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Experimental Modal Analysis of a Turbogenerator Rotor

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Abstract

This research presents a methodology for implementing a modal testing procedure on a turbo-generator rotor to identify its natural frequencies and corresponding mode shapes. Experimental modal analysis is performed using an impact hammer and specialized signal processing software under two distinct rotor configurations. The measured modal parameters are then compared to assess the influence of rotor stiffness in different directions. Subsequently, a numerical model of the rotor is developed using Madyn 2000 software to determine the natural frequencies and mode shapes under free boundary conditions. A comparison between the experimental modal test outcomes (natural frequencies and mode shapes) and the numerical analysis results from Madyn is presented. The natural frequencies obtained from the experimental tests are approximately the same; however, the frequencies obtained from the simulation are up to 5.6% higher than the experimental results. The mode shapes extracted from both the experimental tests and simulations show reasonably good agreement.

Keywords: experimental modal test; bump test; high power generator; rotor dynamics.

1. Introduction

Modal analysis involves identifying the inherent dynamic properties of a structure, namely its natural frequencies, damping ratios, and mode shapes, and utilizing these parameters to construct a mathematical representation of its dynamic behavior. This representation is referred to as the system's modal model, and the corresponding parameters are collectively known as its modal data. Many researches have investigated modal analysis on rotating equipment, especially turbogenerators. A study on a steam turbine generator power plant investigated the rotor's torsional dynamics using both experimental and analytical modal analysis methods [1]. The measured natural frequencies and mode shapes were compared with theoretical predictions to assess the model's accuracy. The research also explored relationships among torsional stresses, angular twist spectra, and generator stator current. The findings provided a deeper understanding of the rotor's dynamic behavior under operational and synchronization conditions. In [2], a novel electromagnetic torsional exciter developed for conducting dynamic vibration tests on large rotating machinery, such as flywheel generators is proposed. The device applies torque through the stator windings of an existing three-phase machine, eliminating the need for physical attachment to the shaft. It's a self-powered, current-controlled circuit that stores and releases magnetic energy to generate adjustable pulsating torques without requiring an external power source. Designed as a compact and mobile system, it enables on-site modal and acceptance testing of large rotors by producing controlled torsional loads at variable frequencies and amplitudes suitable for multi-megawatt applications. In [3], excessive end-winding vibrations in a turbogenerator were investigated using Experimental Modal Analysis (EMA). Increasing vibration amplitudes were observed in the end-winding region, particularly near the exciter end, during frequency sweep tests conducted under sustained operating conditions. Through frequency response evaluation, a global resonance of the stator overhangs was identified, with a 4-node/2-lobe mode shape found to amplify the vibration behavior. Subsequently, corrective modifications were implemented, shifting the resonance frequency to a higher range and substantially reducing vibration levels. Although the complex interaction between excitation forces and structural response could not be fully characterized, the study demonstrated that EMA can be effectively applied to diagnose and mitigate vibration problems in large turbogenerators.

A systematic approach for identifying torsional modal frequencies and mechanical damping in large turbine generators has been proposed in [4]. The method, known as the supplementary-excitation-signal-injection technique, generates controlled torsional vibrations without interrupting generator operation. Accurate identification of modal parameters is achieved through the combined application of modal filtration, an enhanced discrete Fourier transform (DFT), and least-squares fitting, with electrical damping separated to isolate the pure mechanical component. The proposed technique has been validated through simulations and field applications at the Shangdu Power Plant, demonstrating high accuracy, operational safety, and cost-effectiveness while enabling excitation of all significant torsional modes under various operating conditions. In [5], the dynamic behavior of a micro gas turbine plant rotor operating at 65,000 rpm is analyzed to ensure the absence of critical speeds within ±30% of the operating speed. The rotor, consisting of a turbocharger rotor and a starter-generator rotor connected by an elastic coupling, was examined using finite element analysis and experimental modal testing. The effect of bearing stiffness on natural frequencies was investigated to establish acceptable stiffness limits. Based on analytical, numerical, and experimental results, design recommendations were developed, including maintaining bearing stiffness below 10^6 N/m and employing a flexible coupling to keep critical speeds outside the operating range.

Study in [6], presents experimental and numerical modal analyses of a Kaplan turbine model under dry and wet conditions. Using impact hammer and shaker excitation, the main vibration modes of the rotor and runner blades were identified and compared with numerical predictions. Four main vibration modes below 100 Hz were detected, with deviations of about 10% between experimental and numerical results. The presence of water has a greater impact on damping ratios than on natural frequencies, with mode M4 exhibiting the largest added mass effect. Paper in [7], develops and validates a linear dynamic model of a steam turbogenerator exciter rotor–armature system for fault diagnosis and nonlinear interaction analysis. Using Timoshenko beam theory, an analytical model incorporating the linear effects of rotor-armature contact interfaces was created and validated against experimental modal test results under free-free boundary conditions. The comparison confirmed the model's accuracy and enabled the identification of unknown structural parameters.

In [8], two case studies demonstrating the practical benefits of rotor dynamic models in power plant applications using Madyn 2000 software is presented. In the first case, the model enabled precise and efficient balancing corrections for a generator shaft; in the second, it helped identify misalignment-related vibration issues and critical speeds. These models serve as valuable tools for diagnosing complex vibration problems, providing insights into system behavior and parameter sensitivity even when exact predictions are not possible. The study highlights their strategic importance for key shaft lines in maintenance and troubleshooting. Study in [9], examines four rotor systems that necessitate nonlinear dynamic analysis due to bearing-related nonlinearities: a vertical pump, a turbocharger, an electric motor, and a Pelton turbine. In each case, linear modeling proved inadequate, instability arising from unloaded bearings in the pump, nonlinear oil film effects in the turbocharger, high dynamic loads in the motor's rolling element bearings, and extreme bearing forces following bucket loss in the Pelton turbine. Consequently, nonlinear analysis was essential to accurately predict limit cycles, dynamic loads, and system responses under these complex operating conditions. Research in [10], compares isotropic and orthotropic modeling approaches for stacked rotor laminations to evaluate their impact on rotor natural frequencies. Experimental modal testing demonstrates that the orthotropic model aligns significantly better with measured frequencies than the isotropic model. After identifying bearing stiffness and conducting comprehensive finite element (FE) modal and unbalance analyses of the motor, the results confirm that the motor operates safely within vibration limits and that bearing vibration amplitudes remain below IEC thresholds.

2. Turbogenerator structure

A turbogenerator combines a turbine and an electrical generator to convert thermal energy into electricity. At its core, the rotor transmits mechanical power from the turbine while generating the magnetic field necessary for electricity production. Rotating at high speeds, typically between 1500 and 3600 rpm (depending on the prime mover, number of poles, and required frequency), the rotor endures intense centrifugal, thermal, and magnetic stresses, necessitating an extremely precise design and balance. Therefore, understanding and controlling the rotor's dynamic behavior, including vibrations and critical speeds, is essential for the reliable and efficient operation of the entire system. In this research, a high-power generator rotor with a nominal speed of 3000 rpm for modal analysis is considered.

3. Experimental modal analysis

Modal analysis is based on the principle that the vibration response of a linear, time-invariant dynamic system can be expressed as a linear combination of simple harmonic motions, known as the system's natural vibration modes. Each mode is characterized by specific modal parameters, including its natural frequency, damping ratio, and mode shape (which may be real or complex). Each natural frequency corresponds to a distinct mode, and the contribution of each mode to the overall vibration depends on both the excitation characteristics and the system's mode shapes [11]. EMA typically involves three major stages: establishing appropriate boundary conditions, conducting the test under laboratory conditions, and analyzing the collected data using post-processing software [12].

3.1 Boundary condition

Generally, three types of boundary conditions are used to restrict the motion of a system: fixed or clamped support, simply supported, and free boundary conditions. The free boundary condition is the most convenient and commonly used in experimental tests. Therefore, in this test, this boundary condition is achieved by suspending the rotor with two webbing slings, as shown in Figure 1.

Figure 1. Free boundary condition fulfilment

3.2 Model test execution

The modal test of the rotor was conducted using the impact hammer method (bump test). The rotor was suspended by two webbing slings near the bearing locations that its axis remained parallel to the ground (Figure 1). As illustrated in Figure 2, ten defined points on the rotor were marked for data acquisition.

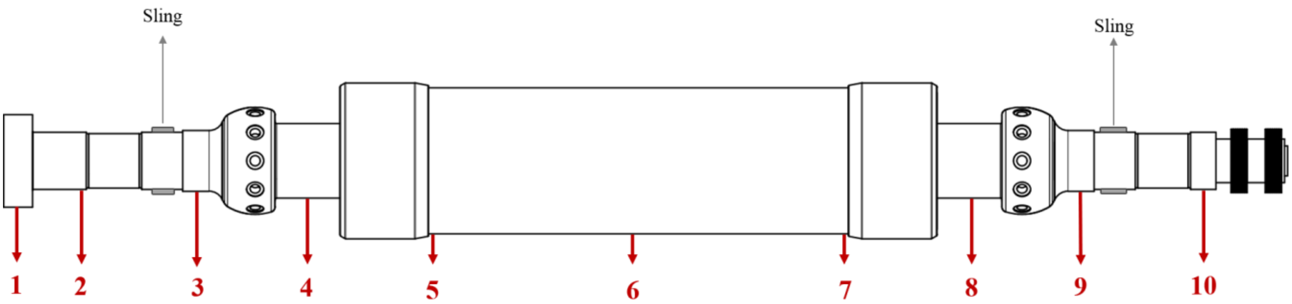


Figure 2. Defined points for data acquisition and impact hammer test

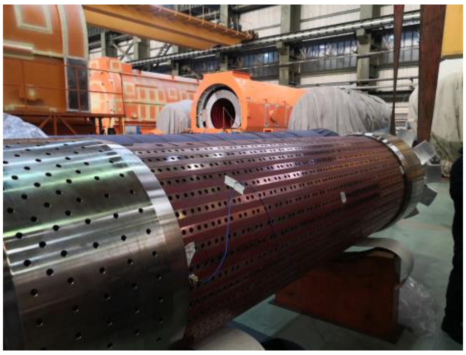
Since the rotor cross-section in barrel part is not symmetric due to the presence of poles and rotor slots, the test was performed in two separate stages, as described below.

3.2.1 First bump test

The rotor was suspended in a way that the line passing through the poles aligned vertically. The accelerometer sensor was attached to the rotor horizontally at 10 specified points. The shaft was then struck horizontally with a hammer at points 1 and 6. The reason for selecting the horizontal direction for both the hammer impact and data acquisition is that the slings effectively restrain the rotor in the vertical direction, providing considerable stiffness along that axis, with the rotor suspended in this configuration. However, in the horizontal direction, it can be reasonably assumed that there is no significant constraint on the shaft; consequently, the displacement amplitude resulting from the hammer impact in this direction will be relatively large.

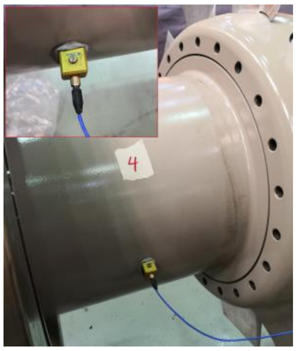
3.2.2 Second bump test

The conditions of this test were similar to those of the first, except that the rotor has been rotated 90 degrees around its axis, positioning the rotor poles on the sides of the shaft (the line passing through the poles oriented horizontally). Same as the first test, the sensor recorded data horizontally at points 1 to 10, and the hammer impact was applied at points 1 and 6, horizontally (Figure 3).



(a)

(b)



(c)

Figure 3. Hammer test implementation, (a) First test, (b) Second test, (c) Sensor placement on the rotor

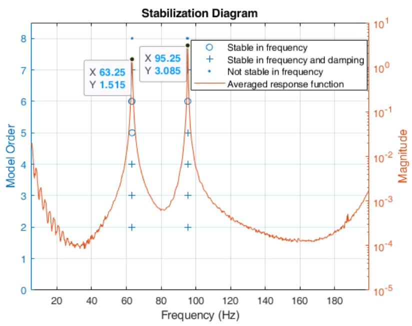
The data acquisition was performed using the “ZonicBook/618E” device and the data processing was carried out with “Ez-Analyst” software. The initial parameters for this processing are presented in Table 1.

Table 1. Data acquisition initial parameters

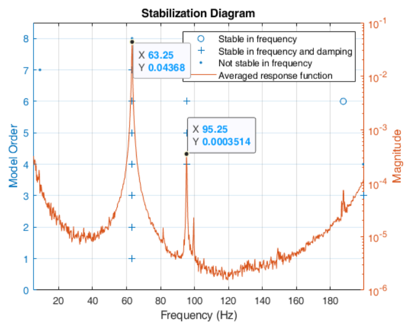
Parameter	Value	Unit
Analysis frequency (f)	200	Hz
Spectral lines (S)	800	-
Nyquist factor (NF)	10.24	-
Frame width (w=S/f)	4	second

3.3 Modal test result

The postprocessing of the obtained data were executed using “MATLAB” software. The stabilization diagrams, which consider Frequency Response Function (FRF) and model order for each test, are shown in Figure 4.



(a)



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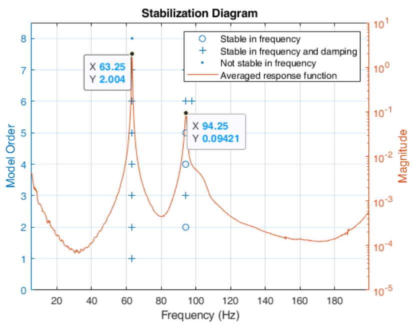

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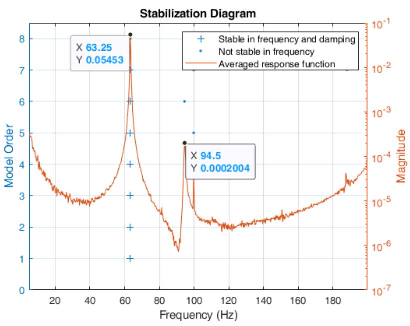
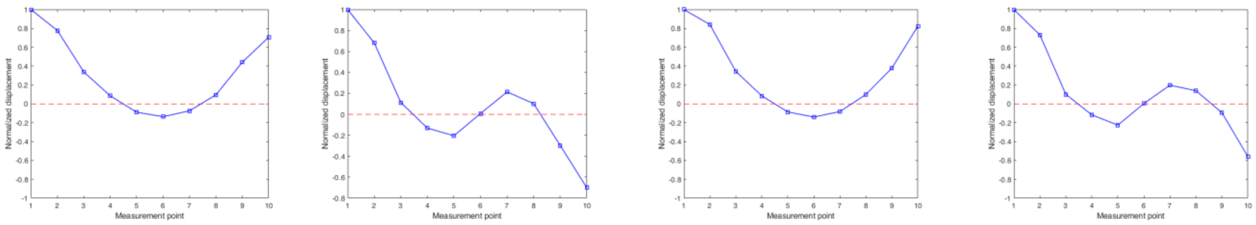


(d)

Figure 4. Stabilization diagram, (a) First test; hammer at 1, (b) First test; hammer at 6, (c) Second test; hammer at 1, (d) Second test; hammer at 6

These diagrams distinguish the real (physical) modes of a structure from false modes caused by computational errors or noise. The procedure involves modeling the system as an nth-order degree of freedom system (n = 1, 2, …), with the corresponding frequencies and mode shapes estimated from test data. Each estimation gives a set of modal frequencies. When these frequencies are plotted against the model order (blue graph), modes that appear at the same frequency across multiple orders are identified as actual (stable), while false modes typically appear randomly and become unstable or disappear at higher orders. Additionally, the red graph represents the averaged FRF magnitude collected from all sensors at a specific hammering point. Based on the stabilization diagram analysis, the frequencies of 63.25 Hz and 95.25 (94.5) Hz are identified as the system's actual natural frequencies.

The normalized mode shape configurations for each test are illustrated in Figure 5. As expected, the first mode shape has two nodes along the shaft (V mode shape) and the second mode shape has three nodes along the shaft (S mode shape).



(a) (b) (c) (d)

Figure 5. Experimental mode shapes, (a) First test; first frequency (63.25 Hz), (b) First test; second frequency (95.25 Hz), (c) Second test; first frequency (63.25 Hz), (d) Second test; second frequency (94.5 Hz)

4. Simulation

To validate the experimental modal test results, a corresponding rotor dynamic model was developed in "Madyn 2000" and configured to reproduce the boundary and mass stiffness conditions of the workshop test.

4.1 Discrete mass model

In "Madyn 2000" software the rotor was modeled under free-free conditions, consistent with the experimental setup in which the shaft was suspended using flexible slings. All bearing stiffnesses and support constraints were removed, and damping was minimized to obtain undamped natural modes (Figure 6).

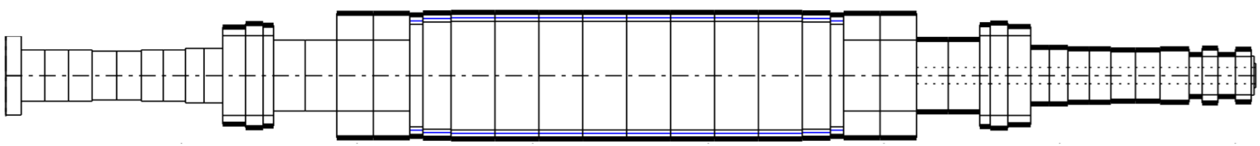
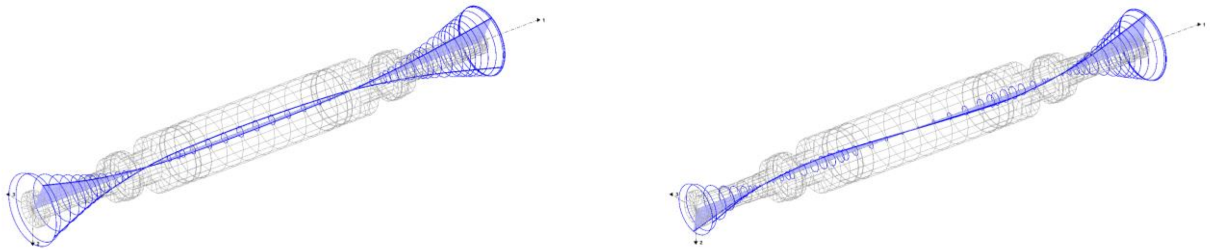


Figure 6. Rotor discrete model in Madyn 2000

4.2 Simulation results

The simulation results indicate that the first and second bending frequencies are 64.5 Hz and 100.6 Hz, respectively. The mode shapes corresponding to these two frequencies are shown in Figure 7. In these figures, Directions 1, 2, and 3 correspond to the rotor shaft axis, the vertical direction (gravity), and the horizontal direction, respectively. Since the rotor model in the software was implemented under the assumption of axial symmetry, the frequencies and mode shapes in the vertical and horizontal directions are identical. The 64.5 Hz frequency corresponds to a two-node (V) mode shape, while the 100.6 Hz frequency corresponds to a three-node (S) mode shape.



(a) (b)

Figure 7. First two bending mode shapes extracted from simulation, (a) First frequency (64.5 Hz), (b) Second frequency (100.6 Hz)

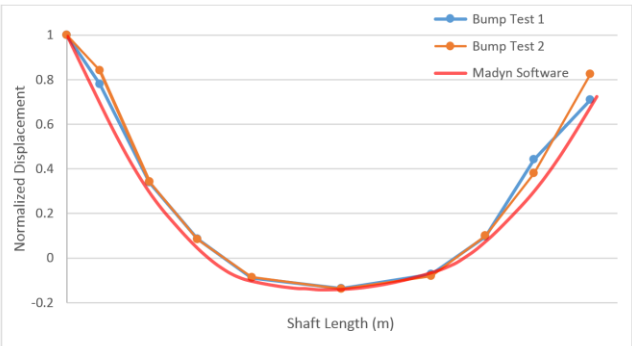
5. Results comparison

The results of the modal tests conducted in the workshop, along with the data obtained from the Madyn software, are summarized in Table 2. The error criterion in this table is based on the values from the First bump test. Since the rotor shaft was modeled as axisymmetric, the vertical and horizontal frequencies obtained from the Madyn analysis are identical. The comparison between the two experimental tests shows that there is approximately no error between the results. Any minor discrepancies may be caused by measurement variations during the testing procedure. In addition, the simulation error of the software relative to the first test is 2% for the first frequency and 5.6% for the second frequency. This difference may be attributed to modeling and simulation inaccuracies in the software.

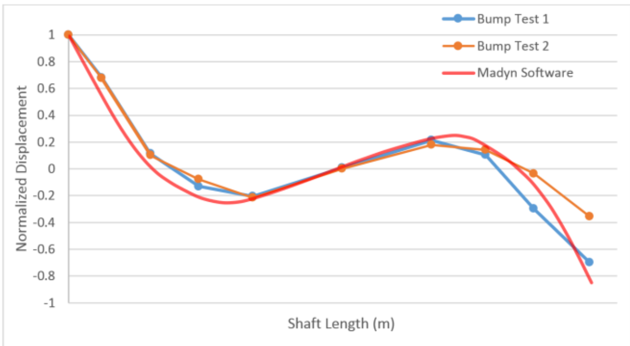
Table 2. Model test and simulation compliance

Result	First Frequency [Hz]	Error [%]	Second Frequency [Hz]	Error [%]
First bump test	63.25	-	95.25	-
Second bump test	63.25	0	94.5	-0.8
Madyn 2000	64.5	2	100.6	5.6

Finally, a comparison of the first and second mode shapes obtained from the modal test and the Madyn software analysis is presented in Figure 8. As shown, the mode shapes produced by the two different approaches (experimental and computational) exhibit nearly identical behavior. Minor discrepancies between the two sets of results may arise from factors such as measurement errors during testing and modeling inaccuracies in the software.



(a)



(b)

Figure 8. Comparison of the mode shapes obtained from EMA and simulation, (a) First frequency, (b) Second frequency

Conclusion

This research described the procedure for conducting a generator rotor modal test to identify natural frequencies and corresponding mode shapes. The experimental modal test was performed using an impact hammer in two different rotor orientations. Additionally, the natural frequencies and mode shapes for the free boundary condition, obtained from Madyn software, were presented. The comparison between the two experimental tests shows that the first natural frequency remained unchanged, while the second natural frequency in the second test was 0.8% lower than in the first test. The comparison between the frequencies obtained from experimental and simulation results reveals an error of about 5%. This difference may be attributed to modeling and simulation inaccuracies in the software. The mode shapes derived from both the experiments and simulations align satisfactorily.

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