



Research Article

Theoretical and Practical Analysis of the Balance Effect on the Hydraulic Turbine-Generator Unit

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ABSTRACT

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The balance condition has a significant influence on the stability, dynamic behavior, and vibration characteristics of hydraulic turbine-generator units operating at high rotational speeds and under large mechanical loads. This study investigates the impact of balance effects on the operational stability of a 150 MW Francis turbine. First, a new nonlinear multi-degree-of-freedom mathematical model incorporating balance effects is developed. The stiffness and damping coefficients required for the model were identified under various operating conditions by measuring both absolute and relative vibrations using sensors installed on the turbine-generator unit. Subsequently, vibration data obtained from the turbine were compared with model predictions before, during, and after balance failure. The results demonstrate that the balance condition plays a critical role in maintaining safe vibration levels and stable operation. Furthermore, the proposed mathematical model successfully captures the dynamic behavior of the turbine under different operating scenarios. This study provides both theoretical and practical insights that can support improved design, installation, and operational strategies for hydraulic turbines.

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Introduction

The accelerating global transition toward low-carbon energy systems has placed renewable power generation at the forefront of sustainable development strategies. As countries aim to reduce greenhouse gas emissions, enhance energy security, and increase the resilience of their power infrastructure, renewable-based electricity production has become a cornerstone of modern energy policy. Beyond their clear environmental advantages, renewable energy technologies offer operational flexibility and long-term economic benefits, making them indispensable elements of future power grids [1]. Among these, hydroelectric power plants stand out as one of the most advantageous options since they utilize a renewable resource, offer relatively low operation and maintenance costs, and provide ancillary services enabled by large water reserves [2]. Additionally, they play a key role in meeting fluctuating energy demand and maintaining grid frequency stability due to their ability to rapidly increase output and perform fast start-stop operations [3]–[5]. To ensure that hydroelectric power plants can effectively respond to the power demands of interconnected systems, it is essential to minimize downtime arising from unexpected faults [6].

In recent years, fault detection methods based on sound and vibration signals have gained increasing attention due to their high sensitivity, capability for continuous dynamic monitoring, and adaptability to complex environments [7]. Various online vibration monitoring systems are deployed across different components of hydraulic turbine-generator equipment, including relative shaft vibration, absolute bearing vibration, axial bearing vibration, stator vibrations, stator rod vibrations, and stator end-winding vibrations [8]. Vibration-based signal processing techniques are widely used to analyze the operational behavior of rotating machines, and these data are further evaluated through vibration condition monitoring devices, enabling stakeholders to access diagnostic information remotely at any time. This allows service providers to implement condition-based maintenance strategies and offer operational support. Furthermore, increased collaboration among stakeholders through shared monitoring platforms enhances maintenance planning and ultimately increases system availability [9]–[11].

Vibrations generated in the shaft systems of hydropower plants primarily arise from coupled hydraulic–mechanical–electrical interactions, and these vibrations may exhibit multi-frequency, regular, semi-regular, or irregular characteristics [12]–[14]. Extensive research on the coupled

structural vibration of turbine-generator units has yielded significant insights into the causes and implications of these dynamic behaviors. Improper bearing adjustments in the turbine-generator system and off-design operating conditions—including bearing failures, shaft cracks, and shaft torsion—can lead to elevated vibration levels, resulting in long-term equipment damage and high repair costs [15]–[18]. Designing mechanical components to withstand certain off-design conditions is therefore a critical aspect of hydroelectric turbine engineering [19]. In this context, several studies have also examined how off-design operating conditions influence stress and damage behavior in turbine components. Unterluggauer et al. [20] investigated fatigue damage in Francis turbine runners subjected to vibration and pressure pulsations under transient operating conditions. Niu et al. [21] examined the effects of rotor mass eccentricity, revealing that variations in eccentricity distance and guide-bearing stiffness significantly influence vibration behavior. Ri et al. [22] stated that unbalanced mass forces, along with the effect of the bearing-related oil film, significantly affect the dynamic response and critical rotational speed of the hydro-generator shaft system. Similarly, Wang et al. [23] performed vibration analysis of the shaft using a finite element model and recommended balancing the mass distribution and increasing shaft stiffness to prevent resonance at rotational frequencies. Rodal [24] explored how vibration signals can be employed to detect short-circuit faults and static eccentricity in synchronous generators with different pole configurations. Zhang et al. [25] demonstrated that dynamic rotor eccentricity increases both the DC and fundamental harmonic components of unbalanced magnetic pull. To evaluate the impact of subsynchronous resonance on turbine-generator shafts, Manikandan and Khan [26] developed a nonlinear damping model to represent torsional interaction and suppress mechanical vibrations caused by this phenomenon. The findings of Shi et al. [27] further show that connected vibration sources can effectively characterize shaft vibration behavior and reveal the influence of various external excitations on fault characteristics.

Rotating shafts constitute essential structural components of engines, wind turbines, and hydraulic turbines. In such systems, oscillations are typically induced by periodic variations in gyroscopic inertia forces over time [28]. Effective vibration control of rotors is therefore critical, especially when the rotor is coupled with other components through mechanical linkages [29]. Accordingly, numerous researchers have developed mathematical models to describe the behavior of rotating elastic systems consisting of shafts and rotors, often using complex differential equations and incorporating gyroscopic forces into the analysis [30]–[32]. Li et al. [33] investigated the vibration characteristics of a hydraulic turbine-generator unit without considering the gyroscopic effect, later demonstrating that gyroscopic forces significantly amplify shaft vibration and thereby influence the safe and stable operation of hydropower plants. Rondon et al. [34] emphasized that bearing stiffness and damping depend not only on load magnitude but also on rotor position within the bearing. This is notable because

unbalance forces and gyroscopic effects are closely related to both rotational speed and acceleration [35].

Paturu et al. [36] conducted free vibration analysis on the shaft of a 70 MW hydroelectric turbine using models developed with beam, mass, and combined elements in ANSYS, examining multiple modes up to the critical operating speed of 3000 rpm. Chaubey et al. [37] monitored shaft operating conditions using neural networks and found that a multilayer perceptron trained by backpropagation outperformed conventional statistical methods.

This study focuses on a turbine-bearing system operating under the hydrodynamic lubrication principle. Its primary objective is to determine the complete unbalance condition of turbine-generator equipment by comparing the mathematical model of the turbine-bearing system with vibration data obtained via fast Fourier transform under balanced and unbalanced operating conditions. By identifying the vibration characteristics of the shaft and turbine-bearing components—which constitute the core elements of a hydraulic turbine unit—this research aims to provide solutions for the operational challenges encountered under optimum conditions in hydroelectric power plants. Additionally, by calculating vibration amplitudes in the shaft and bearing system, the resonance behavior of the complete turbine-generator assembly is determined under vibrational influence. Understanding this behavior supports maintaining the structural integrity of the turbine-generator system within optimal electricity generation limits.

Beyond these contributions, this study introduces a novel and simplified mathematical-model-based approach capable of detecting vibration anomalies in real time, enabling early fault prediction and supporting predictive maintenance strategies in hydroelectric power plants. By integrating model-derived dynamic responses with online vibration monitoring data, the proposed method enhances operational reliability and offers a practical framework for reducing unexpected downtime, improving maintenance planning, and ensuring long-term system health.

Material and Method

This section presents the materials, experimental setup, and analytical methods employed to investigate the dynamic behavior and balance-related vibration characteristics of the hydraulic turbine-generator unit. The procedures used for data acquisition, mathematical modeling, signal processing, and model validation are detailed to ensure reproducibility and to provide a comprehensive foundation for the analysis conducted in this study.

In order to accurately represent the influence of rotor unbalance on the dynamic response of the turbine-bearing system, the mathematical model incorporates the hydrodynamic behavior of the lubricated bearings, making the characterization of pressure and force distribution essential for understanding how imbalance-induced forces propagate through the shaft-bearing interface.

The following equations are used to calculate the pressure and force distribution in the hydrodynamically lubricated bearings [38]. The force on the bearing is calculated according to the semi-Sommerfeld condition, as in Equations 1 and 2. The solution of the integrals in Equations 1 and 2 can be seen in Equations 3 and 4:

$$W_1 = \int_0^{\pi} \int_{-L/2}^{L/2} p.R_1.\cos\Theta d\Theta dy \quad (1)$$

$$W_2 = \int_0^{\pi} \int_{-L/2}^{L/2} p.R_1.\sin\Theta d\Theta dy \quad (2)$$

$$W_1 = -\frac{U.\eta.\varepsilon^2.L^3}{c^2.(1-\varepsilon^2)^2} \quad (3)$$

$$W_2 = \frac{U.\eta.\varepsilon.\pi.L^3}{4.c^2.(1-\varepsilon^2)^{3/2}} \quad (4)$$

The physical model of the rotor-bearing system, established to investigate the dynamic behavior of the vertical hydraulic turbine, along with the bearing interaction mechanisms, is schematically illustrated in Figure 1.

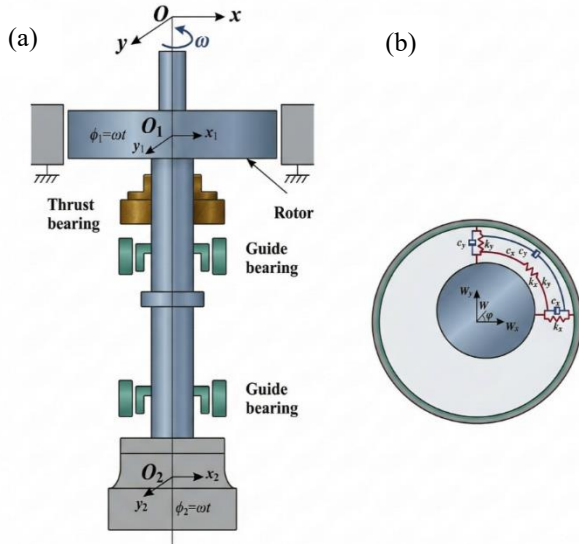


Figure 1. Schematic representation of the shaft and bearing model in vertical hydraulic turbines (a), Dynamic cross-sectional model showing the equivalent stiffness and damping characteristics of the guide bearing (b).

The mathematical model created according to Figure 1 is shown in Equations 5 and 6.

$$mx'' + c_x x' + k_x x = W_1 + m_e \omega^2 \cos \varphi \quad (5)$$

$$my'' + c_y y' + k_y y = W_2 + m_e \omega^2 \sin \varphi \quad (6)$$

Since the differential equations in both axes have the same structure, the solution procedure is exemplified using the y -axis. When Equation 6 is rearranged by moving the system's dynamic properties (damping and stiffness) to the left side, Equation 7 is formed:

$$my'' + cy' + k_y y = W_2 + m_e \omega^2 \sin \varphi \quad (7)$$

If the right-hand side of Equation 7 (representing the hydrodynamic and unbalance excitation forces) is denoted by the letter "b", Equation 8 is formed:

$$b = W_2 + m_e \omega^2 \sin \varphi \quad (8)$$

By substituting Equation 8 into Equation 7, a 2nd-order non-homogeneous differential equation with constant coefficients is obtained as Equation 9:

$$my'' + cy' + k_y y = b \quad (9)$$

The characteristic equation of the second-order system is given in Equation (10). The roots of the characteristic equation are presented in Equation (11). By substituting these roots into the general solution expression and applying the standard variation of parameters procedure, the analytical solution is obtained as shown in Equation (12). The intermediate derivation steps are omitted for brevity, since they follow a well-established textbook methodology.

$$mr^2 + c_y r + k_y = 0 \quad (10)$$

$$r_{1,2} = \frac{-c_y \pm \sqrt{c_y^2 - 4mk_y}}{2m} > 0 \quad (11)$$

$$y = s_1 e^{r_1 t} + s_2 e^{r_2 t} + \frac{bm}{k} \quad (12)$$

W_1 and W_2 forces acting on the bearing in the x and y directions create absolute and relative resonances in x and y directions. Using the resonances formed according to these forces, the stiffness and damping coefficients can be calculated in x and y directions in Equation 13.

$$k_x = \frac{W_1}{x} \text{ (N/m)}, k_y = \frac{W_2}{y} \text{ (N/m)}, c_x = \frac{W_1}{x'} \text{ (Ns/m)}, c_y = \frac{W_2}{y'} \text{ (Ns/m)} \quad (13)$$

Results and Discussion

In this section, the resonance values measured by the sensors and those obtained from the mathematical model are comparatively evaluated for both the balanced and unbalanced operating conditions of the 150 MW Francis hydraulic turbine. The turbine, equipped with a synchronous generator operating at 166.67 rpm and supplied by a net head of 145 m, exhibited significant unbalance during operation. After applying balancing weights and restoring the bearing to its original clearance, the malfunction was eliminated. The following results

analyze how these corrective actions influenced the resonance behavior and overall dynamic response of the turbine-generator unit, highlighting the consistency between the mathematical model and the measured vibration data.

The k_x , k_y , c_x , and c_y coefficients—formed according to the resonance data obtained from the sensors for a power loading period of 0 MW to 150 MW when a hydraulic turbine operates at design values in the unbalanced condition and after the unbalance is removed—are shown in Fig. 2, 3, 4, and 5.

The stiffness (k_x) and damping (c_x) coefficients of the turbine-bearing system exhibit pronounced variations under unbalanced operating conditions as the generator load increases from 0 MW to 150 MW. Sharp increases in k_x , followed by sudden drops, indicate that the unbalanced rotor motion pushes the hydrodynamic bearings toward their operational limits, especially in the 35–90 MW range. In this interval, the combined effects of cavitation inside the turbine and intensified radial hydraulic forces generate significant stiffness peaks, while the corresponding c_x response demonstrates the reduced damping capability of the bearing under excessive dynamic loading. These trends highlight how unbalance distorts the load-carrying mechanism of the bearings and amplifies the transmitted vibration forces. The variation of k_x and c_x under these conditions is illustrated in Figure 2.

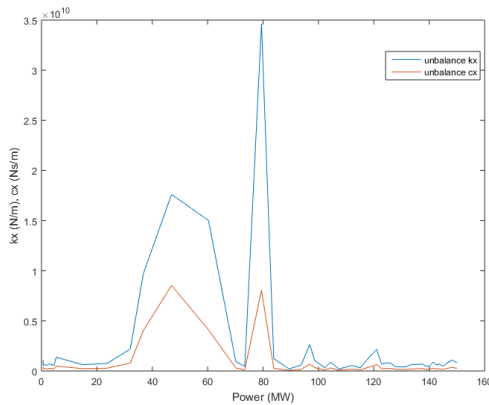


Figure 2. Values of k_x and c_x according to the power change in the hydraulic generator in fault condition (hydraulic unbalance state).

The stiffness (k_y) and damping (c_y) coefficients in the y-direction exhibit markedly different behavior from those observed in the x-direction, revealing the directional sensitivity of the shaft-bearing system under unbalanced operation. As the turbine-generator load increases from 0 MW to 150 MW, the y-direction coefficients show pronounced fluctuations, including several sharp reductions and reversals of stiffness, indicating that the hydrodynamic film becomes unstable under the dynamic radial forces acting asymmetrically on the shaft. These variations reflect the redistribution of unbalanced

hydraulic forces around the bearing clearance, which amplifies the structural response in one direction while reducing it in the orthogonal direction. The contrasting behavior between Figures 2 and 3 demonstrates that the unbalanced radial load does not increase uniformly in all directions, but instead produces maximum stiffness and damping in the x-direction while yielding minimum values in the y-direction at equivalent power levels. This directional imbalance plays a critical role in defining the vibration orbit and the overall stability of the rotor-bearing system. The corresponding variation of k_y and c_y is presented in Figure 3.

The negative k_y and c_y values observed at high unbalance levels indicate oil film instability. When a severe asymmetric radial load pushes the shaft toward the x-axis, the film in the y-direction loses its load-carrying capacity. Cavitation within the oil film disrupts its continuity, creating unstable lubrication conditions, particularly under high-load conditions [39]. This degradation results in stabilizing forces being replaced by destabilizing cross-coupled fluid forces [40]. This condition, defined in the literature as the transition from oil film-induced orbital oscillation to severe hydrodynamic instability (oil whirl to oil whip) [41], manifests mathematically as negative coefficients in our model. This confirms that the bearing operates in a highly unstable regime prior to the balancing intervention.

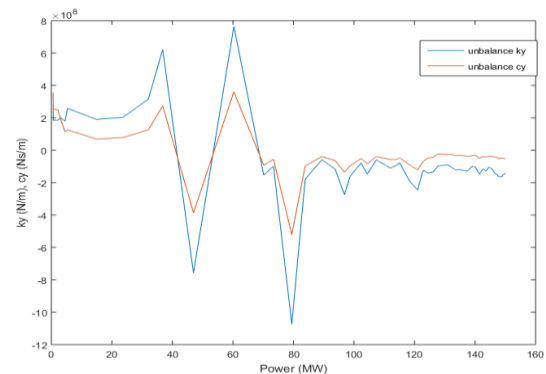


Figure 3. k_y and c_y values according to the power change in the hydraulic generator in fault condition (hydraulic unbalance state).

After the hydraulic unbalance was eliminated through rotor balancing and bearing clearance restoration, the stiffness (k_x) and damping (c_x) characteristics of the turbine-bearing system became significantly more stable compared to the unbalanced condition. As the generator load increases from 0 MW to 150 MW, k_x exhibits a smooth, gradual decline rather than the sharp fluctuations observed in the fault state, indicating a consistent hydrodynamic film thickness and improved load distribution along the bearing surface. Similarly, the c_x values remain low and nearly uniform throughout the entire operating range, demonstrating that the damping mechanism no longer experiences the abrupt dynamic loading caused by unbalanced radial forces. The

absence of sudden stiffness peaks and damping reversals confirms that the bearing operates within its intended dynamic regime, with no signs of cavitation-driven instabilities or oscillation-limit exceedance. These balanced-operation characteristics are illustrated in Figure 4.

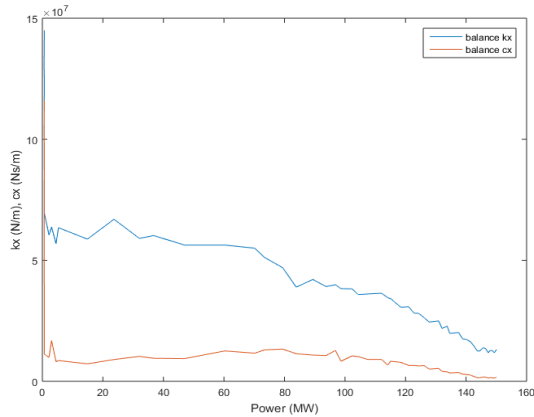


Figure 4. Values of k_x and c_x according to the power change in the hydraulic generator after the fault is eliminated (hydraulic balance state).

In the balanced operating state of the hydraulic turbine generator unit, the stiffness (k_y) and damping (c_y) coefficients in the y direction exhibit a stable and well behaved trend across the full load range of 0–150 MW. Following the elimination of rotor unbalance and the restoration of bearing geometry, k_y maintains a nearly constant value with only minor fluctuations, indicating a uniform hydrodynamic film thickness and proper radial load distribution along the bearing surface. The c_y values also display a low amplitude and consistent pattern, reflecting the restored damping capability of the bearing without the abrupt dynamic variations seen in the fault condition.

A noticeable peak is observed around 8 MW, which corresponds to the initial synchronization interval when the synchronous generator is connected in parallel to the interconnected grid. This peak reflects the transient electromagnetic impulse originating from the power grid during synchronization. Such impulses can occasionally be transmitted through the mechanical chain into the hydraulic components of the turbine, and in rare circumstances, may even cause the turbine generator unit to disconnect from the grid coupled operation.

Despite this short transient response, the overall stability and smoothness of the k_y and c_y curves confirm that the rotor bearing system operates in a dynamically balanced condition across the entire load range. The balanced state behavior of these coefficients is illustrated in Figure 5.

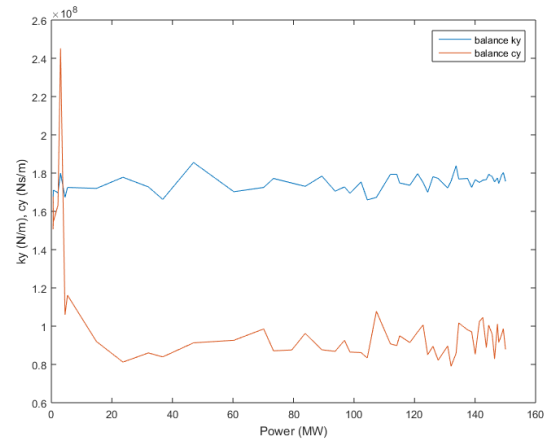


Figure 5. k_y and c_y values according to the power change in the hydraulic generator after the fault is eliminated (hydraulic balance state).

Based on the operating parameters of the real hydropower plant, the mathematical model of the shaft–bearing system was simulated within acceptable limits to reproduce the dynamic behavior observed under hydraulic unbalance. The simulations were performed using the parameter distribution method, with a repetition step of 280 iterations and a time step of 0.1 s, ensuring sufficient temporal and spatial resolution to capture the transient orbit of the rotating shaft. To validate the model, the numerical results were compared with the displacement data measured by the turbine’s vibration sensors under the same unbalanced operating conditions.

Figure 6 presents the comparison between the mathematical model values (“MM”) and the sensor based measurements (“S”), where both datasets display highly similar orbit patterns and displacement magnitudes. In both the model and sensor data, the shaft exhibits intensive motion around 100 μm , with peak displacements approaching approximately 400 μm . The close agreement in the shape, amplitude, and angular distribution of the orbits indicates that the mathematical model accurately captures the hydrodynamic behavior of the bearings and the resulting lateral motion of the shaft under unbalanced conditions. This strong correlation confirms the reliability of the model in representing the actual vibration response of the hydraulic turbine-generator unit during fault operation.

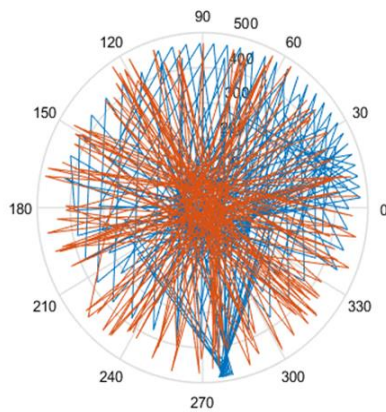


Figure 6. Comparison of the shaft orbit patterns obtained from the mathematical model (MM) and from sensor measurements (S), both under the unbalanced state. The blue curve represents the model results; the orange curve represents the sensor measurements.

After the rotor unbalance was corrected and the hydraulic turbine-generator unit was restored to its balanced operating condition, the sensor-based shaft displacement measurements (S) obtained under this balanced state were compared with the mathematical model (MM) results obtained under the unbalanced condition. Unlike the unbalanced state, in which the orbit of the shaft exhibits large amplitudes and irregular trajectories, the balanced case demonstrates a highly compact and symmetric displacement pattern. This reflects the recovery of stable hydrodynamic lubrication in the bearings and the elimination of excessive lateral forces acting on the shaft.

The marked reduction from the wide orbit predicted by the model for the unbalanced condition to the compact orbit measured by the sensors in the balanced condition indicates the extent of improvement between numerical simulation and real-system behavior. The sensor data show that shaft displacement is reduced to a narrow region around the center, confirming that the rotor operates with minimal eccentricity and that the bearing film thickness remains uniform. This consistency between the model's unbalanced-state baseline and the measured post-correction data validates the robustness of the mathematical model not only in fault diagnosis but also in characterizing the turbine's dynamic response before and after corrective maintenance.

Figure 7 presents a comparison between the mathematical model results corresponding to the unbalanced state and the sensor measurements corresponding to the balanced state.

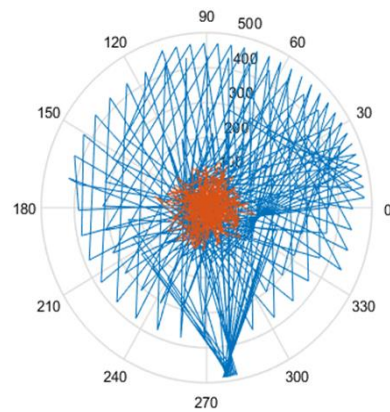


Figure 7. Comparison of the shaft orbit patterns obtained from the mathematical model (MM) under the unbalanced state and from sensor measurements (S) under the balanced state. The blue curve represents the unbalanced-state mathematical model results, while the orange curve represents the balanced-state sensor measurements.

In the balanced operating condition of the hydraulic turbine-generator unit, the mathematical model (MM) and sensor-measured (S) shaft orbits were compared to evaluate the consistency of the system's dynamic response after the unbalance fault was eliminated. In contrast to the wide and irregular orbit trajectories observed during the unbalanced state, the balanced condition exhibits a compact and uniform displacement distribution, indicating that the shaft rotates close to its geometric center with minimized eccentricity. This behavior reflects the recovery of stable hydrodynamic lubrication and the effective redistribution of radial forces once the balancing operations and bearing adjustments were completed.

The displacement amplitudes obtained from both the model and sensor measurements show strong agreement, with the shaft exhibiting small wandering motions around approximately $50\text{ }\mu\text{m}$ and reaching maximum displacements near $100\text{ }\mu\text{m}$. This close correspondence between the simulated and measured orbits demonstrates that the mathematical model successfully reproduces the real physical behavior of the shaft under balanced hydraulic conditions. The similarity in both amplitude and spatial orbit shape confirms that the model accurately captures the stabilized bearing–shaft dynamics and can be reliably used to predict system behavior under normal operating conditions.

The comparison of mathematical model and sensor orbits in the balanced state is illustrated in Figure 8.

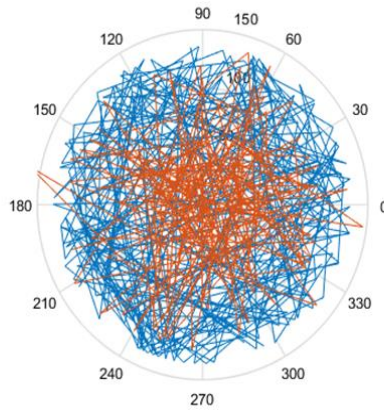


Figure 8. Comparison of the shaft orbit patterns obtained from the mathematical model (MM) and from sensor measurements (S), both under the balanced state. The blue curve represents the model results; the orange curve represents the sensor measurements.

To evaluate the consistency between the mathematical model and the actual dynamic behavior of the hydraulic turbine shaft, the orbit diagrams and time-domain displacement trends obtained in both the unbalanced and balanced operating states were compared using data from Figures 9 and 10. In the unbalanced condition, the turbine shaft exhibits large-amplitude and irregular orbital motion, with displacement values reaching up to approximately 400 μm . This behavior reflects the destabilizing hydraulic and inertial forces acting on the coupled system—comprising the excitation rotor, generator rotor, turbine runner, and the entire rotating mass—causing the shaft to oscillate away from its geometric center. The guide bearings are unable to maintain uniform film thickness under these forces, leading to asymmetric load distribution and significant lateral vibration.

After the balancing procedure and bearing clearance correction, the shaft displacement decreases dramatically, reducing to a compact orbit with peak amplitudes of around 100 μm . The nearly symmetric and centered orbit patterns in the balanced condition demonstrate that the hydraulic turbine operates within stable hydrodynamic lubrication limits. The reduction of shaft displacement by nearly three-quarters indicates that the radial hydraulic forces and mass-eccentricity-driven excitations have been effectively eliminated. Both the orbital diagrams and the time-domain responses clearly show that the shaft moves harmoniously toward the center, confirming the restoration of stable operating conditions.

A direct comparison of Figures 9 and 10 shows that the mathematical model reproduces the shaft displacement behavior obtained from sensor measurements with high fidelity. The similarity in orbit shape, displacement amplitude, and temporal vibration characteristics confirms that the developed model accurately captures both the

faulty and corrected dynamic states of the hydraulic turbine. This agreement verifies the reliability of the mathematical model for diagnosing unbalance faults and predicting post-maintenance system performance.

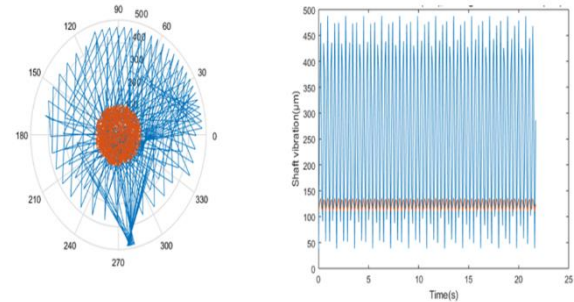


Figure 9. Comparison of mathematical model values for hydraulic unbalance and balance states (blue: unbalance, orange: balance).

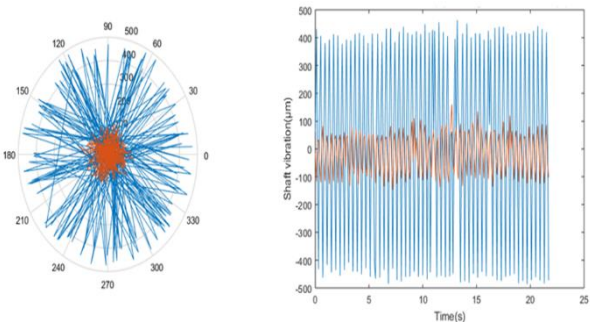


Figure 10. Comparison of sensor values in hydraulic unbalance and balance states (blue: unbalance, orange: balance).

Overall, the comparative evaluations of the orbital and time-domain displacement responses demonstrate that the hydraulic turbine-generator unit exhibits a distinctly different dynamic behavior in unbalanced and balanced operating states. The mathematical model and sensor measurements show strong agreement in both conditions, confirming that the model reliably reproduces the real system dynamics. In particular, the significant reduction in shaft displacement—from approximately 400 μm in the unbalanced state to nearly 100 μm after balancing indicates that the corrective procedures effectively restored stable hydrodynamic lubrication and centralized shaft rotation.

The stiffness–damping trends and orbit behavior identified in this study are consistent with observations reported elsewhere for Francis turbine-generator units. Ri et al. [22] similarly found that sliding-bearing stiffness strongly governs the dynamic response of hydro-generator shafts, a finding broadly consistent with the stiffness–damping sensitivity observed in the present k_x – c_x and k_y – c_y behavior. Rondon et al. [42] reported that tilting-pad bearing stiffness and damping in vertical rotors are highly

sensitive to load and rotor position, a sensitivity consistent in nature with the load-dependent stiffness variation observed in the 35–90 MW range of the present unit. Field measurements on a 144 MW Francis turbine by Bojic et al. [43] likewise showed that relative shaft vibration amplitude at the turbine guide bearing increased continuously with generator load under hydraulic unbalance, while the static orbit position shifted only modestly, closely matching the load-dependent stiffness–damping fluctuations and orbit growth reported here for the unbalanced condition. Similarly, Sapkota et al. [44] observed experimentally that mass unbalance on a Francis runner alters the resonant peak amplitude of the shaft without substantially changing the surrounding frequency content, consistent with the localized stiffness peaks and damping reductions identified in this work rather than a broadband instability. Taken together, these comparisons support the physical plausibility of the coefficients and orbit trends obtained for the present 150 MW unit and situate the findings within the broader body of Francis turbine shaft-vibration research.

These findings collectively validate the accuracy of the mathematical model, confirm the effectiveness of the balancing process, and highlight the importance of monitoring vibration-based indicators for maintaining operational stability. The insights obtained from these analyses form the basis for the broader conclusions presented in the following section.

Conclusions

This study examined the influence of hydraulic unbalance on the dynamic behavior of a 150 MW Francis turbine-generator unit by combining a nonlinear shaft–bearing mathematical model with field vibration measurements. The comparison of model-predicted and sensor-measured responses revealed that unbalance produces a distinctly unstable dynamic regime characterized by large orbit amplitudes, sharp variations in stiffness and damping coefficients, and asymmetric load distribution within the hydrodynamic bearings. In the unbalanced condition, shaft displacement reached approximately 400 μm , and stiffness–damping fluctuations were especially prominent between 35–90 MW, indicating critical reductions in film stability under increasing hydraulic loading.

After balancing the rotor and restoring the bearing clearance, the turbine exhibited a significant improvement in dynamic stability. Peak shaft displacement decreased to around 100 μm , representing a reduction of nearly 75 %, and the orbital motion became compact and centered. The balanced state also produced smooth variations of the load-dependent stiffness and damping coefficients in both radial directions, demonstrating that the corrective maintenance restored the uniformity of the hydrodynamic lubrication film.

The strong agreement between the mathematical model and the sensor measurements in both faulty and corrected conditions confirms the reliability of the model in reproducing the real system dynamics. This provides a valuable foundation for using model-based diagnostics in operating hydroelectric power plants. The results demonstrate that the evaluated stiffness (k) and damping (c) parameters, together with shaft orbit characteristics, can serve as sensitive indicators for detecting early-stage unbalance and monitoring bearing health. The study therefore highlights the potential of integrating such a mathematical model with vibration-based monitoring systems to support predictive maintenance, minimize downtime, and maintain safe operating conditions over the full load range of the turbine.

In summary, the findings show that the proposed modeling approach accurately characterizes the mechanical behavior of the turbine-generator shaft line, effectively distinguishes between balanced and unbalanced conditions, and provides actionable diagnostic information for real-time fault prediction. These outcomes suggest that the methodology can be further developed by incorporating additional physical phenomena such as gyroscopic coupling or unbalanced magnetic pull, and by applying the framework to multiple units to establish generalized diagnostic thresholds for large-scale hydropower systems.

Ethics committee approval and conflict of interest statement

Conflict of interest All authors declare that they have no conflict of interest.

Authors' Contributions

Ozdemir O: Study design, data processes, and manuscript writing and critical review.

Yucel HL: Critical review

Kahraman G: Study design, data processes, manuscript writing.

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