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Dynamic modal analysis during reduced scale model tests of hydraulic turbines for hydro-acoustic characterization of cavitation flows

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Francis turbines operating at off-design conditions experience the development of unfavourable cavitation flows in the draft tube at the runner outlet, which induce pressure pulsations and hydro-acoustic resonances in the worst cases. The assessment of hydropower plant units at off-design conditions is possible by means of one-dimensional numerical simulation, which however requires a proper modelling of the draft tube cavitation flow. The corresponding hydro-acoustic parameters can be identified for a wide number of operating points on the reduced scale model of the machine by modal analysis of the hydraulic test rig. This identification approach is efficient but can however be time-consuming for an industrial project. The paper aims at proposing and validating a faster procedure to identify the eigenfrequencies and the corresponding eigenmodes of a hydraulic test rig featuring a reduced scale model of a Francis turbine operating in off-design conditions. The test rig is excited by injecting a periodical discharge with a rotating valve whose frequency linearly increases from 0 to 7 Hz. Based on the response of the test rig, measured by pressure sensors placed along the pipes, the eigenfrequencies and the corresponding eigenmodes are identified for several operating conditions. The hydro-acoustic parameters are then identified by using a one-dimensional numerical model of the test rig. The results are in very good agreement with those obtained with the standard procedure, i.e. with a stepwise increase of the excitation frequency. This new approach represents an important gain of time and might be applied to assess hydropower plant stability in an industrial context.

1. Introduction

The share of new and renewable energy sources (NRE), such as wind and solar, for the electrical energy supply has rapidly grown in the past decades and is expected to increase further in the future. The intermittent and stochastic nature of such energy sources however compromises the stability of the electrical power network. In this context, the flexible operation

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Nomenclature

a	wave speed (m s^{-1})
D	diameter (m)
Cc	cavitation compliance (m2)
E	specific energy (J kg^{-1})
f	frequency (Hz)
F	Froude number (⁻)
g	gravity acceleration (m s^{-2})
h	piezometric head (mWC)
H	head (mWC)
n	runner frequency (Hz)
n_{ED}	IEC speed factor (⁻)
P	output power (MW)
p	static pressure (Pa)
p_v	vapor pressure (Pa)
Q	discharge (m^3 s^{-1})
Q_{ED}	IEC discharge factor (⁻)
S_h	momentum source (m)
β	void fraction (⁻)
η	efficiency (⁻)
μ''	bulk viscosity (Pa s)
ν	specific speed (⁻)
Π	dimensionless wave speed (⁻)
ρ	water density (kg m^{-3})
σ	Thoma number (⁻)
ω	angular speed (rad s^{-1})
$NPSE$	net positive suction energy (J kg^{-1})
BEP (suffix)	Best Efficiency Point

capability of hydropower plants plays a crucial role by providing primary and secondary grid control and the balance between electrical energy production and consumption.

This however requires an operation of the generating units in off-design conditions, from low load to full load, to adjust their output power according to the network requirements. Such operating conditions are characterized by the development of unfavourable flow patterns involving cavitation [1] and pressure fluctuations that can lead to output power swings, as reported first by Rheingans [2] in the early 1940s. The resulting mechanical vibrations can reduce the life expectancy of the machine, see for instance Lowys et al. [3]. The development of cavitation flows and associated instabilities in hydraulic turbines can be detected by spectral analysis of structural vibrations and pressure fluctuations, as well as by acoustic emission [4,5].

In the case of Francis turbines operating at off-design conditions, a residual swirl is leaving the runner, leading to an inhomogeneous pressure distribution and the formation of a cavitation vortex rope in the draft tube of the machine. At full load conditions, i.e. with a discharge value higher than the value corresponding to the Best Efficiency Point (BEP), an axisymmetric cavitation vortex is developing along the draft tube cone centreline and can enter self-oscillations under certain conditions, inducing severe pressure pulsations in the whole hydraulic circuit and dangerous power swings (Müller et al. [6]; Valentin et al. [7]). At part load conditions, i.e. with a discharge value lower than the value at the BEP, a precessing vortex rope is observed in the draft tube cone. Its periodical precession motion around the cone centreline with a precession frequency comprised between 0.2 and 0.4 times the runner frequency (Nishi et al. [8]; Arpe et al. [9]; Favrel et al. [10]) acts as a pressure excitation source for the hydraulic circuit, leading to the propagation of pressure pulsations throughout the system at that same frequency. Hydro-acoustic resonances arise in the hydraulic system if the precession frequency of the vortex matches one of the eigenfrequencies of the test rig (Fritsch and Maria [11]; Favrel et al. [12]), which leads to a dramatic amplification of the pressure fluctuations. Visualizations of the cavitation vortex rope in Francis turbine draft tube are given in Fig. 1 at both part load and full load. At a very low discharge, i.e. at deep part load conditions, the development of inter-blade cavitation vortices can also be observed, which can induce additional erosion and pressure fluctuations on the runner blades (Yamamoto et al. [13]; Wack and Riedelbauch [14]).

Operating conditions in hydraulic machines are characterized by two dimensionless parameters, the speed factor $n_{ED} = n\,D/\sqrt{E}$ and the discharge factor $Q_{ED} = Q/D^2\,\sqrt{E}$ according to IEC Standards [15], where n (Hz) is the rotational speed of the runner, D (m) is the reference runner diameter, E (J kg^{-1}) is the hydraulic specific energy and Q (m^3 s^{-1}) is the discharge passing through the runner. The pressure level in the draft tube is defined by the Thoma number $\sigma = NPSE\,/\,E$, where $NPSE$ is the Net Positive Suction Energy defined as follows: