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TO,
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SUB: STUDY OF HIGH SHAFT VIBRATION PROBLEM OF TG#6 [500 MW]

Respected Sir,

Thank you for giving NIMDC Private Limited the opportunity to conduct a successful three-day training on "**Vibration Monitoring and Analysis of Turbine Generators**" at Anpara. Executives from various plants, including Anpara, Obra, Panki, Parichha, and Harduaganj, attended the training. During the demonstration of the installed online vibration monitoring system, a discussion was held regarding TG unit #6, and the Chief General Manager requested a preliminary report to inform the line of action.

The online vibration monitoring system's features were demonstrated by analyzing real-time steady-state and transient data from the high-shaft-vibration problem on HPT-TG #6. The initial evaluation showed the HPT front bearing had a shaft vibration amplitude of 320 microns pk-pk, while the rear bearing had 125 microns pk-pk.

- **Bearing No. 2**: Exhibits severe directional preloading, restricting the hydrodynamic oil film and shifting the journal centerline position. [Action]: Verify the EC ratio; it is built into the Bently system.
- **Bearing No. 3:** Suffers from highly asymmetric structural stiffness, indicating an unequal bearing support stiffness matrix that distorts the rotor's dynamic response.
- **Stiffness Calculation at BRG No. 3:** The enclosed calculation indicates a significant 14.62% reduction in effective stiffness, reflecting a substantial structural change that cannot be attributed solely to standard unbalance.

Recommendation for Advanced Diagnostics: To conclusively identify the root cause and prevent catastrophic asset failure, we strongly recommend conducting a comprehensive multichannel steady-state and transient data analysis using a **16+ channel simultaneous acquisition system**.

This advanced study is required to obtain critical parameters—such as all orbits, the EC ratios, the shaft-average centerline plots (SACP), Bode & Polar plots, half & full spectrum plots, Synchronous Amplification Factor (SAF) for the measurement of all bearing damping, separation margin, APHT & Acceptance region plots, DC gap plots, probe rotation, and waterfall & spectrum cascade plots—for the LP and Generator rotors as well.

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This advanced study is required to:

Core objectives of the advanced study

- **Map Precise Trajectories:** Obtain orbit trajectories, centerline plots, and Bode/Nyquist profiles of the Low-Pressure (LP) turbine, Generator rotors, and the exciter (not available during initial study).
- **Assess Cross-Coupling Effects:** Evaluate cross-coupling across the entire machine train to identify structural or hydrodynamic anomalies that could cause self-excited forward whirl or catastrophic damage.
- **Determine Parameter Dependencies:** Evaluate the stability of the entire rotor-bearing system to determine whether the observed vibrations are load-dependent or speed-dependent.
- **Analyze Structural & Alignment Loading:** Assess catenary adjustments and asymmetric bearing loading across the train.
- **Quantify System Damping:** Calculate the Synchronous Amplification Factor Q_S to quantify bearing damping characteristics accurately.

Technical Justification & Risk Mitigation: Rotor stability depends heavily on physical shape, bearing characteristics, and fluid-film damping factors. Misalignment and negative damping introduce a severe risk of system instability. Specifically, negative damping can trigger destructive sub-synchronous instabilities, which can be detected using orbit plots, waterfall plots, and Campbell diagrams to pinpoint issues such as oil whirl, whip, or rubs.

Action Plan & Structural Alignment Assessment: The final corrective action plan will focus on assessing whether structural settlement or thermal growth has altered the catenary alignment at the HP-IP, IP-LP, and LP-Generator couplings. By combining this multichannel dynamic data with a thorough physical inspection of the bearings, we will accurately distinguish structural constraints from pure hydrodynamic instability, providing a definitive roadmap for rectification.

Thank you once again for your trust in NIMDC Private Limited.

With Kind Regards,

Sincerely yours,

17/03/2026

(C.G. Porwal)

Chairman & Managing Director

COPY FOR KIND INFORMATION, PLEASE

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PRELIMINARY REPORT ON ANPARA TPS TG#6

1.0 INTRODUCTION

UPRVUNL, with an installed capacity of 6,134 MW, includes the Anpara Thermal Power Station, with a capacity of 2,630 MW. The 500 MW units at Anpara (Units #6 and # 7), part of the Anpara D expansion, were designed and built by BHEL. Unit #6 (500 MW) began operation in March 2015, and Unit #7 (500 MW) in March 2016.

2.0 HISTORY OF THE WORK DONE SO FAR

TG Unit #6 has been experiencing high shaft vibration since the generator replacement. This problem has caused a 4-day downtime after every trip, raising concerns among site and management personnel.

3.0 OBSERVATIONS

On September 12, 2026, we analyzed steady-state and transient data, including shaft orbit plots, Bode plots, and vibration signatures from the online monitoring system. The analysis was performed at 2997 rpm and 299 MW for bearings 1, 2, and 3. However, data for bearings 4, 5, 6, and 7 couldn't be evaluated for LP and generator behavior due to a Meggitt system or card issue.

4.0 DIAGNOSTIC & DATA ANALYSIS

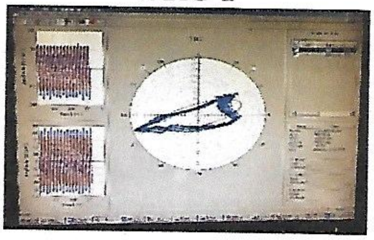
TG Unit #6 (HPT Shaft Dynamics): During the observation period, both the front and rear HPT shaft journals exhibited significant dynamic excursions, with peak-to-peak displacements of 320 microns (front) and 125 microns (rear), which exceed standard operational thresholds (ISO 10816-2 / ISO 20816-2) and necessitate immediate corrective action. Analysis of steady-state and transient data from the online Turbine Supervisory Instrumentation (TSI) revealed distinct diagnostic signatures:

a) **Orbit Plots:** The orbit recorded at bearings 1, 2, and 3 shows that bearing no. 2 is heavily preloaded, with a clear, elongated, banana-shaped trajectory and a dense, concentrated trace, indicating severe mechanical preload or shaft misalignment. The flattened orbit indicates that the shaft is heavily constrained in one radial direction, preventing it from forming a normal circular or elliptical orbit.

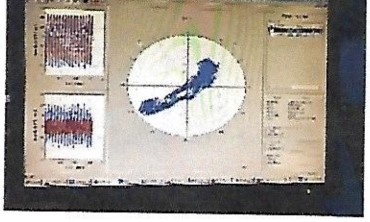
Secondary Indicators: The loop crossover (often appearing like a figure-8 or a loop-within-a-loop) in the orbit pattern indicates the presence of strong non-linear forces, typically caused by extreme reaction forces that push the shaft beyond acceptable limits.

ORBIT PLOT

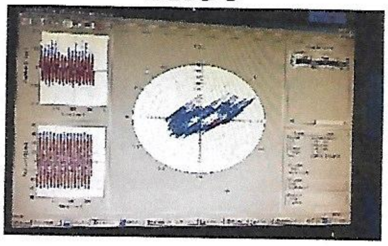
BRG 1



BRG 2



BRG 3

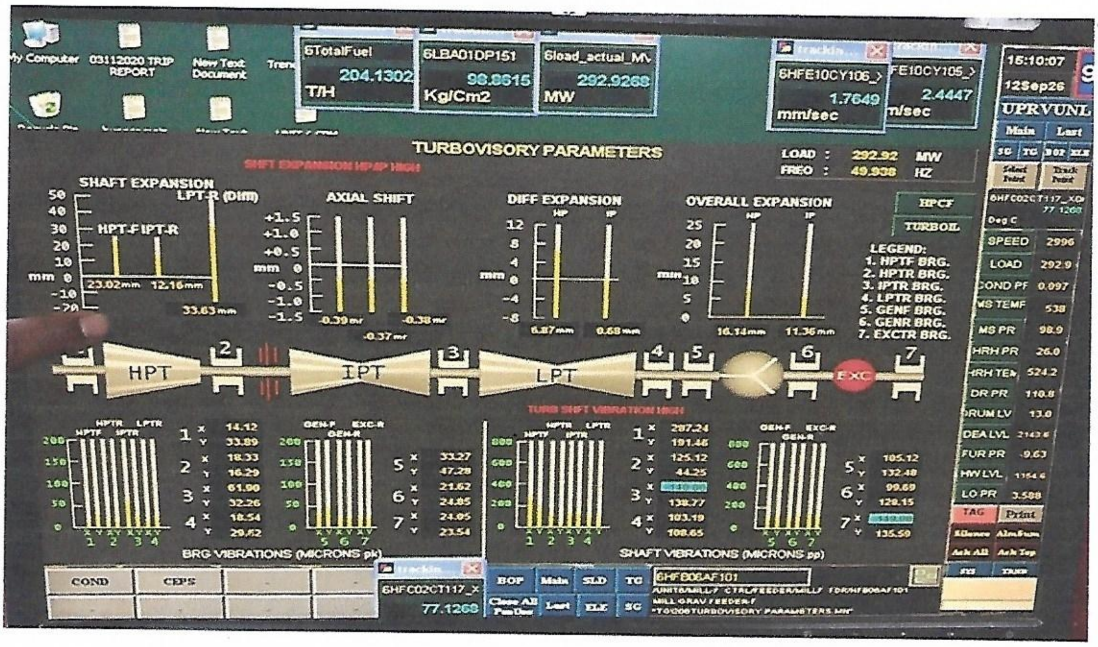


Journal Centerline and Eccentricity Analysis: To validate severe directional preloading on Bearing No. 2 and evaluate the hydrodynamic oil film condition, we calculate the Eccentricity Ratio (ε) and Attitude Angle (φ). These metrics are integrated into the Bently system. The presentation explained the EC ratio. Details are below.

The formula is: $\varepsilon = e/c$, with e representing the journal center displacement and c the radial clearance. A typical journal exhibits ε values between 0.4 and 0.7, whereas an ε close to 1.0 suggests significant preload or oil film failure.

b) **Correlate Thermal Expansion:** At a partial load of 299 MW out of 500 MW, the unit exhibits uneven casing expansion, with an absolute differential expansion (DE) of 6.87 mm, which exceeds critical thresholds for a 500 MW turbogenerator. Normal operating limits for differential expansion in large utility turbines generally range from ±3.0 mm to ±5.0 mm, depending on OEM (Original Equipment Manufacturer) specifications. Values exceeding 6.0 mm trigger an automatic trip zone.

Casing Expansion & Pipe Stress Survey: Inspections should include HPT sliding keys and guide slots to confirm that the turbine casing expands freely as it heats. Additionally, examine main steam piping supports and hangers, as frozen hangers can adversely affect turbine casing alignment and lead to the directional preloading seen at **Bearing No. 2**.

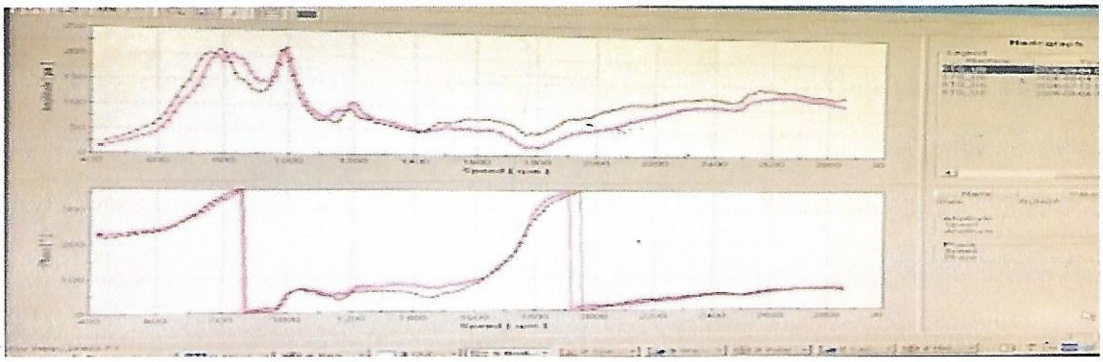


5.0 UNEQUAL BEARING STIFFNESS:

The historical Bode plot suggests that BRG No. 1 & 2 are operating within normal parameters; however, a critical speed shift is noted for (IPT) BRG No. 3, from 895 to 827, likely due to split resonance caused by bearing asymmetry. The critical speed shift is accompanied by an amplitude dip and is caused by split resonance.

5.1 QUANTIFICATION OF ROTOR STIFFNESS REDUCTION DELTA K

The critical speed of the rigid-rotor natural frequency is related to the effective modal stiffness (K) and modal mass (M) by the following relationship: $[\omega(n)]^2 = K/M$, where $\omega = 2\pi N/60$.



i) THE ANGULAR VELOCITY CONVERSION (RAD/SEC)

The IP rotor weight supported between bearings is derived from its mass of 23 tons (23,000 kg). The critical speed has shifted downward from 895 RPM to 827 RPM. Converting these values to angular velocity {rad/s} yields:

Original design critical speed	ω_n1=895 RPM X 2π / 60 = 93.724 rad/s.
Shifted critical speed	ω_n2=827 RPM X 2π / 60 = 86.603 rad/s.

ii) RELATIVE STIFFNESS RATIO (K2/ K1)

Because the physical modal mass M=23000 kg rotor remains a constant parameter, the ratio of the modified stiffness K2 to the original design stiffness K1 is:

$$K2/K1=[\omega\_n2/\omega\_n1]^{\wedge}2 = [86.603/93.724]^{\wedge}2 = [827/895]^{\wedge}2 = [0.92402]^2 = 0.85382$$

iii) SYSTEM STIFFNESS LOSS:

$\Delta K = 1 - 0.8538 = 0.1462$ [14.62% reduction in effective stiffness].

This 14.62% reduction represents an enormous structural shift for a massive 500 MW machine. Standard residual unbalance cannot cause this, indicating a substantial loss of the system's geometric constraints or load-bearing capability at Bearing No. 3.

iv) LIKELY MECHANISM:

Potential issues may stem from structural degradation beneath the pedestal, including cracked or deteriorated epoxy/cementitious grout, loose or broken foundation anchor bolts, or damage to the bearing housing support structure/pedestal seat.

xxx

12/01/2026