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Analytical and Experimental Study of Lateral Vibrations in a High-Speed Gear Unit

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Abstract

This paper presents a comprehensive investigation into a significant lateral vibration issue encountered in a high-speed industrial gearbox used as the central unit of an electro-compressor train. During initial commissioning, the gearbox was unable to reach its rated operating speed due to excessive vibration levels (> 50 µm peak-to-peak), prompting a detailed diagnostic analysis. The root-cause study identified that the critical frequency of the pinion shaft was very close to the gearbox's operating speed, resulting in resonance. Two potential solutions were considered to shift the first critical speed outside the operating range. Increasing the mass of the pinion to lower its critical speed was deemed impractical due to design constraints and standard limitations. Consequently, the alternative approach of increasing the shaft diameter to enhance stiffness and elevate the critical frequency was implemented. Numerical simulations confirmed that this modification successfully moved the critical speed well above the operating range, ensuring safe operation in a subcritical regime. Subsequent mechanical running tests validated the effectiveness of the redesign, showing a substantial reduction in radial vibration amplitudes. Measured vibration levels at both the low-speed and high-speed shafts were below 20 µm (peak-to-peak), confirming the successful mitigation of resonance. The study demonstrates the value of integrating analytical modeling, simulation, and targeted engineering modifications to resolve complex vibration problems in geared rotor systems. The implemented solution not only restored reliable gearbox operation but also provides a practical industrial experience for addressing similar high-speed vibration challenges in industrial mechanical drives.

Keywords: High-speed gearbox; Shaft critical frequency; Vibration analysis; Critical Frequency Shifting.

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1. Introduction

High-speed gear units integrated with centrifugal compressors are critical components in many industrial applications, yet they are susceptible to lateral vibrations that can lead to performance degradation and mechanical failure. These lateral vibrations result from complex dynamic interactions inherent to high-speed rotating machinery, including lateral resonances, gear-mesh and seal excitations, bearing support stiffness and coupling dynamics; mitigation focuses on high-speed (HS) shaft and low-speed (LS) shaft design, bearing design, gear phasing and housing stiffness.

Lateral vibration levels can be reduced through targeted design and operational strategies, as shown by Jenny and Wyssmann, who explored lateral vibration reduction techniques in high-pressure centrifugal compressors [1]. Field verifications by Wachel and Szenasi [2] provided direct evidence of lateral-torsional coupling effects on rotor instabilities, reinforcing the importance of integrated vibration monitoring and control strategies in industrial compressors. Likewise, Pugnet et al. [3] examined how sealing system design influences vibratory behavior. Furthermore, Passeri et al. [4] demonstrated how dynamic design guided by forced response analysis can inform compressor general arrangement to minimize vibrational impact.

Subsequent industrial research has emphasized the necessity for practical observation, assessment, and resolution methods. Case studies by Seon et al. [5] and investigations into super-synchronous vibration phenomena confirm that these behaviors require comprehensive dynamic analysis and tailored mitigation approaches. Iannuzzelli and Elward [6, 7] demonstrated that lateral vibrations can couple with torsional modes, causing instability in geared rotor systems. Hudson and Wittman [8] documented cases of sub synchronous instability in overhung geared centrifugal compressors, demonstrating that targeted bearing modifications can be required to eliminate critical vibration issues. Rakhit [9] introduced an epicyclic gearbox design that mitigates sub-synchronous vibrations, thereby enhancing operational reliability in gas turbine generator applications. Similarly, Maragioglio et al. [10] validated high-speed ratio epicyclic gear technologies that contribute to vibration control in compression systems.

Corcoran et al. [11] highlighted that high-performance coupling can dramatically affect natural frequencies and bearing loads. Similarly, Kirk et al. [12] provided theoretical and practical perspectives on instability mechanisms in geared couplings, establishing foundational principles for analysing and mitigating vibration risks. Other contributions to vibration mitigation include multi-resonance vibration mitigation strategies for large horsepower parallel shaft speed increasers and automatic optimization of dynamic balancing in rotor compressors. Evans et al. [13] and Deng et al. [14] have shown that such approaches yield tangible improvements in reducing lateral vibration amplitudes and enhancing the dynamic stability of high-speed geared machinery. Recent research by Zhang et al. [15] on self-excited vibration mechanisms considering multipoint mesh and variable bearing dynamics deepens the understanding necessary to predict and control lateral vibrations in gear systems. Studies of high-speed main shaft designs, by Bielecki and Costagliola [16], have further contributed to the design basis for stable, vibration-resistant shafting

These combined works establish a framework for understanding, diagnosing, and designing against the various forms of lateral vibration encountered in high-speed industrial gear units. In this study, a persistent lateral vibration problem of an industrial gearbox is addressed through a detailed engineering investigation. A redesign of the high-speed shaft was implemented to modify the system dynamics, with the goal of significantly reducing vibration levels and improving operational stability.

2. Material and Method

A lateral vibration problem encountered in an industrial gearbox, during mechanical running test, is analysed through experimental measurements and numerical modeling, aiming to reveal the key dynamic factors contributing to the vibration and to suggest potential design improvements.

2.1 Gearbox Description

The subject of this study is a high-speed industrial gearbox, operating in conjunction with a centrifugal compressor and subject to measurable lateral vibration issues under mechanical running test conditions. The gearbox is a parallel shaft, double-helical speed increaser, designed based on [17] with nominal input and output speeds specified in accordance with application requirements. The cross-sectional drawing of this gearbox is shown in Fig. 1. All relevant design parameters, such as gear module, shaft dimensions, rated speeds, rated power and teeth numbers are presented in Table 1. This data will support dynamic modeling, analysis, and redesign tasks.

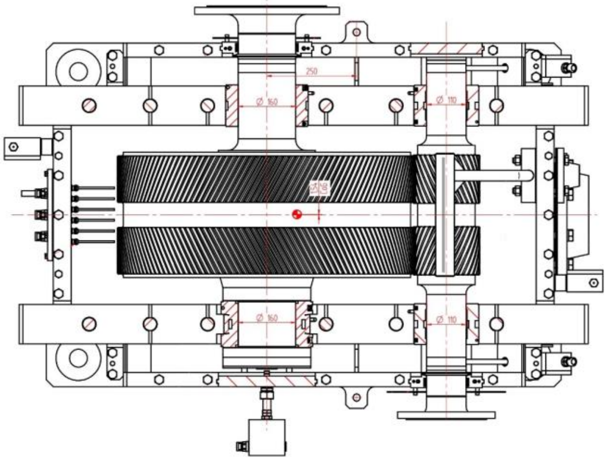

Figure 1. The cross-sectional drawing of the double-helical speed increaser gearbox, studied in this paper.

Table 1. Summary of the gearbox design parameters.

Design Parameter	Unit	Values	Notes
Gear Unit Rated Input Speed	rpm	2,989	Operating Speed Range:
Gear Unit Rated Output Speed	rpm	14,077	80% to 105% of the rated speed
Gear Unit Rated Power	kW	8,910	-
Gear Module	mm	5	-
Gear Wheel Number of Teeth	-	146	-
Pinion Number of Teeth	-	31	-
Gear Shaft / Pinion Shaft Diameter	mm	160 / 110	-
Gear Shaft / Pinion Shaft Mass	kg	1585 / 124	-
Gear Shaft / Pinion Shaft Moment of Inertia	$kg.m^2$	114.28 / 8.96	
Low Speed / High Speed Half Coupling Mass	kg	58.2 / 21.5	Attached to the shaft end.

2.2 Vibration Measurement Setup

A comprehensive vibration monitoring system, existing in the gearbox manufacturer test shop, was implemented. Ten radial and axial vibration detectors (proximity non-contact probes for displacement measurements), two accelerometers, and one key-phasor were installed at key locations on the gearbox housing and near the bearings to capture translational and rotational dynamic response. The number of radial and axial vibration probes are listed in Table 2.

Table 2. Summary of the gearbox design parameters.

Probe	Total Number	Placement Details
Radial Vibration Detectors (proximity non-contact probes)	4	X-Y probes at gear bearings
Radial Vibration Detectors (proximity non-contact probes)	4	X-Y probes at gear bearings
Axial Vibration Detectors (proximity non-contact probes)	2	Dual probes at the thrust bearing
Key-phasor (one event per revolution probe)	1	Gearbox input shaft
Accelerometer	1	Gearbox housing (gear coupling end)
Accelerometer	1	Gearbox housing (pinion coupling end)

Sensor placement was chosen based on API 670 [18] guidelines. Data acquisition was performed with a digital system capable of high sampling rates to ensure accurate frequency domain representation. The sampling rate of at least twice the highest frequency component of the signal suffices, according to the Nyquist–Shannon sampling theorem. But in practical vibration monitoring, sampling rates often exceed the minimum Nyquist requirement to improve data quality and reliably capture gear mesh harmonics (in this paper the sampling rate of four times the maximum gear-mesh frequency is considered). Baseline vibration data were collected during mechanical running test at no-load condition. The gearbox vibration measurement setup is shown in Fig. 2.

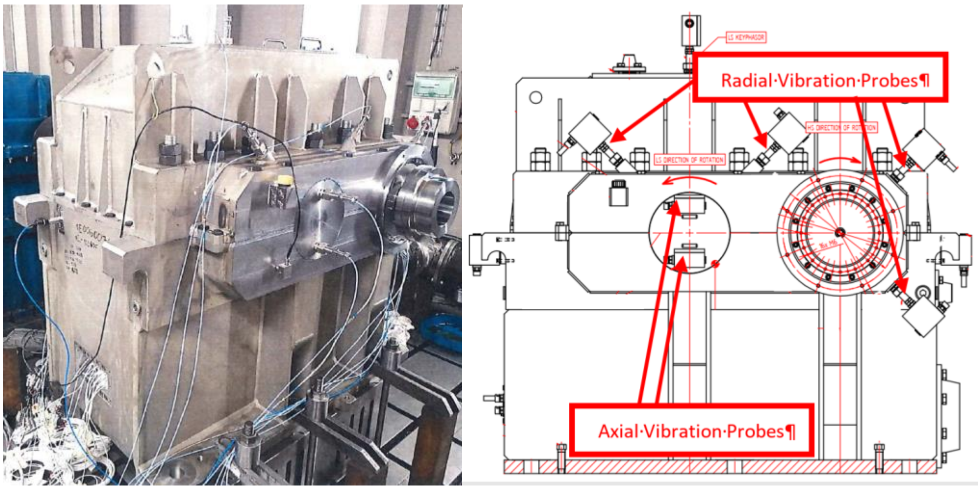

Figure 2. Gearbox vibration measurement setup.

2.3 High-Level Vibration

The high-speed gearbox (as detailed by the specifications in Table 1) was designed, manufactured, and scheduled for a mandated mechanical running test to validate its structural integrity and dynamic performance under operating conditions. Mechanical running test is a requirement of API 613 in industrial commissioning.

During the pre-test phase, as the rotational speed was being incrementally increased toward the rated speed, an excessive level of vibration was detected. This dynamic instability was quantified, with the vibration amplitude exceeding the acceptance threshold of 50 µm (peak-to-peak). Consequently, to mitigate the immediate risk of damage and to protect the integrity of the test rig, the mechanical running test was immediately terminated. The gearbox was therefore unable to achieve its full rated operating speed.

Following the test cancellation, a comprehensive Root Cause Analysis (RCA) was initiated. The primary objective of this technical investigation was to isolate the fundamental mechanism responsible for the excessive vibration (e.g., rotor imbalance, misalignment, structural resonance, or manufacturing tolerance deviation) and to develop a definitive engineering solution. The implementation of the necessary corrective actions (based on the RCA findings) was a prerequisite for any future attempt to resume and successfully complete the mechanical running test.

The systematic RCA identified a fundamental issue related to the dynamic characteristics of the gearbox. Specifically, the analysis revealed that one of the critical frequencies of the high-speed rotor (corresponding to the 1st flexible mode of the pinion shaft) was found to be in close proximity to the gearbox's rated operating speed (as shown in Figs. 3 and 4). This near-coincidence of the critical frequency and the rated speed led to a resonance phenomenon. When the gearbox was operating near this speed, resulting in the excessive vibration amplitude that led to the test cancellation. Thecalculations of the critical frequencies are performed using FEM, by software ROTLAT. Number of nodes and elements are presented in Figs. 4.

To eliminate the gearbox resonance vibration during normal operation, two primary engineering solutions were proposed, both aimed at sufficiently separating the critical frequency from the rated speed. This separation is necessary to ensure the unit operates within an acceptable separation margin (typically defined by API 684 [19]). The two proposed strategies for modifying the critical frequency were as follows:

- Lowering the Critical Frequency (Supercritical Operation)
- Increasing the Critical Frequency (Subcritical Operation)

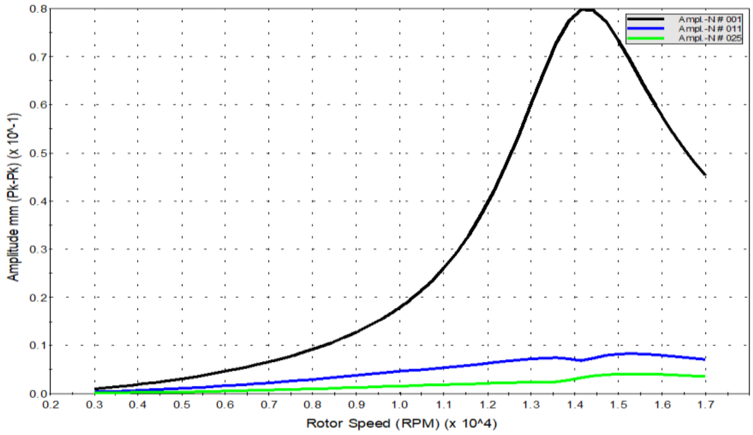

Figure 3. Frequency response of the pinion shaft. Critical frequency ≈ 14,400 rpm.

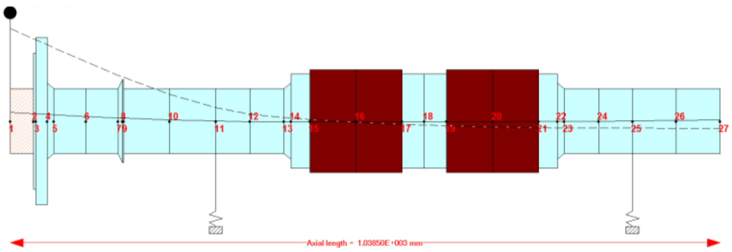

Figure 4. The first flexible mode shape of the pinion shaft, corresponding to the critical frequency of Fig. 3.

2.4 1st Proposed Modification: Lowering the Critical Frequency

The first solution for the problem involved engineering modifications to decrease the first critical frequency such that it falls significantly below the rated operating speed (safety margin value varies, based on amplification factor, AF [19]). This would ensure that the gearbox operates in a supercritical speed regime. To lower the critical frequency and make considerable margin relative to the rated speed, it was proposed to add an extra mass to the end of the pinion shaft, as shown in Fig. 5 (compare to Fig. 4). This added mass was specifically designed to be mounted without interference, freely inserting into the coupling hub cavity during assembly.

2.5 2nd Proposed Modification: Increasing Critical Frequency

Alternatively, this solution involved modifying the rotor-bearing system to increase the first critical frequency such that it is substantially above the rated operating speed (safety margin value varies, based on AF [17]). This would place the rated operation in a subcritical speed regime. The suggested modification was increasing the pinion shaft diameter, in order to increase its stiffness and its critical frequency, accordingly. The high-speed shaft design was evaluated and then modified with the aim of increasing the separation margin from excitation frequencies and reducing sensitivity to lateral excitation. The cross-sectional drawing of the initial pinion shaft and the modified one are presented in Fig. 6.

2.6 Post-Modification Testing

After implementation of the appropriate modification, the gearbox was subjected to identical mechanical running test, to investigate the performance of the modified gearbox.

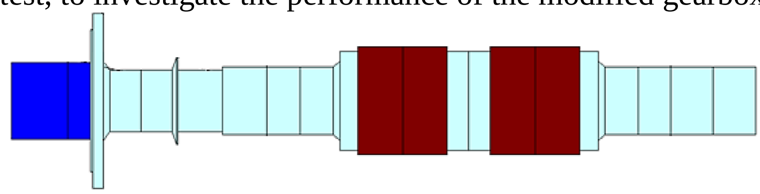

Figure 5. Adding extra mass to the end of the pinion shaft, in order to reduce its critical frequency.

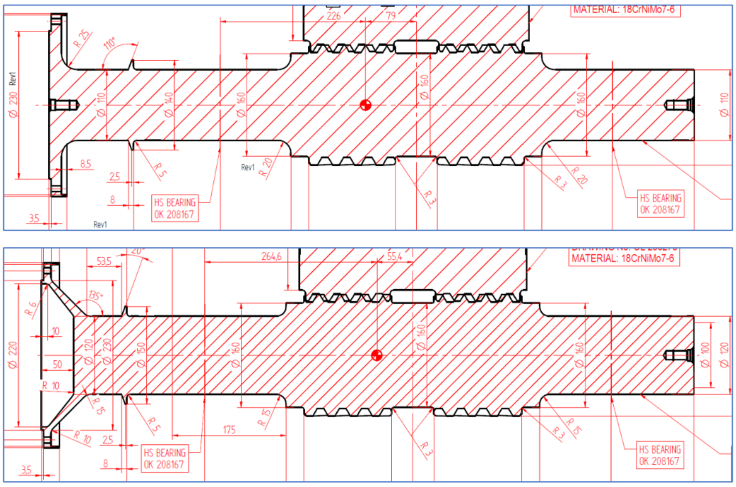

Figure 6. The cross-section of the initial pinion shaft (above) and the modified pinion shaft (bottom).

3. Results and Discussion

In this section the results of two proposed modifications are presented and analysed.

3.1 1st Proposed Modification: Added Mass to the Pinion Shaft

The first solution, namely adding an extra mass to the pinion, in order to lower the critical speed, was the fastest and the most economically feasible option to ensure the rotor operates away from its critical speed. Numerical simulations indicated that this solution would create a substantial margin between the critical speed and the rated operating speed, as shown in Figs. 7 and 8.

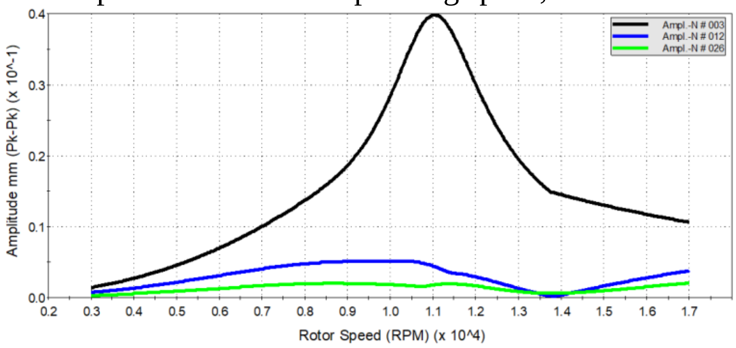

Figure 7. Frequency response of the pinion shaft, including an extra mass. Critical frequency ≈ 11,050 rpm.

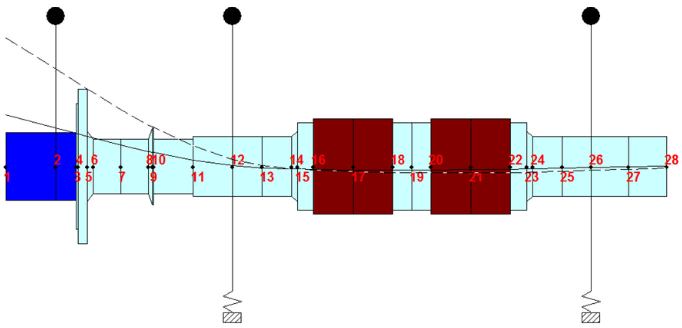

Figure 8. The first flexible mode shape of the pinion shaft, corresponding to the critical frequency of Fig. 7.

But this solution was rejected due to two main reasons:

- API 613 (2.5.4.1) states that "Shafts shall be machined from one-piece, heat-treated steel".
- Although the critical speed is moved far away from the rated speed, it is still in the operating speed range. Because the compressor driver is variable speed and would be operated in a speed range of 11,260 rpm to 14,780 rpm (see Table 1).

3.2 2nd Proposed Modification: Pinion Shaft Redesign

Following the rejection of the initial solution, the gearbox manufacturer ultimately agreed to redesign the pinion shaft, despite the higher cost, to resolve the resonance issue. In the new design, the pinion shaft diameter was increased from 110 mm to 120 mm. Numerical simulations indicated that this modification would raise the critical speed to values well above the operating speed range (>17,000 rpm). The corresponding simulation results are presented in Figs. 9 and 10.

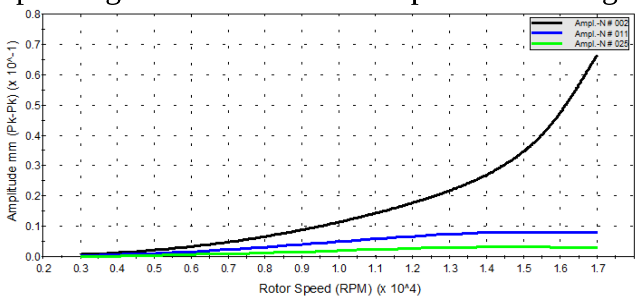

Figure 9. Frequency response of the redesigned pinion shaft.

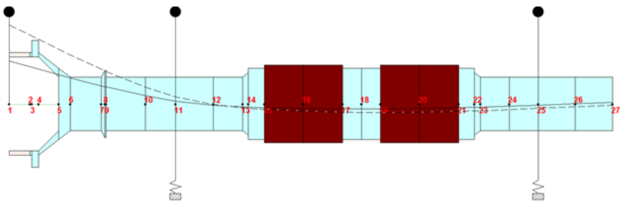

Figure 10. The finite element model of the redesigned pinion shaft.

3.3 Final Testing Results

Following the modification of the pinion shaft, the final mechanical running test of the gearbox was conducted. During this test, the measured levels of unfiltered radial vibration at both the low-speed and high-speed shafts were found to be less than 20 µm (peak-to-peak), indicating a marked improvement in dynamic behavior. Prior to modification, the vibration amplitude of the HS shaft exceeded 50 µm (peak-to-peak) before reaching the rated speed. Representative vibration signals recorded at the drive end of the pinion shaft are presented in Fig. 11. The corresponding spectrum is also presented at the bottom of Fig. 11. The figure quality is limited due to the vibration records being obtained as screenshots from the monitor of the vibration measurement device.

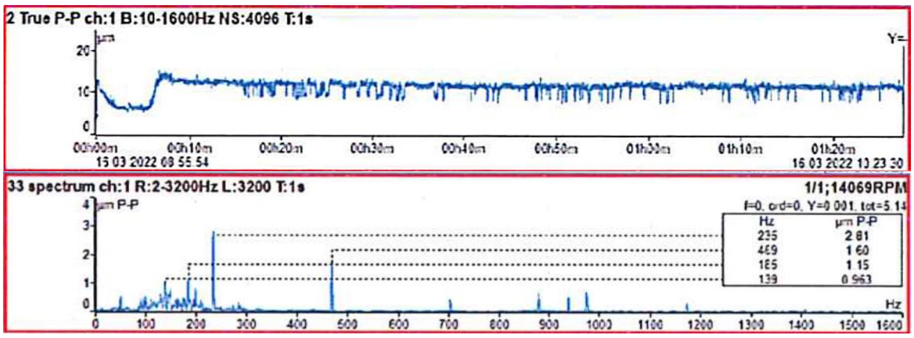

Figure 11. Top: The radial unfiltered vibration of the HS shaft drive end; Bottom: Corresponding spectrum.

4. Conclusion

This investigation addressed a significant lateral vibration problem encountered in a high-speed industrial gearbox operating as the middle equipment of an electro compressor train. Diagnostic testing initially revealed that the gearbox was unable to reach its rated operating speed due to elevated vibration levels, necessitating a halt to commissioning and prompting a root-cause analysis. The analysis showed that a critical frequency of the pinion shaft was so close to the gearbox's rated operating speed, leading to a resonance phenomenon. Two different approaches were proposed to shift the first critical speed well below or well above the operating speed range. Increasing the mass of the pinion, via adding extra mass to the shaft, to reduce its critical speed was found to be impractical. Therefore, the second approach, which involved increasing the shaft diameter to increase its stiffness and raise the critical frequency, was implemented.

By implementing a strategic redesign of the pinion shaft, the critical frequency was effectively separated from the gearbox's working speed range. Numerical simulations supported this approach by demonstrating an increased margin between the critical speed and operating speed, ensuring the system operated safely within a subcritical speed regime. Subsequent mechanical running tests verified a substantial reduction in the amplitude of radial vibrations, with levels at both the low-speed and high-speed shafts falling below 20 μm (peak-to-peak).

The results highlight the effectiveness of combining analytical modeling, simulation, and engineering modification in resolving complex vibration issues in geared rotor systems. This industrial experience not only restored reliable gearbox operation but also provides a valuable reference for troubleshooting similar vibration challenges in high-speed mechanical drives.

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