Francis-99 Workshop 3: Fluid structure interaction

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Abstract. Francis-99 is a series of three workshops, which provides an open platform to the hydropower researchers. It gives the possibility to explore capabilities/skills on futuristic turbine design and development. Under the Francis-99 test cases, complete design and data of a Francis turbine are provided. The measurements are conducted on state-of-the-art facility at the Waterpower Laboratory, NTNU. The first workshop was organized during 15-16 December 2014, which focused on the steady state operating condition of the Francis turbine, i.e., best efficiency point, part load and high load. The second workshop was organized during 14-15 December 2016, which focused on the transient operating conditions, i.e., load variation and start-stop. The third workshop was organized during 28-29 May 2019, which focused on fluid structure interactions. In the third workshop, two test cases were provided: (1) Hydrofoil and (2) Francis turbine. The hydrofoil test case aimed to investigate fundamental research, and the turbine test case aimed to investigate applied research. https://www.ntnu.edu/nvks/f99-third-workshop

1. Introduction
Flexible electricity demand and low profit margin have pushed hydro turbines to their extreme limit. The turbines are operated at unfavorable loads, which has raised concerns and challenged the existing design philosophy. The critical requirement for modern turbines is high efficiency over the wide operating range. In addition to the experiments, computational fluid dynamic (CFD) techniques have been extensively used in turbine design [1]. Experiments with CFD optimization managed to overcome the challenges up to certain extent. However, while crossing the threshold limit, there are significant challenges in the numerical modelling due to complex flow condition in the turbine. A verified and validated technique applied to an operating load may not work for another load [2]. Therefore, there is a need proper verification and validation of numerical techniques, especially for hydro turbines. Due to confidentiality, the turbine design approaches, adopted by the manufacturers, are not available in public domain. This makes difficult for engineers/researchers to explore their skill and knowledge in evaluating the turbine designs. The Waterpower Laboratory through Francis-99 provides an open access of turbine design and data to the researchers. They can work on the open test cases enhance their knowledge, capabilities and skills. The researchers can use these data and perform numerical studies by applying different tools and techniques.

Third Francis-99 workshop is the continuation of previous two workshops. The first workshop aimed to study the steady state operating conditions [3,4], and the second workshop aimed to study the transient operating conditions (load variation and start-stop) [5]. The third
workshop focused on fluid structure interaction (FSI) analysis under steady state operating condition. Unlike the previous two workshops, the third workshop provides extra test case of hydrofoil that allows simplified study of FSI to understand the mechanics behind the fluid structure coupling. A turbine includes stationary and rotating components. The interaction between the runner blades and guide vanes, is critical when the frequency of the rotor-stator interaction approaches the runner natural frequency. In recent years, several turbines have been exposed to heavy fatigue loading and crack propagation in the blades [6]. The fatigue loading and the failures are associated with the hydrodynamic force and the response from the mechanical structure for the given condition. For safe and reliable design of the turbines, detailed understanding of FSI is essential. However, hydro turbine is a complex structure and extremely challenging to understand the behavior as mechanical response is dependent on the operating condition. While designing the turbine, factor of safety based on traditional design and experience is considered. However, it is not proved to be reliable all the time. The FSI is dependent on several parameters: (1) hydrodynamic damping, (2) nearby structure and submergence level, (3) mode-shape, (4) freestream velocity and vortex shedding, (5) damping during cavitation, (6) material properties, (7) rotational speed, (8) natural frequency of individual and combined structure, (9) flow compressibility, (10) wave propagation speed [7,8].

2. Scope of the workshop
Scope of the third workshop is fluid structure analysis under steady state operating conditions. More specifically, parameters such as study of mode-shape, nodal-diameter, deformation, fatigue loading, estimation of fatigue life, individual/combined natural frequencies, hydrodynamic damping, harmonic response, etc. were investigated. The Hydrofoil test case focused on fluid structure analysis (one-way or two-way). Fundamental research is the main objective and how the approach (applied on hydrofoil) can be useful for the complex structure such as turbine blades. CFD analysis of the hydrofoil encouraged if sophisticated turbulence modeling approaches such as, detached eddy simulation, large eddy simulation and direct numerical simulation, are used that will help to understand the mechanics of vortex shedding and the resonance. In addition to the harmonic response, modal analysis, modal reduction, acoustic model, two-way FSI was encouraged. The turbine test case focused on fluid structure analysis (one-way and/or two-way) and optimize the time/cost of numerical modelling.

3. Test cases
3.1. Hydrofoil
The test facility was developed in the Waterpower Laboratory, which consisted of long piping loop and a test-section for the hydrofoil [9]. The aim was to determine the added-mass, natural frequencies and hydrodynamic damping with flow velocity. To measure flow rate in the test-section, magnetic flowmeter (uncertainty is in the order of 0.1%) was mounted at far downstream of the test-section. The test rig is comprised of an upstream convergent and downstream diffusor, these change the pipe section from 0.3 m diameter pipe to a square 0.15 m × 0.15 m, measured internally. The hydrofoil is in this square section along with most of the instrumentation. The test section was designed to ensure enough stiffness, in order to provide a grounded support for the test structure, i.e. the hydrofoil, as is good practice for modal testing. This also enables the assumption of stiff walls for boundary conditions when performing structural simulations. The hydrofoil geometry was long and slender (figure 1), with a thickness of 0.012 m and a cord length of 0.25 m. After 0.15 m from the leading edge, the thickness was tapered down to 4.5 mm at the trailing edge, before being chamfered and rounded on one side, as is normal for runner blades. The hydrofoil was milled from a single piece of aluminum alloy, and grooves were milled for instrumentation and cables. The hydrofoil was mounted zero-degree angle of attack and was then excited using piezoelectric MFCs actuators from PI Ceramic. Two MFCs were mounted
on each side of the foil at the width wise center, close to the trailing edge. By exciting these in a sinusoidal pattern phase-separated by 180°, a bending action is induced in the blade. As the piezoelectric patches are driven by a high voltage signal and is therefore supplied by E-835 DuraAct Piezo Driver Module.

![Figure 1. Hydrofoil (span = 0.15 m).](image)

The tests were performed using a stepped-sine excitation, in which a series of constant-frequency excitations are performed in order to avoid transient effects when moving through a resonant region. Each measurement consisted of around 60 excited frequencies, and each measurement was repeated several times in order to obtain sufficient statistics for an assessment the uncertainty in both damping and natural frequency. The tests were performed with water velocity 0-25 m s⁻¹, with steps of 5 m s⁻¹. The test rig was pressurized in order to obtain cavitation-free conditions.

![Figure 2. Natural frequency of hydrofoil with respect to flow velocity.](image)

![Figure 3. Hydrodynamic damping with respect to flow velocity.](image)

### 3.2. Turbine

A model Francis (known as Francis-99) turbine available at the Waterpower Laboratory, NTNU, was used for the measurements. The turbine includes 14 stay vanes, 28 guide vanes, a runner with 15 blades and 15 splitters, and an elbow-type draft tube. The runner inlet and outlet diameters are 0.63 m and 0.347 m, respectively. Detail about the test facility and the turbine is available in the literature [10,11].

Pressure measurements in the blade channel were conducted under resonance condition. One of the concerns while measuring pressure loading was the complex assembly. The runner blades are joined to the crown and band with bolts, which means there is a possibility for rotational
asymmetry related to the blade mounting. In addition, the blades joined to the band, at the trailing edge, are free (around 94 mm length). In this area, variation in gap between the blades and the band was observed. By exciting the blades with the impact hammer, it was clear from the sound response that the blades have different pre-loading (possibly pretension). A second concern was the crown with several holes and cable channels for sensors and data acquisition system. This may create asymmetrical stiffness, meaning the bending stiffness of the crown is dependent on the bending direction. Hence, there is possibility of high uncertainty in the strain measurements at these locations. Considering the concerns, the turbine test case was limited for specific (trustworthy) study of FSI based on pressure measurements only. The data of pressure amplitudes pertained to rotor-stator interaction frequencies and the harmonics are provided.

Figure 4 shows locations of pressure sensors in the runner. The guide vane passing frequency $f_{gv}$ and the harmonic are determined at the sensor locations. Measurements are conducted at five operating points, keeping same opening angle of guide vanes; however, the rotational speed of the runner and the head values are different, which are presented in table 1. From these measurements, we found that the amplitudes of RSI frequency varied significantly and at certain operating condition, the amplitudes are relatively high [12]. This may be the indication of resonance in the turbine at that operating condition. This could be interesting to investigate numerically.

![Figure 4. Locations of pressure sensors in the blade channel.](image)

| Operating point | Flow rate (m$^3$ s$^{-1}$) | $n_{ED}$ | $Q_{ED}$ | Head (m) | Speed (rpm) |
|-----------------|-----------------------------|---------|---------|-----------|-------------|
| BEP-1           | 0.134                       | 0.179   | 0.154   | 5.2       | 219.8       |
| BEP-2           | 0.160                       | 0.176   | 0.156   | 7.2       | 254.3       |
| BEP-3           | 0.183                       | 0.178   | 0.154   | 9.6       | 297.8       |
| BEP-4           | 0.209                       | 0.178   | 0.155   | 12.6      | 340.5       |
| BEP-5           | 0.232                       | 0.180   | 0.154   | 15.55     | 381.7       |

The pressure sensors were initially calibrated statically with the use of dead weight tester as primary reference. The resonance frequencies of pressure sensors are above 10 kHz, hence it is assumed that the dynamic uncertainty is very low and only repeatability and hysteresis remains in the uncertainty due to covariance. A repeatability test was conducted at 1 Hz with a pressure alternating between 100 kPa and 90 kPa absolute pressure. Amplitudes of pressure fluctuations pertained to rotor stator interactions are presented in table 2. A vibration test with the runner in air was conducted to analyze the pressure sensors vibration sensitivity. The results did not give any additional uncertainty. The RSI amplitudes are normalized for head at that operating
point. The uncertainty includes the 95% measurement uncertainty and 95% probability of the amplitude variation found from with the use of short time fast Fourier transform. The analysis was performed with window length equal to 100 periods of the RSI signal and with 50% overlap.

Table 2. Amplitudes pertained to rotor stator interaction \(f_{gw}\). The amplitudes are in percentage of head value at the corresponding operating point.

| Location | BEP-1    | BEP-2    | BEP-3    | BEP-4    | BEP-5    |
|----------|----------|----------|----------|----------|----------|
| R1       | 0.979 ± 0.034 | 0.053 ± 0.024 | 1.077 ± 0.021 | 1.074 ± 0.019 | 1.082 ± 0.018 |
| R2       | 0.714 ± 0.036 | 0.775 ± 0.027 | 0.802 ± 0.020 | 0.803 ± 0.015 | 0.818 ± 0.014 |
| R3       | 0.550 ± 0.025 | 0.591 ± 0.018 | 0.605 ± 0.015 | 0.611 ± 0.013 | 0.623 ± 0.011 |
| R4       | 0.295 ± 0.024 | 0.337 ± 0.023 | 0.348 ± 0.023 | 0.348 ± 0.015 | 0.359 ± 0.013 |

Figure 5. Amplitudes of pressure fluctuations of rotor stator interactions (second harmonic).

Table 3. Amplitudes pertained to rotor stator interaction \(2f_{gw}\). The amplitudes are in percentage of head value at the corresponding operating point.

| Location | BEP-1    | BEP-2    | BEP-3    | BEP-4    | BEP-5    |
|----------|----------|----------|----------|----------|----------|
| R1       | 0.086 ± 0.034 | 0.109 ± 0.029 | 0.152 ± 0.024 | 0.091 ± 0.020 | 0.076 ± 0.017 |
| R2       | 0.072 ± 0.033 | 0.097 ± 0.028 | 0.167 ± 0.024 | 0.091 ± 0.021 | 0.059 ± 0.017 |
| R3       | 0.056 ± 0.020 | 0.080 ± 0.019 | 0.154 ± 0.019 | 0.072 ± 0.016 | 0.041 ± 0.011 |
| R4       | 0.042 ± 0.031 | 0.050 ± 0.026 | 0.124 ± 0.023 | 0.045 ± 0.019 | 0.025 ± 0.016 |

The second harmonic of the guide vane passing frequency \(f_{gw}\) has an increasing trend towards the measurement at 280 Hz as shown in figure 5. This indicates a resonance situation with nodal-diameter 4. The objective of third workshop is to perform FSI simulation to analyze the pressure in the runner channel during the resonance. Only one CFD analysis may be enough to find the excitation pressure. To avoid CFD analysis, other excitation methods could
be used to excite the runner with a nodal-diameter 4 pressure field. One can replicate the shape presented in figure 5 using FSI analysis; amplitudes are not necessary to match exactly with the experimental data.

4. Summary
The third workshop was organized during 28-29 May 2019, which focused on fluid structure interactions in Francis turbine. Two test cases were provided: (1) Hydrofoil and (2) Francis turbine. The hydrofoil test case aimed to investigate fundamental research, and the turbine test case aimed to investigate applied research. Parameters such as study of mode-shape, nodal-diameter, deformation, fatigue loading, estimation of fatigue life, individual/combined natural frequencies, hydrodynamic damping, harmonic response, etc. were investigated. In total 11 papers were submitted to the workshop and around 35 participants from different countries, Canada, Spain, Germany, Croatia, Sweden, North Macedonia, Norway, etc., joined the workshop.

On the first day, the workshop was started with opening speech by Chairman, Professor Torbjørn K. Nielsen then summary of Francis-99 workshop by Chirag Trivedi. The first keynote presentation delivered by Professor Eduard Egusquiza from UPC Spain. The second keynote presentation was delivered by Dr. Bjarne Børresen (Multiconsult-Norway), “Runner cracking – ten years later.” He emphasized needed research to reduce the risk of runner crack and requirement of credible as well as combined effort to overcome the cracking problem. After the keynote speech, research from different authors of Francis-99 papers was presented.

Second day was started with keynote speech by two eminent professors from Laval University-Quebec, Professor Claire Deschênes and Professor Sébastien Houde. Presentation was focused on measurements on propeller turbine, Francis turbine, fluid structure interactions, strain gauge measurement, computational fluid dynamic analysis, basic and applied research, etc. Then, authors of Francis-99 papers presented their work. Professor Ole Gunnar Dahlen led discussion on next Francis-99 workshop series. Should we continue the Francis-99 workshop series and, if yes, what should be test case? The discussion was very much informative extended beyond one hour. Several researchers provided their wishes for next workshop/series. Large part of discussion was focused around hydrofoil type test case for future workshop that will help to understand fluid structure interactions in detail with robust verification and validation.

The workshop was ended with closing remark from the Chairman, and the acknowledgment by Chirag Trivedi. The workshop series was contribution of several researchers, authors and universities during 2009-2019. This could have not been very successful without teamwork and dedicated effort as well as financial support from different research projects in the Waterpower Laboratory. Later, laboratory visits were arranged that included smart grid laboratory, smart house and the Waterpower Laboratory.

All the geometries and the experimental data can be downloaded from the official website (https://www.ntnu.edu/nvks/francis-99)

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- Chirag Trivedi, Editor-in-Chief
- Ole Gunnar Dahlhaug, Editor
- Pål-Tore Selbo Storli, Editor
- Torbjørn Kristian Nielsen, Editor

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