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Fluid-structure interactions in Francis turbines: A perspective review

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ABSTRACT

Competitive electricity prices and reduced profit margins have forced hydraulic turbines to operate under critical conditions. The demand for extended operating ranges and the high efficiency of the turbine runners have forced manufacturers to produce lightweight runners. A turbine runner sometimes experiences resonance when a forced (flow-induced) excitation frequency approaches the runner's natural frequency, resulting in failure. The cost of structural failure after commissioning is prohibitive. To attain a reliable and safe runner design, understanding of the structural response to flow-induced excitations is important. High amplitude pressure pulsations cause fatigue loading of the blades, which develop cracks over time. The amplitudes are dependent on the flow conditions, type of turbine and stator/rotor vane combinations. The structural response is dependent on the material properties, flow-induced damping and natural frequencies. Moreover, in a hydraulic turbine, changes in flow velocity from less than $1\ \mathrm{m\,s^{-1}}$ to over $40\ \mathrm{m\,s^{-1}}$ create challenges in predicting the response.

The main objective of this article is to review the studies conducted on fluid-structure interactions within hydraulic turbines. Several aspects are reviewed, such as flow-induced excitation, added mass effect, hydrodynamic damping, and blade flutter. Both experimental and numerical studies are discussed in this article. This review also discusses the consequences of an increased number of transient cycles, such as load variation, start-stop and total load rejection, on the turbines and the fatigue loading. Finally, an attempt is made to highlight the important requirements for prospective fluid-structure analysis to fill current gaps in the literature.

1. Introduction

Hydropower is an important source of renewable energy, contributing approximately 20% of global electricity generation [1]. Hydraulic turbines are generally operated at the design point at which the flow mainly follows the geometry, minimizing losses and thus optimizing hydraulic efficiency [2,3]. A flexible electricity market and competitive prices force the turbines to operate under off-design steady and transient conditions, under which the flow parameters cannot cope with the geometrical parameters [4,5]. Power generation at off-design conditions significantly affects the dynamic stability of the turbine [6,7]. A hydraulic turbine comprises both rotating (runner) and stationary (guide vanes) components, and the interaction between these components can induce high-amplitude dynamic pressure on the blades [8,9]. The blades are susceptible to fatigue-induced crack(s) when the number of fatigue cycles exceeds the threshold limit [10–12].

Fluid-structure interaction has played an important role in the design and development of hydraulic turbines over the last three decades [13,14]. Parameters such as added mass, damping, blade flutter, stress concentration, and fatigue loading have been investigated [15,16]. However, one of the main challenges is the estimation of runner natural frequency under a prototype operating condition. A Francis runner is an assembly of band, crown and highly skewed blades. Deformation in one of the components thus affects the other components and alters the vibration characteristics. The natural frequency reduction may be higher than 50% compared to that observed in air [17]. A large reduction in natural frequency becomes a concern as it approaches the frequency of forced excitation, i.e., rotor-stator interaction. When the reduced natural frequency coincides with the forced excitation frequency, resonance and sometimes failure can follow [18]. Moreover, in the competitive electricity market, turbine runners are manufactured using lightweight materials to achieve high efficiency at a reduced cost and accelerated start-stop conditions. The thin blades are prone to flutter at the resonant frequency, leading to fatigue cracks, sometimes early in their lives [5,19,20]. Several incidents of fatigue damage have been reported over the past decade

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[21–24].

The objective of this paper is to review the studies conducted on fluid-structure interaction in Francis turbines and to highlight opportunities for potential design improvements. Key challenges in obtaining reliable information of the critical parameters, flow induced excitation, added mass effect, hydrodynamic damping, blade flutter, and fatigue loading are discussed. Considering the recent trend of turbine operation and electricity generation, consequences for the turbine blades under transient operating conditions are discussed [4,25]. In the summary, the need for further studies on fluid-structure interactions in the turbines is discussed.

2. Forced excitation

In a hydraulic turbine, flow-induced vibrations are mainly associated with the draft tube vortex rope, Von Karman vortices, turbulence, cavitation and rotor-stator interaction (see Fig. 1). Vortex rope is a phenomenon observed in single regulated turbines, in which the blades are fixed. At part load, the small guide vane angle creates eddies that pass through the blade passages and interact with blade trailing edge vortices of high tangential velocity and swirling strength [26]. The developed helical vortex core creates a recirculation zone termed a rotating vortex rope (RVR). The observed RVR frequency varies from 0.2 to 0.4 times the runner rotational speed. The high amplitude pressure pulsations induced by RVRs can be decomposed into two components, namely, rotating and plugging (axial direction) [27]. Despite being a low-frequency phenomenon, vortex ropes lead to fatigue and cracks in the runner. Muntean et al. [24] found the maximum displacement at the trailing edge toward the crown after performing computational fluid dynamics (CFD) analysis and finite element analysis (FEA) at part load. Usually, the pressure pulsations induced by the vortex rope are mitigated by air injection. Experiments have shown that such injections may sometimes enhance the pressure pulsations [28]. At higher flow rates, the vortex rope becomes elliptical, and the frequency can be up to 5 times the runner speed [29,30]. The pressure pulsations have been found to be strongly related to the cavitation number. The pressure amplitudes may be large in model turbines but usually non-existent in prototypes [30]. The reason is the lack of Froude number similarity between model and prototype.

Von Karman vortices develop behind the trailing edge of a structure; their shedding frequency is related to the flow velocity and characteristic length. They may be present at the trailing edge of stay vanes, guide vanes and runner blades and are categorized as a high-frequency phenomenon. It is well known and documented that some turbines have experienced premature fatigue and cracks because of a lock-in between the frequencies of Von Karman vortex shedding and the blade natural frequency [31]. An oblique trailing edge type profile helps to reduce the vibration associated with the Von Karman vortices [32]. The oblique trailing edge allows simultaneous detachment of the alternate vortices, leading to their collision, which weakens the core of the vortices and decreases the vibration amplitude.

The frequencies of rotor-stator interaction and flow turbulence are generally near the runner natural frequencies. For mid-head to high-head Francis turbines, frequencies related to the rotor-stator interaction and their amplitudes are important [30]. High-amplitude pressure pulsations at the rotor-stator interaction frequency induce high cycle fatigue to the blades and sometimes catastrophic damage [10]. Fig. 2 shows the ratio of the stress amplitudes from rotor-stator interaction to the total dynamic stress expressed as one-half the peak-to-peak amplitude for different runner types. The values are based on strain gauge measurements conducted on Francis turbines at the rated operating condition [8]. Approximately 80% of the total stresses are related to rotor-stator interaction in high-head turbines. Sufficient distance between blades and guide vanes reduces the stress amplitudes, as observed in low-head turbines [33]. In this paper, forced excitation frequency refers to the frequency of flow-induced pressure pulsation due to rotor-stator interaction in the turbines. Resonance is one of the main challenges in high-head turbines because the frequency of forced excitation is close to the runner natural frequencies. The first natural frequency is generally observed between 70 and 300 Hz.

The rotor-stator interaction frequency and its amplitude can be estimated using CFD techniques [34–36]. Fig. 3 shows the dynamic pressure loading on a high-head Francis runner blade. The oscillations correspond to the frequency of forced excitation, 156 Hz, observed in the runner. The amplitudes are approximately 15% of the head. The frequency in the stationary and rotating domains is computed using Eqs. (1) and (2), respectively.

$$f_r = n \cdot Z_{gv} \quad \text{(Hz)} \tag{1}$$

$$f_s = n \cdot Z_b \quad \text{(Hz)} \tag{2}$$

where n is the runner speed in revolutions per second, and Z_{gv} and Z_b are the numbers of guide vanes and rotating blades, respectively. The developed pressure field due to rotor-stator interaction is computed using the following equations [35,37–39]:

Pressure field related to the runner:

$$p_r = \cos(mZ_b\theta_r + \phi_m). \tag{3}$$

Pressure field related to the guide vane:

$$p_s = \cos(nZ_{gv}\theta_s + \phi_n). \tag{4}$$

Combined pressure field:

$$p = A_{mn}\cos(nZ_{gv}\theta_s + \phi_n)\cos(mZ_b\theta_r + \phi_m), \tag{5}$$

$$p = A_{mn}\cos[mZ_b\Omega t - (mZ_b - nZ_{gv})\theta + \phi_m + \phi_n] + A_{mn} \cos[mZ_b\Omega t - (mZ_b + nZ_{gv})\theta + \phi_m - \phi_n]. \tag{6}$$

Eq. (6) is used to determine the pressure field as a function of space and time during rotor-stator interaction. The diametrical mode (k) due to rotor-stator interaction is estimated using Eq. (7):

$$k = mZ_b \pm nZ_{gv}; \tag{7}$$

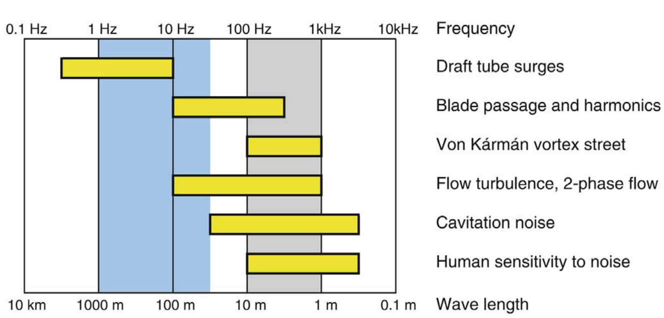

Fig. 1. Overview of the frequencies observed in a hydraulic turbine [30].

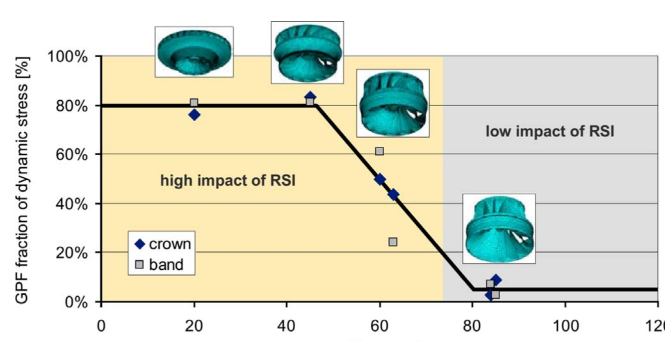

Fig. 2. Level of dynamic stresses caused by rotor-stator interaction in hydraulic turbine [8]. RSI=rotor-stator interaction frequency, nq=$N \cdot Q^{0.5}/H^{0.75}$; where N is the runner speed in revolutions per minute, Q is the flow rate in $m^3 s^{-1}$, and H is the head in m.