Operational Modal Analysis of Francis Turbine Runner Blades Using Transient Measurements

M Gagnon1, Q Dollon2, J Nicolle1 and J-F Morissette1

1 Institut de recherche d’Hydro-Québec (IREQ), Varennes, QC, J3X 1S1, Canada
2 École de technologie supérieure, Montréal, QC, H3C 1K3, Canada

gagnon.martin@ireq.ca

Abstract. To prevent dynamic amplification or resonance during normal operation, which could result in large blade deformations and premature fatigue failure, structural mode frequencies are predicted using numerical models during design. However, such numerical models could benefit from validation using data obtained during commissioning. Here, we propose the use of asynchronous transient experimental measurements to identify regions where interaction between structural modes and Rotor-Stator Interaction (RSI) harmonics can be detected. From these regions, modal parameters can be estimated and compared with numerical predictions. This paper presents the methodology used to extract these modal parameters and estimate their uncertainties using data from a Francis runner recently commissioned by Hydro-Québec.

1. Introduction
Turbine runners are designed to prevent interactions between structural modes and RSI generated synchronous pressure pulsations that could result in large blade deformations and premature fatigue failure. During design, structural mode frequencies are predicted using numerical models to avoid any such interaction. However, these numerical models must be validated. Measurements made during runner commissioning provide the first opportunity to do so. Models can be validated by attempting to match the frequencies detected in operation with those obtained by simulation [1] or by using Operational Modal Analysis (OMA) [2]. OMA uses the structural response from background random excitation during steady state operation to extract modal parameters. However, a rotating structure such as a turbine runner can exhibit a coupled response due to the close proximity of several of its modes. Thus, mode separation using classical methods is difficult to implement. Furthermore, as soon as the structure begins to rotate, RSI limits the number of modes observable given the excitation shape [3].

To circumvent these difficulties, we have used the methodology developed by Dollon et al., 2019 [4]. In this methodology, asynchronous transient experimental measurements are used to identify regions in the signal where the interaction between structural modes and RSI harmonics can be detected. From these regions, modal parameters can be estimated and compared to numerical predictions. Our case study is a recently commissioned Hydro-Québec medium head Francis runner, different from the unit initially used to develop the methodology. Our goal is to replicate the results of the previous study [4] using a different runner dataset and compare the modal characteristics thus obtained with those predicted by our numerical simulations.
In this paper, we apply a novel methodology to quantify modal frequency, damping and mode shape using commissioning field measurements. Then, the methodology and results obtained are discussed to contribute to the verification and validation of prototype Francis turbine runner blade numerical models.

2. Case study

The data used in this study comes from field measurements made on a medium head Francis turbine recently commissioned by Hydro-Québec, with a specific speed $n_\text{q} = 57$. The runner has 13 blades and its distributor has 20 guide vanes. From these characteristics, we can calculate the RSI excitation shapes and frequencies presented in Table 1 [5, 6]. The nodal diameters are obtained using the equation below:

$$ m \times Z_\text{g} + v = k \times Z_r $$

(1)

where $v$ is the nodal diameter, $Z_r$ is the number of runner blades, $k$ is the runner blade harmonic index, $Z_\text{g}$ is the number of guide vanes and $m$ is the corresponding harmonic number. Notice that positive and negative nodal diameters represent excitation in opposite rotating directions. This can also be observed in the calculated phase shift between two consecutive blades, calculated using the equation below:

$$ v \times 2\pi / Z_r $$

(2)

| RSI Harmonics | Excitation shape nodal diameters | Phase shift between consecutive blades [rad] |
|---------------|---------------------------------|---------------------------------------------|
| 1             | $2 \times 13 - 1 \times 20 = 6$ | 2.89                                        |
| 2             | $3 \times 13 - 2 \times 20 = -1$| -0.48                                       |
| 3             | $5 \times 13 - 3 \times 20 = 5$  | 2.41                                        |
| 4             | $6 \times 13 - 4 \times 20 = -2$ | -0.97                                       |
| 5             | $8 \times 13 - 5 \times 20 = 4$  | 1.93                                        |
| 6             | $9 \times 13 - 6 \times 20 = -3$ | -1.44                                       |
| 7             | $11 \times 13 - 7 \times 20 = 3$ | 1.44                                        |

To avoid amplification due to resonance, excitation shapes and frequencies should not match structural modes of the runner. Structural modes were predicted using the numerical model shown in Figure 1 and are presented in Table 2. Please note that including the runner shaft and guide bearing significantly affects the modal frequencies for modes ND1 and ND2. Furthermore, it should be noted that some modes tend to cluster around similar frequencies. For amplification or resonance to occur, excitation and structural modes shapes must be compatible, i.e. capable of interacting [3]. This constraint is both an advantage that may be exploited to avoid resonance during operation since only one structural mode shape is compatible with each RSI harmonic, and a hindrance to modal identification through OMA using background noise.
Figure 1. Overview of turbine runner numerical model

Table 2. Numerical modal analysis results for turbine only and for combined turbine, shaft and guide bearing

| Modes  | Turbine only | Turbine + Shaft + Guide bearing |
|--------|--------------|---------------------------------|
| ND2    | 28.2 Hz      | 26.9 Hz                         |
| ND1    | 28.7 Hz      | 13.7 Hz                         |
| ND6    | 30.4 Hz      | 30.3 Hz                         |
| ND5    | 31.0 Hz      | 31.0 Hz                         |
| ND4    | 31.9 Hz      | 31.9 Hz                         |
| ND3    | 32.3 Hz      | 32.3 Hz                         |
| ND1-2  | 48.3 Hz      | 44.0 Hz                         |
| ND2-2  | 52.9 Hz      | 52.0 Hz                         |
| ND3-2  | 61.4 Hz      | 61.4 Hz                         |

For our case study, the runner was instrumented using four strain gauge rosettes per blade, on two consecutive blades. An example of the signal obtained during startup is shown in Figure 2. In this figure, while the runner is at standstill between 0 and 115 s, we observe many amplification zones that seem to match the structural modes predicted by numerical simulation. However, as soon as the turbine begins to rotate, these zones disappear. From then until 390 s, we observe a significant stochastic response, which is typical of the part load region. Afterwards, once the part load vortex rope disappears, the strain response is dominated almost exclusively by RSI. Although such data, obtained during normal turbine operation, is useful for the characterization of the dynamic behavior of runners and for fatigue analysis, it is not well suited to OMA.
On the other hand, turbine run-up and coast-down conditions allow for the use of rotational excitations to perform a multi-sine sweep across a large frequency band. Such conditions are thus better suited to OMA in rotating machinery [7]. On a Francis turbine, it is in general difficult to achieve a very slow run-up during startup. Guided vanes and their control system are not designed for such a slow run-up. On the other hand, a relatively slow coast-down is obtained when the runner is stopped, either during a normal runner shutdown or load rejections all the way to a full stop. The turbine control system can also be configured to achieve low constant guide vane opening and closing rates so as to slow down the run-up and coast-down as shown in Figure 3. Even then, run-up remains slightly too fast, thus limiting the number of observable structural modes. Coast-down is much slower, which allows for the observation of many resonance regions associated with RSI harmonics, circled in red in Figure 3 (right).
Each RSI harmonic generates an excitation shape compatible with only a few structural modes, which solves the problem of mode coupling and allows for a better matching of the observed runner structural modes. A phase spectrogram [8] may be used to investigate the phase shift between two consecutive blades as shown in Figure 4. Using this figure, the frequencies and mode shapes of observed resonance regions can be matched by looking at the values obtained for each RSI harmonic. Using Figures 3 and 4, resonance regions can be located and expected mode shapes can be identified prior to modal identification using OMA.

Figure 3. Turbine runner run-up and coast-down. Time signal (left) and Short-Time Fourier Transform (STFT) with resonance regions circled in red (right)

Figure 4. Turbine runner run-up and coast-down phase spectrogram
3. Operational Modal Analysis (OMA)

Once potential resonance regions have been identified, OMA can be used to quantify natural frequencies, damping coefficients, mode shapes and modal forces. While not implemented in this study, OMA can be enhanced to also assess uncertainty [9], which is required for model validation [10, 11]. In this study, we have used OMA to estimate frequencies and damping coefficients. Mode shapes have been identified with a high level of certainty using phase differences between two consecutive blades as presented in the previous section. The methodology used was Order Based Modal Analysis (OBMA), developed by Dollon et al., 2019 [4].

The first step of OBMA is Order Tracking (OT). This allows for the use of a classical ambient modal identification algorithm to estimate modal information. OT converts the signal from the time domain to the angular domain. The Fourier transform of this signal results in orders rather than frequencies. Each order is a fraction of the angular velocity, which in this case corresponds to the rotating frequency. Knowing the rotation speed at every time step, it is possible to generate a map of orders versus rotation speed (RPM) as shown in Figure 5 (left). From this map, the spectrum for a given order can be extracted, as shown in Figure 5 (right) for the first RSI harmonic (order 20), extracted between 150 and 300 s of the study dataset.

![Figure 5. First RSI harmonic (order 20) between 150 and 300 s. Order-RPM map (left) and Order Tracked Amplitude spectrum (right)](image_url)

The methodology used to obtain Single Degree of Freedom (SDoF) responses is similar to Frequency Domain Decomposition (FDD) [12]. In this technique, a bandwidth is selected using the Modal Assurance Criterion (MAC) [13], which allows for the separation of close uncoupled modes and noise. Then, for modal identification, a SDoF transfer function is obtained using a maximum likelihood estimator. In some cases, the Signal to Noise Ratio (SNR) is too low and the bandwidth is selected by performing a sensitivity analysis on the estimated parameters. For more details on the OBMA methodology used, please see [4].

4. Results

The results obtained using OBMA are presented in Table 3. A wide range of modes are identified. The most critical mode shape is ND6 since it corresponds to the excitation shape of the first RSI harmonic. ND6 is also the only mode that is detectable during both run-up and coast-down. In both cases, the values obtained are similar. Furthermore, for higher RSI harmonics, the SNR is low and a sensitivity analysis has been used to define the bandwidth for identification purposes.
Table 3. OBMA results

| RSI | Harmonics | Order | Time interval | Nodal diameter | Blade phase shift [rad] | Frequency [Hz] | Damping [%] | Bandwidth selection* |
|-----|-----------|-------|---------------|----------------|------------------------|----------------|-------------|----------------------|
| 1   | 20        | 150-350 | ND6           | 2.89           | 30.8                   | 5.92           | MAC        |
| 1   | 20        | 600-900 | ND6           | 2.89           | 29.7                   | 4.69           | MAC        |
| 2   | 40        | 1150-1350 | ND1         | -0.48          | 11.3                   | 8.02           | MAC        |
| 3   | 60        | 850-1150 | ND5           | 2.41           | 31.7                   | 0.56           | MAC        |
| 4   | 80        | 900-1050 | ND2           | -0.97          | 44.4                   | 1.14           | Sens       |
| 4   | 80        | 1150-1350 | ND2         | -0.97          | 23.7                   | 1.45           | MAC        |
| 5   | 100       | 1050-1375 | ND4          | 1.93           | 32.1                   | 0.00           | Sens       |
| 6   | 120       | 975-1125 | ND3           | -1.44          | 57.6                   | 0.43           | Sens       |
| 6   | 120       | 1050-1375 | ND3          | -1.44          | 31.8                   | 0.23           | Sens       |
| 7   | 140       | 1025-1125 | ND3          | 1.44           | 57.1                   | 0.43           | Sens       |

*Sens = Sensitivity analysis, MAC = Modal Assurance Criterion

5. Discussion

The comparison of OBMA and numerical modal analysis results presented in Table 4 shows that ND1 and ND2 are significantly overestimated. The overestimation is lower when the shaft and guide bearing are included, but the relative error for these modes remains significantly higher than for the other modes. The experimental values obtained through OBMA are valuable for numerical simulation verification and calibration. However, without uncertainty values for both OBMA and numerical modelling results, it remains difficult to judge whether a model is valid.

Table 4. Numerical modal analysis frequency relative error

| RSI Harmonics | Order | Time interval | Nodal diameter | OBMA [Hz] | Turbine only | Turbine + Shaft + Guide bearing |
|---------------|-------|---------------|----------------|-----------|--------------|-------------------------------|
| 1             | 20    | 150-350       | ND6            | 30.8      | -1.3%        | -1.6%                         |
| 1             | 20    | 600-900       | ND6            | 29.7      | 2.4%         | 2.0%                          |
| 2             | 40    | 1150-1350     | ND1            | 11.3      | 154.0%       | 21.2%                         |
| 3             | 60    | 850-1150      | ND5            | 31.7      | -2.2%        | -2.2%                         |
| 4             | 80    | 900-1050      | ND2            | 44.4      | 19.1%        | 17.1%                         |
| 4             | 80    | 1150-1350     | ND2            | 23.7      | 19.0%        | 13.5%                         |
| 5             | 100   | 1050-1375     | ND4            | 32.1      | -0.6%        | -0.6%                         |
| 6             | 120   | 975-1125      | ND3            | 57.6      | 6.6%         | 6.6%                          |
| 6             | 120   | 1050-1375     | ND3            | 31.8      | 1.6%         | 1.6%                          |
| 7             | 140   | 1025-1125     | ND3            | 57.1      | 7.5%         | 7.5%                          |

Note that in addition to frequencies, OBMA allows for the estimation of damping coefficients for each mode. However, the operating conditions under which damping is estimated are quite different from normal operation. Most of the modes other than ND6 are identified while the runner is rotating with its guide vanes fully closed, which can have a significant effect on hydrodynamic damping [14].

6. Conclusions

In this study, it has been shown that Francis turbine runner modes can be identified using data obtained under run-up and coast-down transient conditions. Our case study demonstrates that turbine control systems can be used to achieve slower than normal run-up and coast-down speeds, thus facilitating modal identification. By comparing OBMA and numerical simulation results, we have shown that numerical models can be verified and calibrated using observations made during commissioning. However, although damping can be estimated using experimental measurements, the operating conditions under which this data is obtained hinder any comparison with expected normal operating
values. Work remains to be done to assess uncertainty and understand the physics behind the values obtained.

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