes B1 to B19). The first part of the spiral casing between the turbine inlet and the guide vane N°1 is modeled by the pipe B0. The diffuser of the pump turbine is modeled by the pipe A1. The energy transfer through the impeller is modeled by the pressure “source” VS1 which head is function of discharge $H=H(Q)$ according to the slope of the pump turbine characteristics linearized around the operating point of interest. The connection between the stationary part and the rotating part is achieved through 180 valves connecting each guide vane to each impeller vane. The 180 valves are controlled by the flow distribution between the stationary part and the rotating part according to the impeller angular position $\theta(t)$. The valve head loss is calculated to ensure the idealized discharge evolution presented Figure 4.

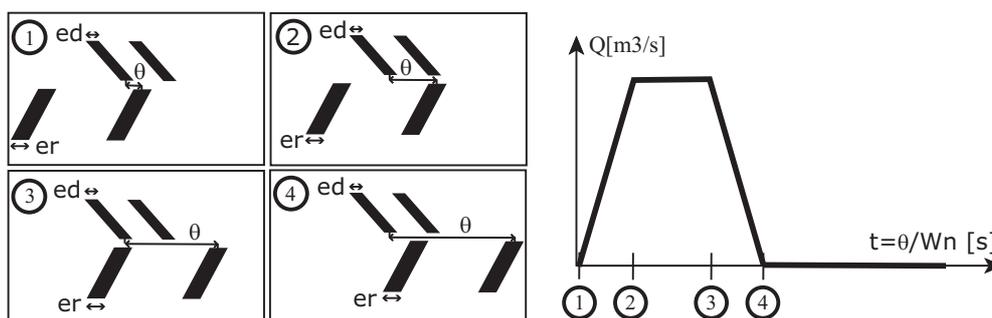


Figure 4 Idealized discharge evolution between an impeller vane and a guide vane as function of the spatial coordinate of the impeller θ .

Assuming a constant impeller speed the discharge evolution between one guide vane and one impeller vane is function of the connection area between them. During the rotation of the impeller 4 phases are identified:

- phase 1: the impeller blade start to pass in front of the first blade of a guide vane, the

- discharge between the guide vane and the impeller vane increases linearly accordingly to the connection area increase until the impeller blade reaches the second blade of the guide vane;
- phase 2: the discharge between the guide vane and the impeller vane remains constant until the second impeller blade reaches the first blade guide vanes, the connection area being constant;
 - phase 3: the discharge between the guide vane and the impeller vane decreases linearly according to the connection area decrease;
 - phase 4: the discharge between the guide vane and the impeller vane is kept to zero as the connection area is zero until the phase 1 is reached again.

The discharge evolution described above acts like a sliding slot between the 20 guide vanes and one impeller vane. As a result there are 9 slots for the full pump turbine, one for each impeller vane. Each slot angle being shifted by $2\pi/Z_o$. The discharge law can be modified in order to take into account the thickness of both the guide vanes and impeller blades, e_o and e_b respectively. The thickness can be expressed as equivalent angle measured in degrees at the vaneless gap radius. For the first part of investigations of this paper, the thickness of the impeller blades is taken arbitrary equal to 4° , the real one being 3° . The consideration of the blades thickness induces discontinuity in the overall discharge law. It means that the point 3 of the discharge law of an impeller blade does not correspond to the point 2 of the following impeller blade but is shifted of the value of e_b . The lack of discharge between two consecutives impeller blades is the source of the excitation mechanism of this RSI model.

Hydroacoustic Parameters Determination

For the determination of the RLC terms of the hydroacoustic model of a pipe the following values must be determined for each pipe:

- the length, L ;
- the cross section area, A ;
- the friction coefficient, λ ;
- the wave speed, a .

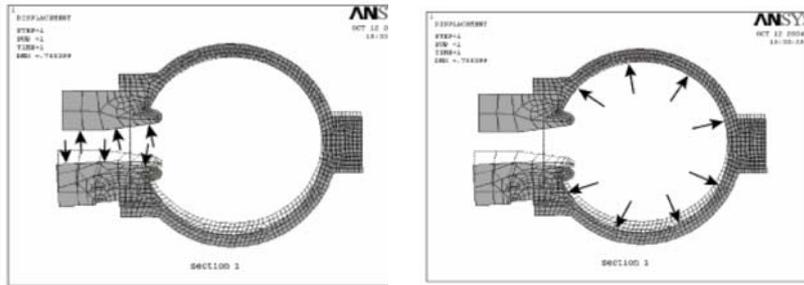
The determination of the length and cross section has been done using the structural characteristics of the pump turbine scale model. The friction coefficient of the pipes has been evaluated to be $\lambda = 0.02$ for all the pipes. The determination of the wave speed is described below.

Wave speed determination

The wave speed a for the pipe is given by:

$$a^2 = \left[\rho \left(\frac{1}{E_{\text{fluid}}} + \frac{1}{A} \frac{\Delta A}{\Delta p} \right) \right]^{-1} \quad (11)$$

Where ρ is the water density, E_{fluid} is the bulk modulus of the water and $A^{-1}\Delta A/\Delta p$ is the rated area increase due to pressure increase. The wave speeds of the spiral casing and of the distributor are estimated using Finite Element Method -FEM- calculations on 2D cross sections.



Section 1: $a=1052$ m/s
 Section 2: $a=1063$ m/s
 Section 3: $a=1111$ m/s
 Section 4: $a=1170$ m/s

Figure 5 Determination of the wave speed in the distributor channel (left) and in the spiral casing (center), results of calculations (right)

For the 4 cross sections defined in Figure 3, the calculated values of the wave speed are listed Figure 5. The wave speed of the distributor channels is found to be 800 m/s with small deviation between the 4 sections. The same value is chosen for the wave speed of the impeller blades.

Simulation of the Pump Turbine RSI

The pump turbine scale model of interest is installed on the test rig PF3 of the EPFL Laboratory for Hydraulic Machines. The hydroacoustic model of the test rig is including the pump turbine. A full description of the model of this test rig can be found in (Ref 16). The test rig is made of 2 feedings pumps in series, a piping system, the pump turbine model itself and a downstream tank as presented Figure 6. The chosen operating condition for the simulation of the hydroacoustic behavior of the test rig corresponds to the nominal operating point with no cavitation, a discharge of 0.23 m³/s, a test head of 45 m and a rotational speed of 980 rpm. Each pipe of the spiral casing is considered as 1 element, and each guide vane and impeller vane is modeled with 3 elements. This spatial discretization ensures that a wavelength at the frequency $18f_b$ ($k_1 = -2$) is represented by 20 elements. In agreement with the CFL criteria, an integration time step $dt = 85.034$ μ s corresponding to 0.5° of impeller rotation is selected. After convergence of the simulation to a periodic behavior, the simulation is continued for 10 impeller revolutions.

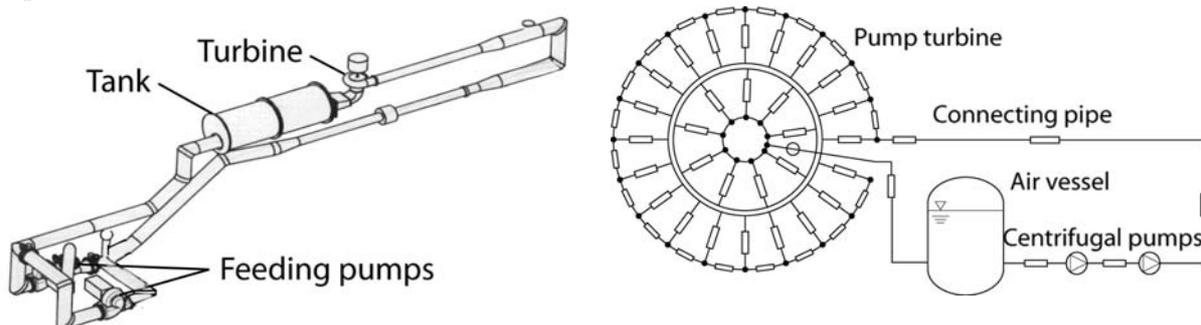


Figure 6 EPFL test rig PF3 and hydroacoustic model of the closed loop test rig N°3.

The resulting pressure fluctuations in the spiral casing and in the guide vanes close to the vaneless gap are properly non dimensionalized and presented in a waterfall diagram as a function of the dimensionless frequency f/f_b and the angular coordinate, see Figure 7. In these diagrams the angular origin is taken at the spiral casing inlet and therefore $\theta = 360^\circ$ correspond to the tongue.

The pressure fluctuation in the vaneless gap, Figure 7, presents significant amplitudes for the expected frequencies $f/f_b = 9, 18, 27, 36, 45$, etc. and represents less than 0.5% of the head. The analysis of the phase, not presented here, of these pressure fluctuations shows clearly the diametrical pattern of the pressure mode shapes with for example 2 minima and 2 maxima, i.e. 4π phase shift, for the frequency of $f/f_b = 18$. The pressure fluctuations in spiral casing Figure 7 are characterized by a standing wave for the frequency $f/f_b = 18$ with a wavelength of approximately $4/3$ of the spiral casing length. A pressure node is located at $\theta = 150^\circ$ while amplitude maxima take place at $\theta = 20^\circ$ and $\theta = 230^\circ$. This standing wave influences the amplitude of the diametrical mode of the pressure fluctuation in the vaneless gap see Figure 7.

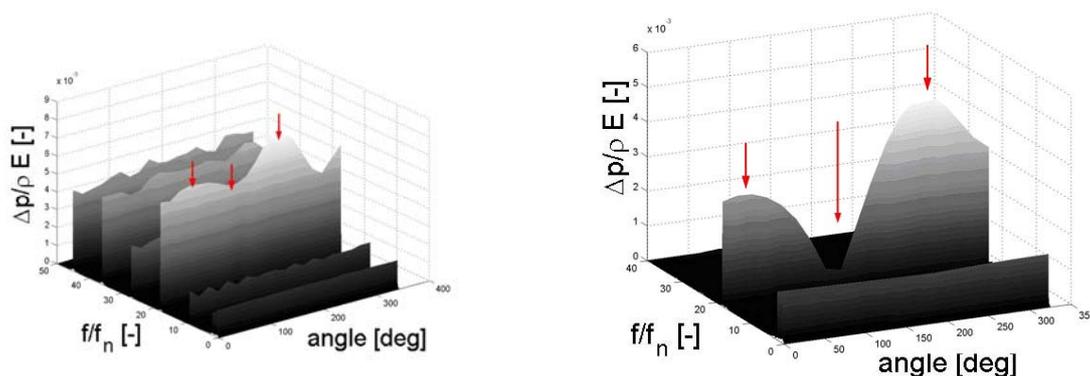


Figure 7 Waterfall diagram of the pressure fluctuations in the guide vanes close to the vaneless gap (left) and in the spiral casing (right).

The patterns of the pressures fluctuations for $k_1 = -2$ are illustrated Figure 8. The representation is obtained by filtering the time signal of the pressure fluctuations of each spatial node in the pump turbine using a pass band filter around the frequency of interest. For the stationary frame it corresponds to $f/f_b = 18$. In the rotating frame it corresponds to $f/f_b = 20$. The representation is done for the initial time and for a $1/4$ period later, Figure 8. The pressure amplitude of every node of the pump turbine is represented on the z axis for the spatial position $[x, y]$ of the node. The bold solid line is the connection between each last node of the guide vane and point out the diametrical mode shape pattern expected for $k_1 = -2$ with 2 maxima/minima. The diametrical mode rotates in the opposite direction of the impeller rotation as predicted by the analytical analysis above. The diametrical mode rotates with a spinning speed of $f_b = -9$. It means that using a pressure transducer in the vaneless gap, the frequency of $f/f_b = 18$ is measured. The interaction between the standing wave in the spiral casing and the diametrical mode are clearly pointed out through the pressure in the guide vanes where pressure fluctuations have similar amplitudes. The results obtained for $k_1 = -4, 5, 7$ show also good agreements with the analytical prediction. However, the $k_1 = 7$ diametrical mode is the last resolvable mode with a resolution of 20 pressure signals. As presented in Figure 7, there is no standing wave in the spiral casing for these frequencies.

From these results it can be conclude that hydroacoustic model of the pump turbine allows to obtain and to visualize the RSI pressure fluctuation patterns. Standing wave and diametrical modes are pointed out and corresponding spinning speeds are properly obtained too. The high amplitude of the

standing wave in the spiral casing for $k_1 = -2$ is probably related to a natural frequency of the hydraulic system which is close. However, it is important to notice that rotating impeller corresponds to unsteady boundary conditions at the end of the guide vanes. It makes the modal analysis difficult to be done by using linearized approaches. One of the major advantages of simulating the dynamic behavior of such a system in the time domain is that it offers the possibility to take into account such non-linearities. The modeling based on valve network driven accordingly by the flow distribution between stationary and rotating parts appears to be efficient for the simulation of the hydroacoustic part of the RSI phenomenon. This approach has the advantage to provide also the pressure fluctuations due to RSI in the rotating impeller.

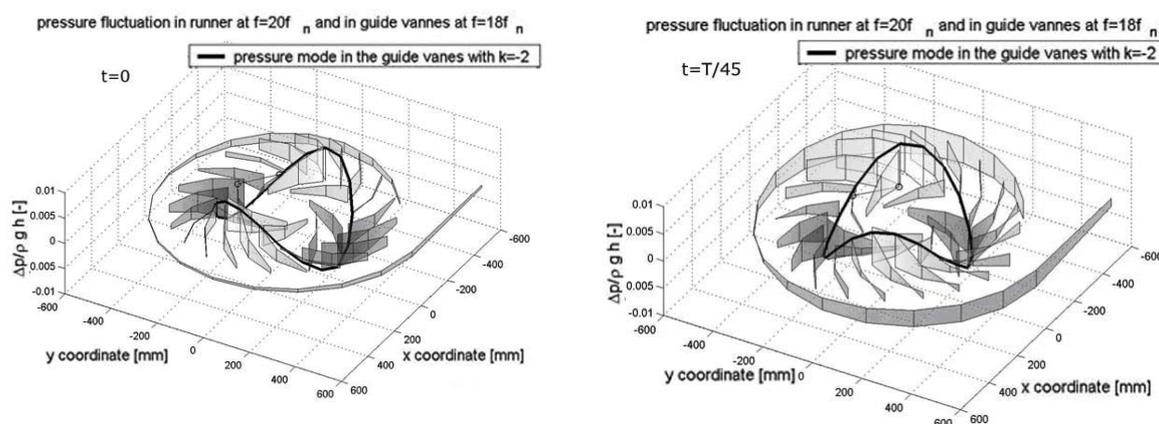


Figure 8 Pressure fluctuations patterns for $k_1 = -2$, $f/f_b = 18$, $t = 0$ (left) and $t/T_b = 1/45$ (right).

Parametric Study

As the hydroacoustic parameters determination is based on a simplified approach and some parameters are selected arbitrarily, a parametric study is performed in order to assess the sensitivity of the hydroacoustic model to these parameters. Therefore, the influence of the following parameters is investigated in this paragraph:

- blade thickness;
- impeller vane wave speed;
- guide vane wave speed;
- impeller rotating frequency.

The influence of these 4 parameters is investigated considering the frequency, $f/f_b = 18$, mode $k_1 = -2$.

Blade thickness

The model of the excitation is based of the flow distribution between the stationary part and the rotating parts. The resulting discharge law between the 20 guide vanes and the 9 impeller blades can account for the guide vanes and impeller blade thickness. The equivalent thickness of these blades affects the evolution of the connection area between a guide vane and a impeller vane. The Figure 9 presents the 4 possible configurations which are : (A) no blade thickness, (B) consideration of guide vanes thickness only, (C) consideration of impeller blade thickness only and (D) consideration of both

guide vanes and impeller blade thickness. The 4 different evolutions of the discharge law of the configurations (A), (B), (C) and (D) are represented on the right part of Figure 9.

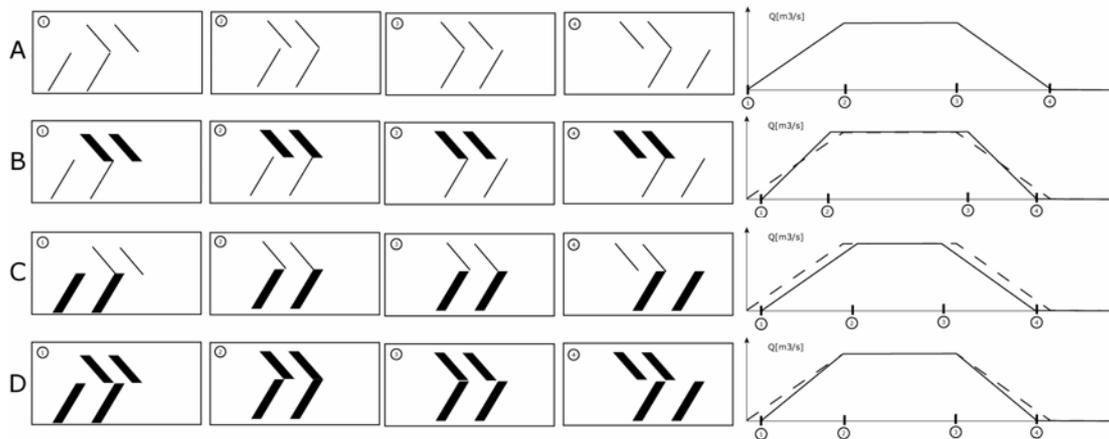


Figure 9 Discharge law resulting from the guide vanes and impeller blade thickness consideration.

Considering the configurations (B), (C) and (D) 3 series of simulation has been performed considering different blade thickness e . The resulting amplitudes of the pressure fluctuations in the vaneless gap are presented Figure 10. The amplitude corresponding to configuration (B) and (C) provides similar pressure amplitudes in the vaneless gap for identical thickness. The combination of both thicknesses, configuration (D), induces much higher pressure amplitudes. These results show that the higher the blockage effects due to blade thickness, the higher pressure amplitudes in the vaneless gap.

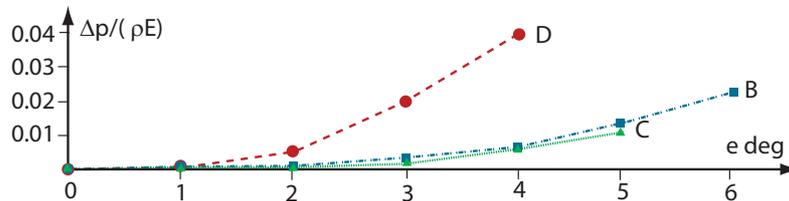


Figure 10 Pressure fluctuations amplitudes in the vaneless gap for the blade thickness arrangement of Figure 9 for $f/f_b = 18$.

A dynamic pressure amplification factor is used to evaluate the influence between the vaneless gap pressure amplitudes and the amplitudes in the spiral casing. It allows separating the influence of the parameter on the RSI excitation and on the dynamic response of the system. The amplification factor is defined as the ratio between the pressure at any point in the spiral case and the average pressure in the vaneless gap and is expressed as:

$$DA = \frac{P_{\text{spiral}}}{P_{\text{vaneless gap}}} \quad (12)$$

Figure 11 presents the dynamic amplification factor for 3 different guide vanes thicknesses. It appears that even if the absolute amplitudes of the pressure fluctuations increases, as the excitation in the vaneless gap increases, the dynamic amplification factor remains constant for the 3 tested thicknesses. One interesting consequence of this result is that even if the excitation parameters are arbitrary, the

dynamic amplification will be meaningful. It would also mean that if the pressure fluctuations can be measured in the vaneless gap during the scale model tests, it could be possible to predict the amplitude of pressure fluctuations in the spiral case.

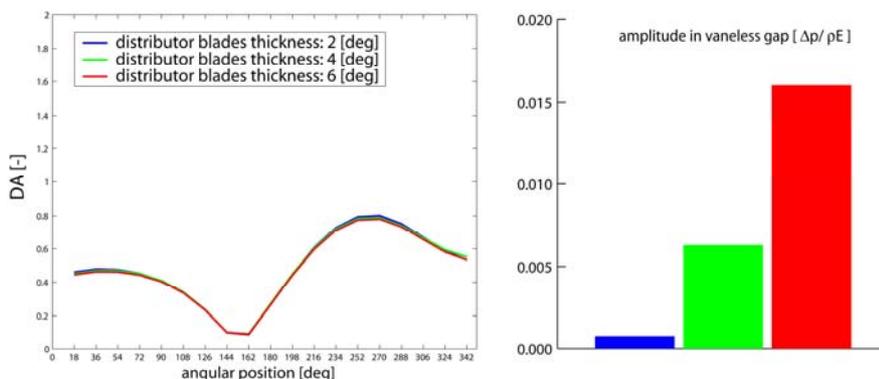


Figure 11 Influence of the distributor blade thickness for $f/f_b = 18$.

Impeller vane wave speed

The influence of the impeller vane wave speed is investigated by simulating the dynamic behavior of the system considering three different values for this parameter. The chosen values for the impeller wave speed are 700 m/s, 800 m/s and 900 m/s. Figure 12 presents the dynamic amplification factor DA for the 3 different impeller vane wave speeds.

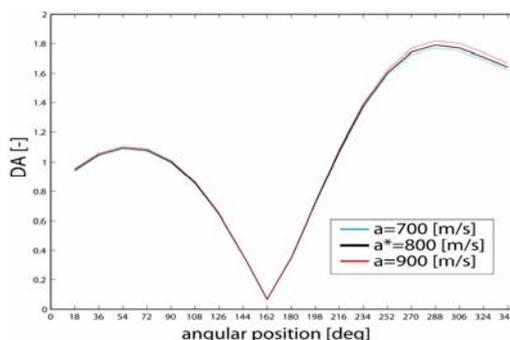


Figure 12 Influence of the impeller vane wave speed on the dynamic amplification between spiral case and vaneless gap for $f/f_b = 18$.

It is clearly shown that the influence of the impeller vane wave speed is negligible. It means that for pump turbine the combine effect of small vaneless gap and the impeller rotation acts like dead end for such frequency. However, it is right as long as the diffuser remains short compared to the system dimensions.

Guide vane wave speed

4 different values of the guide vane wave speed are chosen to investigate the influence of this parameter on the dynamic amplification between vaneless gap and spiral casing. The chosen values for the guide vane wave speed are 700 m/s, 800 m/s, 900 m/s and 1000 m/s. The resulting dynamic amplification DA and mean value of vaneless gap pressure fluctuation are presented Figure 13.

The mean pressure fluctuation in the vaneless gap remains constant for the 4 cases. However, the dynamic amplification is strongly influenced by the guide vane wave speed. This is related to the fact that traveling time between vaneless gap and spiral casing affects strongly natural frequencies of the hydraulic system. It seems that in this case reducing the guide vane wave speed makes a natural frequency of the hydraulic system closer to $f/f_b = 18$.

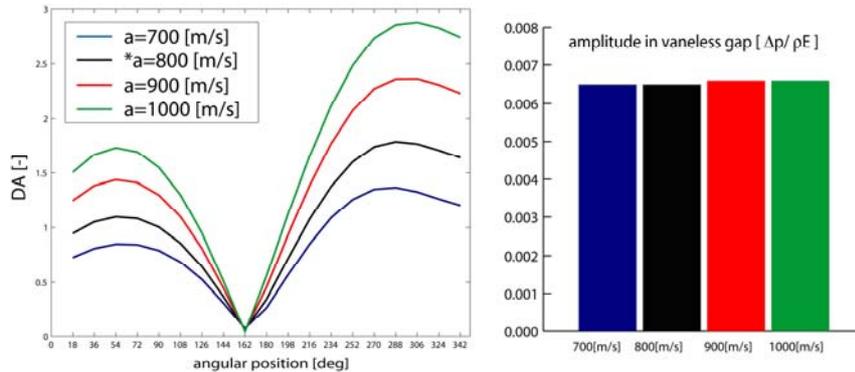


Figure 13 Influence of the guide vane wave speed for $f/f_b = 18$.

Impeller rotating frequency

4 different impeller rotation speeds are chosen to investigate the influence of this parameter. The chosen values corresponds to $f = 18 \cdot f_b = 292, 296, 300, 304$ Hz . The resulting dynamic amplification DA and mean value of vaneless gap pressure fluctuation are presented Figure 14.

The impeller rotating speed has strong influence on the dynamic amplification between the vaneless gap and the spiral case. This is due to the change of excitation frequency, which seems to become closer to a natural frequency of the hydraulic system when the rotating frequency decreases. This result is coherent with the influence of the guide vane wave speed. In one case the natural frequency of the hydraulic system increases with the increase of guide vane wave speed, and in the second case the excitation frequency is reduced until it matches the hydraulic natural frequency. However, in both cases the amplitude of the pressure fluctuations in the vaneless gap remains constant.

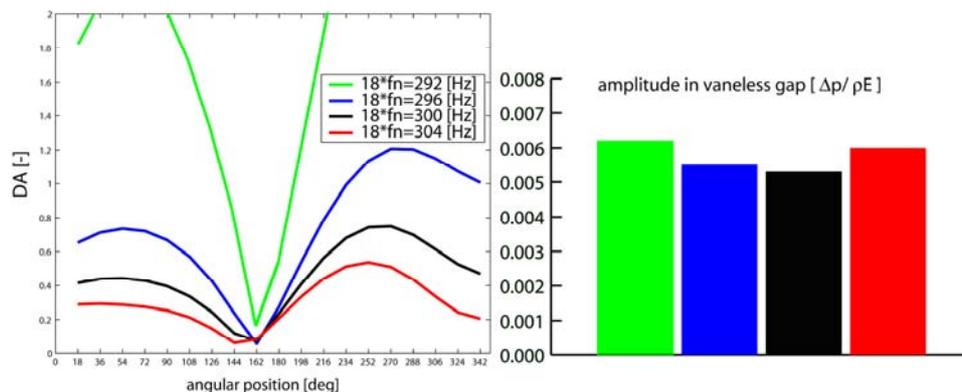


Figure 14 Influence of the impeller rotating speed for $f/f_b = 18$.

Conclusion

Rotor stator interaction in Francis pump turbine is characterized by rotating diametrical mode shapes that can excite natural frequency of the hydraulic system and resulting in standing wave in the spiral casing. This paper presents a new approach for the modeling of the hydroacoustic part of the phenomenon. A one-dimensional simulation model of Francis pump turbine is set up. The hydroacoustic parameters of the model are determined from the geometrical and structural characteristics of the machine and the test rig.

The resulting RSI model comprises a network of 180 valves connecting each guide vane with each impeller vane. The valve opening functions are driven accordingly with the flow distribution between the stationary part and the rotating parts. The analysis of the pressure fluctuations shows that rotating diametrical pressure mode and standing wave are properly simulated. Influence of the standing wave on the diametrical pressure mode is pointed out. The chosen excitation model appears to be realistic for the investigation of the hydroacoustic part of the RSI.

Moreover, a parametric study is carried out. It results that the wave speed in the impeller blades has not significant influence. Guide vane wave speed and impeller rotating frequency influence strongly the amplitude of the standing wave because of the closeness of a natural frequency of the hydraulic system.

Overall, this model gives satisfactory qualitative results and needs to be validated experimentally. However, if the dynamic amplification gives reliable trends, the determination of the absolute pressure fluctuations amplitudes is more challenging. The excitation model is based on the flow distribution between the stationary parts and the rotating parts of the machine and is arbitrary. Its parameters are only related to connection area between guide vanes and impeller blades evolution due to impeller rotation. To obtain a more reliable quantitative result, the excitation model should be improved and be able to take into consideration all the parameters affecting RSI like the gap amplitude, the wake angle of the impeller blade and the blades thickness etc. This could be achieved through the coupling of a one-dimensional code like SIMSEN for the modeling of the hydraulic system with a CFD tool for the modeling of the RSI excitation.

Acknowledgments

The authors would like to thank particularly all the partners of the HYDRODYNA Eureka Project No 3246, i. e. ALSTOM Hydro, EDF-CIH, GE Hydro, VA TECH Hydro, VOITH-SIEMENS Hydro Power Generation for their financial support and assistance. The HYDRODYNA project is funded by the CTI, Swiss Federal Commission for Technology and Innovation, Contract award No 7045.2 EUS. For this paper, the authors took advantages of the development of the SIMSEN hydraulic extension, developed in collaboration with P. Allenbach, Dr. A. Sapin, and Prof. J.-J. Simond from the EPFL Laboratory for Electrical Machines under the following contract awards: CTI No 5750.1 EBS, PSEL No 215 Scapin, EDF-CIH, HMD 420.210.3459.

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