Transient Pressure Measurements in the Vaneless Space of a Francis Turbine during Load Acceptances from Minimum Load

R Goyal1, 2, B K Gandhi1, and M J Cervantes2

1 Indian Institute of Technology Roorkee, Roorkee, 247667 Roorkee, India
2 Department of Eng. Sci. and Math, Luleå University of Technology, 97187 Luleå, Sweden

E-mail: goel.rahul87@gmail.com

Abstract. Increased penetration of solar and the wind impels the designers of the hydroelectric power generation unit to provide more flexibility in operation for the stability of the grid. The power generating unit includes turbine which needs to sustain sudden change in its operating conditions. Thus, the hydraulic turbine experiences more transients per day which result in chronic problems such as fatigue to the runner, instrument malfunctioning, vibrations, wear and tear etc. This paper describes experiments performed on a high model (1.5:1) Francis turbine for load acceptances from the minimum load. The experiments presented in the paper are the part of Francis-99 workshop which aims to determine the performance of numerical models in simulations of model Francis turbine under steady and transient operating conditions. The aim of the paper is to present the transient pressure variation in the vaneless space of a Francis turbine where high-frequency pulsations are normally expected. For this, two pressure sensors, VL1 and VL2, are mounted at the vaneless space, one near the beginning of the spiral casing and the other before the end of the spiral casing. Both are used to capture the unsteady pressure field developed in the space between guide vanes and runner inlet. The time-resolved pressure signals are analyzed and presented during the transient to observe the pressure variation and dominant frequencies of pulsations.

Keywords: Hydropower, Francis turbine, transients, pressure fluctuations, vaneless space, rotor-stator interaction.

1 Introduction

The renewable energy from solar, wind, and hydro have shown greater interest over the last few decades. The electricity generation from solar and wind is not consistent and produces significant disruption in electricity grid network operation [1-2]. This needs to be balanced by a flexible source of electricity generation which is able to maintain the stability of electricity grid network. Hydropower plays an important role in order to maintain the grid stability. The speedy response of hydraulic turbine during peak and base load demand makes it enable to compensate the grid demand during intermittency. Consequently, this has led to the dynamic instabilities in turbine components due to random operation of the turbine to the grid network [2-3]. This has also significantly affected the overall performance of the turbines.

Load variations in a Francis turbine are accompanied by rapid opening and closing of the guide vanes. Flow through the turbine increases with the guide vane openings during load acceptances. The process starts from a steady state condition of the turbine when the guide vanes angle is set to a particular position, such as no load and the minimum load. The minimum load condition of the turbine is generally required to obtain synchronization between turbine and generator and after this; the required steady state condition is achieved. The runner angular speed does not change during load variation process because the turbine is connected to the generator which is rotating at a synchronous speed.
Experimental investigations are generally performed under steady state and few transients operating conditions. The pressure was measured in the stationary domains, namely vaneless space and draft tube of the turbine during different load variation conditions [4]. The maximum fluctuations in the turbine were associated to the instabilities of rotating and separated flow field in the vaneless space [2-3].

No measurements in the vaneless space during load acceptances from minimum load have been reported. The present measurements focus on the load acceptances operation from minimum load condition of a mode Francis turbine. The measurements are the part of a series of workshop Francis-99 which aim to provide an open access of the complete design and data of model Francis turbine during steady and transient operations (https://www.ntnu.edu/nvks/francis-99). The aim of the present study is to investigate the pressure fluctuations in the vaneless space during load acceptance from the minimum load.

2 Experimental apparatus and procedures

2.1 Model turbine

A high head model Francis turbine was used in the present investigation. The model turbine is a 1:5.1 scaled model of a prototype turbine installed at Tokke Power Plant, Norway. The complete description of the model and prototype turbines is reported in the literature [3]. The model turbine is installed on a test rig which can be operated in both open and closed loop configurations. The test rig is available at Water Power Laboratory (WPL) of Norwegian University of Science and Technology (NTNU), Norway. A schematic of the test rig is presented in Fig. 1. An open loop configuration was used for the present measurements in which the water from the basement can be pumped to the overhead tank. The water from the overhead tank then flowed down to the upstream pressure tank connected to the turbine inlet. The head to the turbine was maintained almost constant by keeping a uniform water level in the overhead tank. The draft tube of the model turbine was connected to a downstream pressure tank, which was open to the air, and the water was released back to the basement. The model turbine is integrated with 14 number of stay vanes, 28 number of guide vanes, a Francis runner with 15 number of splitters (Half-length blades) and 15 number of full-length blades, and an elbow-type draft tube. The discharge to the turbine was measured by a magnetic flow meter which is installed at the inlet pipeline to the turbine. A differential pressure transducer was used to measure the head across the turbine.

![Schematic diagram of model Francis turbine test rig installed at Water Power Laboratory, NTNU.](image-url)
2.2 Instrumentation and Calibration

The instrumentation and calibration were carried out according to the guidelines available in IEC and ASME-PTC standards on hydraulic turbines [5-7]. The pressure measurements were recorded using a National Instruments (NI) Compact Reconfigurable Input /Output (cRIO) model 9074 (≤ 400 MHz) with a 24 bit, ±60 V analog to digital converter (ADC). The data were sampled at 5 kHz with a separate ADC for each channel. The operating flow parameters such as discharge, inlet and differential pressures, atmospheric pressure, the angular speed of the runner, shaft torque to the generator, bearing friction torque, turbine axial force, and guide vanes angular position were acquired simultaneously through the same data acquisition system. The discharge was measured using a magnetic flow meter (KROHNE IFS 4000 series). In addition to the base instrumentation, two sensors were mounted in the vaneless space, one near the beginning of the spiral casing and one near the end as shown in Table 1 and Figs. 2 (a-b). The radial distance of the sensors in Table 1 is made dimensionless by dividing with the runner radius \( r = 174.5 \text{ mm} \). The uncertainty of the instruments used in the measurements was determined according to the guidelines available in IEC 60041/60193 [5-7]. The uncertainties of the calibration of the pressure sensors along with the accuracy, as provided by the manufacturer, are described in the literature [8-10]. The total estimated uncertainty was ±0.15% in the hydraulic efficiency at best efficiency point (BEP).

![Figure 2](image-url)  
Figure 2. Sensor placements in the model Francis turbine, (a) Top view, (b) Side view. The positions 1A and 1B are the placements of pressure sensors VL1 and VL2.
Table 1 Position of the pressure sensors

| Sensor | Placement | Dimensionless Radial Distance | Type | Uncertainty |
|--------|-----------|------------------------------|------|-------------|
| VL1    | 1A        | 1.23                         | Kulite | 0.01%       |
| VL2    | 1B        | 1.84                         | Kulite | 0.01%       |

2.3 Measurement program

The operating conditions for the measurements were inspired by the operating conditions presented in Francis-99 (I), but with some changes. The transient measurements were preceded and followed by corresponding steady conditions on a model Francis turbine. In all measurements, the runner speed in revolutions per minute (rpm) i.e. 333, was maintained constant to achieve the realistic operating conditions as that of the prototype. The maximum hydraulic efficiency (92.4 %) of the model turbine was estimated at guide vane angle ($\alpha$) = 9.8°, speed factor ($n_{ED}$) = 0.179, and discharge factor ($Q_{ED}$) = 0.15, which was marked as BEP. The modified part load (PL) and high load (HL) conditions were used in order to get the similar runner speed as that of the minimum load. The measurements were performed in an open loop configuration to get the realistic condition without significant variation in effective head available to the turbine. The specifications of the steady state operating conditions used are presented in Table 2.

Table 2 Steady state operating points

| Guide vane angle $\alpha$ (°) | Head (m) | Flow rate $Q$ (m$^3$/s) | Speed factor $n_{ED}$ (-) | Discharge factor $Q_{ED}$ (-) | Hydraulic efficiency $\eta$ (%) |
|-------------------------------|---------|--------------------------|---------------------------|-------------------------------|--------------------------------|
| 0.8                           | 12.14   | 0.02                     | 0.18                       | 0.01                          | 20.9                           |
| 6.7                           | 11.87   | 0.14                     | 0.18                       | 0.11                          | 90.1                           |
| 9.8                           | 11.94   | 0.20                     | 0.18                       | 0.15                          | 92.4                           |
| 12.4                          | 11.88   | 0.24                     | 0.18                       | 0.18                          | 91.8                           |

The speed factor and discharge factor in Table 2 can be computed as:

\[
Q_{ED} = \frac{Q}{D^2 \sqrt{E}} \quad (-)
\]

\[
n_{ED} = \frac{nD}{\sqrt{E}} \quad (-)
\]

where, $Q$ is the discharge in kg m$^{-3}$, $D$ is the reference diameter of runner in m, $E$ is the specific hydraulic energy in J kg$^{-1}$, $n$ is the angular speed of runner in s$^{-1}$.

Transient measurements were performed by changing the angular positions of the guide vanes from the minimum load (0.8°) to PL (6.7°), BEP (9.8°) and HL (12.4°). The angular position of guide vanes was measured by the voltage signal from the servomotor mechanism. A LabVIEW program with linear coefficients of voltage signals was made to control the exact movement of the guide vanes. A linkage mechanism with servomotor was used for the movement of the guide vanes. The uncertainty
of ± 0.044° in recorded positions of the guide vanes was always available because guide vanes were acting against the hydrodynamic forces.

3 Results and Discussion

The obtained results of flow variables ($H$, $Q$, $\alpha$, and $T_{\text{GEN}}$) and vaneless space pressure measurements for transients are presented and discussed. Detailed investigations of load acceptance from minimum load are presented with special emphasis pressure fluctuations and frequency content, i.e., rotor-stator interaction and standing pressure waves.

3.1 Flow variables

The representation of turbine load acceptances from the minimum load (ML) with a variation of basic flow variables such as the head ($H$), discharge ($Q$), torque ($T_{\text{GEN}}$), and guide vane position ($\alpha$) of model Francis turbine are shown in Figure 3. The values $\alpha$, $Q$, $H$, and $T_{\text{GEN}}$ in figures were normalized with the maximum value corresponding to 1 (100%) and a minimum value corresponding to 0 (0%) taking the uniform length of the signal using Eq.1.

$$X[-] = \frac{X - X_{\text{min}}}{X_{\text{max}} - X_{\text{min}}}$$

The generator in the WPL was connected to a direct current (DC) converter. The converter unit supplies and takes power to and from the generator so that the selected operating point could be maintained. The converter also fixes the rpm of the generator at the selected value of the turbine runner. In the present study, the generator was rotating at a constant angular speed of 333 rpm which was the synchronous speed. During the measurements at load acceptances, the guide vanes of the model Francis turbine were opened up to the required operating condition. The average angular speeds ($\omega_{\text{gv}}$) of the guide vanes movement during the measurement was 0.0022 rad/s.

The transient variation of discharge and generator torque during three different load acceptances process almost follow the trend of guide vanes movement as shown in Figs. 3 (a-c). The maximum output torque from the generator was 9.4, 457.2, 695.9 and 791.4 N-m at ML, PL, BEP and HL operating conditions, respectively. The torque fluctuations (Fig. 3b) for the opening of the guide vanes at BEP were attributed to the oscillations in runner angular speed. The trend of head variation during all transients was almost similar. The head was the most effective parameter during the load acceptances. Initially, a sudden drop in head values was observed up to 3-4 s after the transient. Thereafter, head starts to increase and settles around 11 s after the guide vanes stop. The range of the head variation during the load acceptances from ML was 11.76 to 12.64 m.
3.2 Pressure fluctuations

The time-dependent pressure fluctuations during load acceptances from a minimum load of the turbine are shown in Fig. 4. The two sensors, namely VL1 and VL2, located in the vaneless space were selected to show the pressure variation in the turbine. The values of guide vane angle \( (\alpha) \) and pressure \( (P_{\text{norm}}) \) in Fig. 4 was normalized using Equation (1). During load acceptance from minimum load to part load, the overall pressure in the vaneless space was observed to increase from 148 kPa to 163 kPa (9 %) and the transient pressure fluctuations in the vaneless space (VL1 and VL2) was observed to decrease by ±2.1 % after the transient as shown in Fig. 4 (a). The pressure fluctuation in the vaneless space was attributed to the wakes captured in the vaneless space by the interaction of runner blade and guide vanes.

During load acceptance from minimum load to best efficiency point, the overall pressure in the vaneless space was observed to increase from 148 kPa to 167 kPa due to rapid opening of the guide vanes as shown in Fig. 4(b). The transient pressure fluctuation in the vaneless space was decreased by ±0.7%. The absolute pressure in the vaneless was observed to increase from 148 kPa to 178 kPa during the load acceptance from the minimum load to high load as shown in Fig. 4 (c). The transient
pressure fluctuation in the vaneless space was decreased by ± 1.1 %. The different cycles of pressure signal fluctuation in the vaneless and draft tube (Fig. 4) show the existence of high and low-frequency oscillations in the draft tube which is discussed later in the next sections.

Figure 4. Transient variation in normalized pressure at sensor location VL1 and VL2 during turbine load acceptances from minimum load; (a) Minimum load to part load, (b) Minimum load to best efficiency point, (c) Minimum load to high load.

3.3 Spectral analysis

Spectral analysis of pressure time signals was carried out to investigate the dominant frequencies of the fluctuations in the vaneless space of the turbine. Welch’s method with Hanning window on the fluctuating part of the pressure and velocity data was used for the spectral analysis. A built-in function of MATLAB spectrogram was used to present the time-dependent spectral analysis. The function works on the Goertzel algorithm. The window size of 4 s and 2 s with 95% overlap was selected for the spectral analysis of the pressure time signals. The color bar in the spectrograms shows the power spectral density (PSD) of the frequency analysis in logarithmic scale. Equation (2) was used for the PSD analysis, presented in the spectrograms. The spectrograms for the load acceptances from minimum load are presented and discussed.

$$PSD_{log} = 10 \times \log (10 \times PSD)$$  \hspace{1cm} (2)
The PSD analysis of the pressure signals VL1 and VL2 (see Table 1 and Fig. 2) during load rejection and acceptances is presented in Figures 5 to 7. PSD shows the strength of the corresponding properties (pressure) as a function of frequency. It shows the strong and weak frequency variations in any signal. The y-axis in the Figs. 5 to 7 is normalized using the runner rotational frequency ($f_n$) of 5.55 Hz.

**Figure 5.** Spectrograms of pressure time signals during load acceptance from minimum load to part load; (a) VL1, (b) VL2. Black solid line is the guide vanes opening with y-scale to the right.

**Figure 6.** Spectrograms of pressure time signals during load acceptance from minimum load to best efficiency point; (a) VL1, (b) VL2. Black solid line is the guide vanes opening with y-scale to the right.

The spectrograms of the all load acceptance cases show that for the pressure signals at VL1 and VL2, the normalized blade passing frequency (30) is present in the turbine as shown in Figures 5 to 7. The harmony (15) of the blade pass frequency is also observed in the vaneless space during load acceptances. The PSD strength of the blade pass frequency was observed to increase with the higher load operating condition of the turbine. The maximum strength was observed at HL operating condition of the turbine due to reduced vaneless space between guide vanes trailing edges and runner leading edges. Seen in spectrograms, the pressure signal at location VL1 shows more PSD strength as
compared to that of the VL2. This may be attributed to higher flow rate near VL1 location which is located near the beginning of the spiral casing. The normalized frequency of 18 (100 Hz) was observed in both the pressure signals (VL1 and VL2). This frequency is always present in the signals and is attributed to the one-third harmonics of the three-phase rectifier and second harmonics of the grid frequency. It is believed that the frequency will not be available in the system with the generator off.

Other normalized frequencies of 7.5, and 2.9 were also observed in the spectrograms during load acceptances. Assuming a sound wave velocity of 932 m/s, normalized frequencies of 7.5 and 2.9 were calculated for the upstream and downstream pipe lengths to the turbine. The corresponding lengths are 14.5 and 5.6 m, starting from the pressure tank water level to the runner inlet and the runner outlet of the downstream tank water level, respectively. The standing wave frequencies (7.5 and 2.9) start to appear in the system after the transient. During the load acceptance from ML to PL, the normalized frequency of 7.5 was observed to appear first at the location VL1 and then VL2. Significant PSD strength for another standing wave frequency of 2.9 was observed only at location VL1. During the load acceptance from ML to BEP, the normalized frequency of 7.5 and 2.9 was observed to appear simultaneously at the locations VL1 and VL2. Again the normalized frequency of 7.5 was observed to appear first at the location VL1 and then VL2 during load acceptance from ML to HL. The maximum PSD strength of the standing wave frequencies of 7.5 and 2.9 was observed at location VL1.

Figure 7. Spectrograms of pressure time signals during load acceptance from minimum load to high load; (a) VL1, (b) VL2. Black solid line is the guide vanes opening with y-scale to the right.

4 Concluding remarks

The measurements for load acceptances operations of Francis turbine from the minimum load were performed to investigate the pressure variation and dominant frequencies of fluctuation in the vaneless space. The oscillations in generator torque were observed due to fluctuations in runner rotational speed during load acceptance from ML to BEP. Head was observed as the most affected parameter during load acceptances and the range of head variation was 11.76 to 12.64 m. The maximum pressure fluctuations (± 2.1%) in the vaneless space were observed for the load acceptance condition from ML to PL. The absolute pressure in the vaneless space was ranging from 148 to 178 kPa for all cases. Spectrograms of the pressure signals showed that the PSD strength of blade passing frequency was
most dominant in the turbine vaneless space. The maximum strength was observed during the load acceptance from ML to HL due the reduced space vaneless between the guide vanes trailing edges and runner blades leading edges. The normalized frequencies of 7.5 and 2.9 were observed and explained as resulting from standing waves in the system due to the downstream and upstream boundary conditions. The maximum PSD strength of the signals was observed at the location VL1 which is located near the beginning of the spiral casing.

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