Practical Implications of Hydraulic Phenomena in Francis Turbines

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Abstract. Unexpected hydraulic phenomena in Francis turbines may have severe implications on the operation of new, upgraded, or existing units. This article describes several such phenomena, including the effects of inter-blade channel vortex cavitation, turbine runner uplift under overspeed conditions, runner seal induced unit vibration, high-load and high part-load instabilities, and draft tube vortex induced penstock resonance. Case studies are provided based on the author’s direct involvement with the respective projects. The article includes descriptions of the hydraulic phenomena, their impact on unit operation, investigations to eliminate or mitigate their effects, solutions implemented, and some lessons learned.

1. Introduction

Many hydraulic phenomena in Francis turbines are well understood, and performance characteristics are predictable with high accuracy based on similar design reference projects, hydraulic turbine model testing, and increasingly sophisticated transient and two-phase flow Computational Fluid Dynamics analyses. Nevertheless, some hydraulic phenomena lead to unexpected consequences on full-scale turbines (in this article referred to as ‘prototype’ turbines), which can result in undesirable restrictions of the available operating range, accelerated mechanical wear and tear, or premature unit failures.

This article presents several case studies of such phenomena and describes the impact on unit operation, investigations to eliminate or mitigate their effects, solutions implemented, and provides some learnings for future projects where applicable.

2. Case Study A: Inter-Blade Channel Vortex Cavitation

This power station was originally commissioned in 1969-71 and the turbines were upgraded with new design runners and guide vanes between 2005 and 2007. The vertical-shaft Francis units are nominally rated 131 MW, operating at 250 rpm synchronous speed over a net head range of ~145-175 m.

The upgraded turbines exhibit good efficiency, hydraulic stability, blade inlet and areal cavitation characteristics, largely better than contractual requirements. However, the occurrence of strong inter-blade channel vortex cavitation on the prototype was cause for concern.

Strong intermittent vortex cavitation was first observed on the hydraulic model under high net head and near the bottom end of the continuous power output range. Owing to the single regulation of Francis turbines, inter-blade channel vortices are always present once the discharge is reduced sufficiently below best efficiency discharge. Typically, inter-blade channel vortex cavitation is not considered particularly damaging if vortices are stable and their vapour cores do not touch blade surfaces. For this reason, their presence is generally considered acceptable in the lower temporary power output operating range, and sometimes even within parts of the continuous operating range.

However, for this particular runner design, the prototype inter-blade channel vortex cavitation was exceptionally strong and site testing confirmed its presence for discharge lower than ~65% of best
efficiency discharge (~67–73 MW output, depending on net head). The resulting effects of this inter-blade channel vortex cavitation can be summarised as follows:

i) High noise levels in the turbine pit and near the draft tube access hatch.

ii) Significant vibration of the embedded draft tube cone, resulting in discoloured water-based paste oozing from concrete to steel interface areas.

iii) High vibration levels of ancillary equipment in the vicinity of the draft tube cone, resulting in damage to the tailwater depression water level sight-glass and level switches, and the runner band seal cooling water supply pipework.

Following recommissioning of the first upgraded turbine the unit was deliberately operated extensively in the channel vortex cavitation range above the lower continuous power output limit. The contractual prototype cavitation erosion inspection after 8,000 hours showed no consequential damage to the runner. Having proven acceptability of channel vortex cavitation with regards to potential runner damage, the focus shifted to further site testing and agreement on an acceptable solution for reducing noise and vibration levels. The most effective and finally implemented arrangement consists of injecting ~200 l/s (FAD) of compressed air at ~8 bar into the runner band shroud area above the runner band seal. The air is distributed circumferentially around the band shroud via rotation of the runner and dragged into the draft tube cone via the seal leakage flow. It creates a dampening cushion between the draft tube cone wall and the collapsing channel vortex cavitation bubbles, and thus significantly reduces noise and vibration.

![Figure 1. Turbine cross-sectional drawing showing location of band shroud air admission.](image1)

![Figure 2. Ancillary equipment vulnerable to vibration and effects of draft tube cone vibration (after 8,000 h without air admission).](image2)

The inter-blade channel vortex cavitation observed on the prototype did not result in unacceptable shaft vibration, bearing housing vibration, spiral case inlet or draft tube cone pressure fluctuations. Therefore, care needs to be taken when developing specifications and contract conditions to ensure adequate mechanisms are included to deal with any unusual and undesirable phenomena.

The refurbished units at this station have been in operation for ~13 years. In recent years hydrological conditions and grid system demand constraints have resulted in operation of a few of the units in the channel vortex cavitation range for extended periods of time. The air admission solution has proven adequate, the turbine ancillary equipment is no longer exposed to excessive vibration, and the turbine runners have not shown any signs of cavitation erosion to date.
3. Case Study B: Runner Seal Induced Vibration and High-Load Instability

The vertical-shaft Francis units at this station are nominally rated 80 MW, operate over a net head range of ~138 – 158 m at a synchronous speed of 250 rpm, and were commissioned in 1977. Each unit at this station has on average required a major overhaul every ten years, including complete unit disassembly and significant turbine refurbishment work. This exceptionally high overhaul frequency is predominantly driven by excessive turbine related vibration, aggressive suction side blade inlet cavitation, and some generator failure incidents.

Unit vibration has been problematic since commissioning and appears to relate to turbine runner seal wear, which remains an ongoing problem. As there is no particulate erosion issue at this site, it is suspected that the runner seals make contact during high transient overspeed events due to a combination of two phenomena. Firstly, unit speed during high-load rejections reaches approximately 97% of full runaway speed, i.e. a considerable portion of the seal clearance is taken up by runner band expansion and high vibration associated with transient operating conditions. Secondly, to facilitate rapid response to small frequency deviations with long penstocks, the units are equipped with turbine relief valves (TRVs), which are sized at only 10% of rated turbine discharge. The TRVs are direct-acting via a dash-pot arrangement off the guide vane regulating ring. A drawback of this particular design is that once the TRV has fully opened, guide vane servomotor closure is controlled via the dashpot orifice rather than the servomotor orifices, resulting in an unbalanced force of ~46 t on the regulating ring and bearing housing, and thus causing a radial shaft displacement of ~0.3 mm, which is likely to contribute to the runner seal wear issue.

In addition, the turbines exhibit high-load hydraulic instability under high tailwater level conditions. Although the turbines are nominally of identical design, this instability is particularly prevalent on one of the units above ~75 MW. It manifests itself in high spiral case and draft tube pressure fluctuations, random draft tube vortex core implosions, significant axial shaft vibration and generator output power oscillation. The existing turbine runner crown and band shroud air admission assists with mitigating the effects of the hydraulic instability. In recent years it has become necessary to implement an auto-derating; unit load is reduced to 70 MW as soon as thrust bracket axial displacement in excess of 0.2 mm is detected for more than three seconds.

During recent hydraulic stability testing on one of the units a narrow band of significant hydraulic instability was observed immediately above best efficiency. This instability appears to relate to a resonance phenomenon in the crown and/or band shroud air admission pipework.

A turbine upgrade project is currently underway. Particular attention was paid to careful model test observations of hydraulic instability phenomena under all operating conditions, and to the adequacy of the proposed runner seal design. Factory acceptance testing of the replacement runners and guide vanes was recently completed, and the site installation work is scheduled to commence in early 2020.

4. Case Study C: Excessive Axial Up-Thrust

The 1950’s design vertical-shaft Francis units at this power station are rated 21 MW at ~25 m net head and 125 rpm synchronous speed. Despite a mass of the rotating components of ~167 t, there have been at least three incidents where the turbine runner developed sufficient up-thrust force for the rotating components to contact stationary counter-faces. Each time the consequence was considerable damage to bearing housings, covers and mounting flanges, and to generator rotor fan blades and stator air guide vanes. The weight of the rotating components is adequate to compensate for any up-thrust associated with operation in the normal operating range including load rejections. However, the turbine runner develops excessive uplift during emergency shutdown from overspeed conditions. Such phenomena are commonly experienced with Kaplan and propeller turbines and are caused by the rotation of the runner at significant speed in a static water column after guide vane closure. This effect can also be significant in high specific speed Francis runners due to their high degree of axial flow.

In order to better understand the uplift phenomenon, carefully controlled site tests were conducted, whereby the unit was operated disconnected from the grid at constant rotational speeds above synchronous speed, and a governor shutdown solenoid trip was initiated. Figure 3 shows pertinent test results recorded during such an emergency shutdown; the shaft lift tendency can clearly be seen at the end of the guide vane closing stroke. It should be noted that under the test conditions shown approximately 78% of the weight of the rotating masses was taken off the thrust bracket.
Figure 3. Shaft axial lift during governor emergency shutdown from 152% of synchronous speed.

Typical solutions to such uplift problems include experimentation with modified guide vane closing characteristics within the limits of hydraulic transient parameters, or installation of a suitably designed sacrificial turbine runner contact face under the turbine headcover. However, at this site the former could not readily be tested with the existing mechanical governors, and the latter was prohibitive due to the extensive outage durations associated with complete unit disassembly. Instead it was decided to design, manufacture and install an anti-uplift bearing on each unit, with any uplift forces being transmitted from the shaft coupling via sacrificial aluminium bronze wear pads to the underside of the thrust bracket. The bearing was designed for 110% of maximum up-thrust expected during an emergency shut-down from runaway speed conditions.

Figures 4 and 5. Arrangement drawing and photograph of anti-uplift bearing.
5. Case Study D: High Part-Load Instability
During commissioning a high part-load instability phenomenon was discovered on the vertical-shaft Francis turbines at this station, which operate under a net head of ~350 m and have a turbine runner diameter of 2.65 m. The high part-load instability is random in nature and occurs in a relatively narrow discharge range at about 85% of best efficiency discharge, i.e. well within the continuous operating range. Figure 6 clearly shows the somewhat random and sharp pressure fluctuations associated with the draft tube vortex instability.

![Figure 6](image)

This instability phenomenon resulted in significant axial vibration of the thrust bracket and in radial vibration of the dismantlable free-standing draft tube cone; in both cases fatigue considerations were the main concern. The solution implemented included:

i) Completion of a targeted site testing regime to gain a better understanding of the instability phenomenon, to capture data for fatigue assessments and to establish a performance baseline.

ii) Installation of central air admission and draft tube fins; as to be expected these mainly provided improvements in the typical part-load rather than the high part-load operating range.

iii) Completion of detailed fatigue analysis studies for all relevant components and structures, and implementation of identified necessary plant modifications.

iv) Implementation of a rigorous inspection and online monitoring program.

In recent years there have been many projects where such high part-load shock-type phenomena have occurred. They appear to be more prevalent in modern runner designs, which is likely to be exacerbated by fine-tuning hydraulic layouts for ever increasing performance requirements.

6. Case Study E: Draft Tube Vortex Induced Penstock Resonance
The two nominally 63 MW rated vertical-shaft Francis units at this underground power station operate under 206 m net head and at 428.6 rpm. The units suffer from random high part-load instability in the 52 – 57 MW range. They also exhibit large pressure oscillations, associated generator output oscillations and vibration predominantly in the 47 – 52 MW range; peak to peak output oscillation of up to 18% were present while operating in this load range. A targeted test campaign resulted in a better understanding of the instability phenomena and led to a suitable solution that could be implemented relatively inexpensively.

The root cause of the excessive pressure and power output oscillations was that the frequency of the part-load draft tube rope matched the second harmonic frequency of the penstock section downstream of the penstock bifurcation. This was evident as the magnitude of the oscillations depended on the loading of the neighboring unit: the oscillations were only present with the
neighboring unit shut down or lightly loaded, and it disappeared with increasing penstock flow. These characteristics were confirmed via a theoretical impedance model of the entire hydraulic system (refer Figure 8). It is suspected that a secondary resonance is caused by the generator natural frequency also being close to the draft tube vortex frequency.

Air admission via the draft tube cone (~60 l/s FAD) proved to be extremely effective in solving the resonance problem; it shifted the main draft tube vortex frequency and eliminated a range of higher frequency components, thus decoupling the system. Pressure fluctuations, output fluctuations and unit vibration were reduced to levels equivalent to those observed under best efficiency operation (refer Figure 7). Air admission was further very effective at masking the effects of the random instability phenomena. After completion of the test campaign a jet pump was installed to provide the required air admission whenever the unit is operating in the critical operating range, thus providing a relatively simple solution to a decade-old problem.

![Figure 7. Draft tube cone pressure versus time at 54.4 MW output: without air admission (blue), with draft tube air admission (red).](image)

![Figure 8. Impedance model of hydraulic system.](image)

7. Lessons Learned
While not all prototype hydraulic phenomena can be reliably predicted via turbine model testing, the contractual turbine model test provides a very important verification of the hydraulic design and every effort should be made to identify and investigate any undesirable characteristics. It is recommended to investigate relevant operating regions of interest in suitably small guide vane opening and net head steps, to cover the entire feasible prototype tailwater level range, and to adjust Thoma number reference levels over a range wider than normally applicable for runner blade cavitation observations.

Site commissioning testing should be carried out throughout as much of the discharge, net head and tailwater level operating ranges as practically possible in search for unwanted phenomena. Draft tube vortex related instabilities can be sensitive to backpressure and thus may only be present under certain tailwater level conditions. Often there are limitations on hydraulic conditions that can be tested during commissioning; limited repeat testing under different conditions may be necessary to complement original test results thus ensuring plant safety.

When preparing technical specifications and contract conditions, consideration needs to be given to the definition, identification and resolution of undesirable hydraulic phenomena. These are not always covered by traditional definitions of pressure oscillations, vibration or cavitation erosion. Also, specified design verifications should include rigorous frequency analyses of the turbine, waterways, generator, shaft system and connected electrical system.

8. Conclusions
Although performance characteristics of Francis turbines are nowadays well understood and to a large degree predictable, some hydraulic phenomena can result in problematic behaviour of prototype turbines. Solutions to undesirable hydraulic phenomena may include mitigation via air admission, implementation of non-operation zones and unit de-ratings, targeted monitoring of incremental damage to key components, or runner design modifications. The choice of solution adopted often depends on the financial impact that any plant restrictions may have, the risk of premature unit failure, and the cost of plant modifications that are necessary to alleviate the problem.