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Research Paper

Static Balancing of Centrifugal Fan

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Abstract:
Centrifugal fans are critical in cement manufacturing for airflow regulation and dust management but often suffer from vibration-related faults due to harsh operational conditions. Excessive vibration, commonly caused by rotor unbalance and misalignment, can lead to premature component failure. This study investigates a cost-effective method of vibration reduction through field-based static balancing. A 9-blade, rigid-mounted centrifugal fan operating at 1475 RPM was monitored using the SKF Quick Collect CMDT391 sensor and Emerson CSI 2140 analyzer. Initial vibration analysis revealed high amplitudes exceeding ISO 10816 limits. Mechanical faults such as impeller misalignment, loose bolts were corrected. The impeller was then divided into 40° segments, and static balancing was performed using trial weights of 5 g, 17 g, and 20 g at the light spot (12 o'clock). Final correction weight of 34.55 g was calculated using the influence coefficient method and welded in place. Initial horizontal vibration at the fan drive end was 17.38 mm/s. After mechanical corrections, it reduced to 12.48 mm/s. Following static balancing, vibration further dropped the vibration to 3.91 mm/s an overall reduction of 82 %. Similar improvements were observed across all measurement points. This demonstrates that systematic vibration diagnosis, basic mechanical correction, and static balancing can significantly reduce vibration levels. In resource-constrained industrial settings, this approach offers a practical alternative to dynamic balancing, ensuring equipment reliability, extended lifespan, and operational efficiency.

Key Word*: Vibration; Static Balancing; centrifugal fan;*

I. Introduction

In the context of predictive maintenance, vibration analysis provides insights into the condition of rotating components by interpreting the amplitude and frequency characteristics of vibrations. It helps maintenance professionals identify and classify faults before they progress to critical levels. According to ISO 10816, vibration severity can be evaluated systematically to determine whether a machine is operating within acceptable [1]

Centrifugal fans are essential to the operation of cement manufacturing plants, where they are responsible for managing airflow, regulating kiln and mill temperatures, and supporting dust collection systems. These fans typically operate in dusty, high-load environments with rigid mounting conditions factors that increase their vulnerability to vibration-related faults. High vibration levels in such fans can lead to a cascade of mechanical failures, reduced reliability, and unplanned downtime, all of which have a significant impact on production and maintenance costs.

The motivation for this study arises from the observation that while the consequences of excessive vibration can be catastrophic, the underlying causes are often relatively simple such as rotor unbalance, loose bolts, or improper alignment. Unfortunately, these faults frequently go unnoticed or are neglected in daily maintenance routines. However, with regular vibration monitoring and attention to subtle operational changes, these issues can be identified early and resolved with straightforward corrective actions.

This study, therefore aims to demonstrate how cost-effective vibration-based monitoring techniques, coupled with manual static balancing can be employed to resolve high vibration issues in centrifugal fans. Using a case example from a cement plant, the paper documents the fault diagnosis, corrective steps, and improvements in vibration levels. The overall objective is to highlight how simple maintenance practices grounded in routine condition monitoring can significantly improve equipment performance, reduce downtime, and support sustainable industrial reliability.

Vibration monitoring is a cornerstone of predictive maintenance for rotating equipment, particularly in critical assets such as centrifugal fans. These fans are extensively used in industrial settings including cement manufacturing where continuous exposure to dust, high mechanical loads, and misalignment often leads to unbalance and subsequent mechanical failures [2, 3]

Rotor imbalance is the most common source of excessive vibration in centrifugal fans. It occurs when the mass distribution around the rotor is uneven, producing centrifugal forces that generate high vibration amplitudes typically at 1X (rotational frequency)[4]. If unaddressed, this condition accelerates bearing wear, damages couplings, and undermines structural integrity.

To correct imbalance, two primary balancing methods are applied: static balancing and dynamic balancing. These differ in procedure, precision, and application depending on the machine's design and operating environment.

Static Balancing

Static balancing involves adjusting the mass distribution of a stationary rotor so that its center of gravity aligns with the axis of rotation. It is particularly beneficial for:

i. Low-speed, rigidly mounted centrifugal fans where imbalance is primarily static[5]
ii. Situations with limited access or safety constraints, where rotating the fan is impractical,
iii. Field conditions with minimal resources, due to its simplicity and low cost.

However, static balancing has limitations:

i. It cannot correct couple or dynamic imbalance, which is more prevalent in longer or flexible rotors[6]
ii. Residual vibration may persist at higher speeds or in rotors with distributed mass asymmetry.

Dynamic Balancing

Dynamic balancing analyzes the rotor in motion and corrects both static and couple unbalances. It is ideal for:

i. Medium- to high-speed centrifugal fans and rotors with flexible shafts[7]
ii. Comprehensive balancing across multiple planes, resulting in more precise vibration correction,
iii. Extended machine life, through reduction in bearing and seal stresses.

Despite its accuracy, dynamic balancing presents several challenges:

i. It demands specialized equipment and technical expertise,
ii. Requires the rotor to be run under controlled conditions—posing safety and access concerns, especially in dusty, high-temperature environments like cement plants,
iii. Increases maintenance cost and downtime compared to simpler methods.

Industrial Considerations for Cement Plants

In cement manufacturing environments, dust, elevated temperatures, and restricted accessibility complicate the application of advanced balancing and monitoring systems [3] These factors often interfere with sensor accuracy and limit the feasibility of in-situ dynamic balancing. Under such conditions, manual static balancing guided by vibration data remains a practical and effective alternative. When executed systematically, it can significantly reduce vibration, particularly in rigid-mounted, moderate-speed centrifugal fans where imbalance is the dominant issue.

Recent studies support a hybrid diagnostic approach that combines vibration analysis, physical inspection, and operational context to improve fault detection and corrective action [8]. This methodology aligns with the present study, where fan imbalance was successfully addressed through manual static balancing, yielding a notable reduction in vibration amplitudes.

II. Material and Methods

Materials

The materials and tools utilized were selected for effective field-based static balancing of a rigid-mounted centrifugal fan operating under dusty conditions typical of a cement manufacturing plant.

Basic Tools and Consumables

The basic tools and consumables used for the analysis are:

(i) Trial Weights (5 g, 17 g, 20 g) as presented in figure 1: Used iteratively to determine the optimal mass for counteracting static unbalance.

Figure 1: Trial weights

(ii) Electrode (E6013 - Ø 3.2 mm) of measured weight 21.4 g and full length of 350 mm divided into 4 pieces with each weighing 5.35 g was used
(iii) Wire Brush: Employed to clean impeller surfaces for accurate marking and proper weight adhesion.
(iv) Metal Chalk/Marker: Used to indicate the heavy spot and demarcate positions on the impeller.
(v) Measuring Tape and Protractor: Facilitated accurate angular segmentation of the impeller (40° per section for a 9-blade fan).

Diagnostic Equipment

(i) The machine in question is a 9-blade industrial fan (centrifugal fan) running at 1475 RPM and mounted rigidly as shown in figure 2.

Figure 2: centrifugal fan

(ii) SKF Quick Collect CMDT391 Sensor: Captured overall vibration velocity in horizontal (H), vertical (V), and axial (A) directions.
(iii) Emerson CSI 2140 Vibration Analyzer: Used for fault diagnosis and confirmation of corrective results through spectrum and trend analysis.

Methodology

This section outlines the systematic approach used for diagnosing vibration anomalies, correcting mechanical faults, and conducting static balancing to reduce vibration amplitudes.

1. **Initial Vibration Measurement and Diagnosis:**
   Vibration readings were recorded at Fan Drive End (FDE) and Fan Non-Drive End (FNDE) in horizontal (H), vertical (V) and axial (A) directions. Vibration signals were evaluated using ISO 10816 criteria. The dominance of 1X (fundamental) frequency indicated unbalance, while significant axial components suggested misalignment. Fault confirmation via inspection corroborated multiple mechanical issues which includes coupling misalignment and looseness.

2. **Fault Corrections:**
   pre-balancing rectification work included aligning of impeller shaft and corrected elevation as well as tightening of all loosed ends. After the correction, a follow-up vibration measurement and analysis showed improvement but still indicated residual unbalance, warranting balancing intervention.

3. Static Balancing Procedure:

Surface Preparation with the use of wire brush to remove contaminants and buildup of materials from the impeller to ensure accurate balancing and secure placement of weights. The impeller was manually rotated and allowed to settle. The spot at 6 o'clock position was identified as the heavy spot and marked as 0° presented in figure 3. The impeller was divided into nine 40° intervals, corresponding to the 9 blades. At 12 o'clock was designated for trial weights as it is directly opposite the heavy spot. A 5 g trial weight was mounted at 12 o' clock. The fan was manually rotated to balance the high sport. Subsequently, 17 g and 20 g were tested. The 20 g weight gave the best settling behavior, effectively offsetting the heavy spot. The fan was assembled back and operated with the 20 g trial weight at 12 o'clock and vibration measured and recorded.

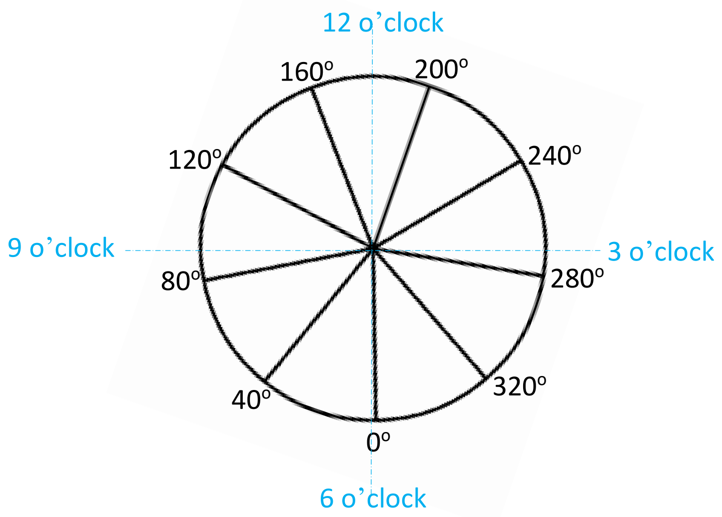

Figure 3: Impeller divisions for static balancing reference points

4. Correction Weight Calculation:

Based on system dynamics, a final correction weight was determined for effective balancing using influence coefficient method. Equation 1 was used for the calculation.

$$Correction\ weight\ (W) = \frac{T.\ A}{A-B} \quad \ldots (1)$$

Where: T = Trial weight

A = Initial vibration (before applying weight)

B = Vibration after trail weight

W = Calculated weight.

The correction weight determined was welded at 12 o'clock. Care was taken to clean the weld surface beforehand for proper adhesion. The fan housing was assembled back with all bolts torque-checked for mechanical integrity with the system returned to operational readiness. Vibration measurement was taken with both SKF Quick Collect and CSI 2140 analyzer with significant reductions of vibration observed across all points.

III. Result

This chapter presents and discusses the detailed findings from the vibration analysis, mechanical interventions, and static balancing conducted on the fan. As outlined in the methodology guided a structured diagnostic and corrective process aimed at reducing excessive vibrations to acceptable operational levels.

The primary objective of this chapter is to demonstrate how a systematic approach, combining mechanical corrections, shaft alignment, and static balancing, effectively mitigated severe vibration issues and restored the fan's performance to within ISO 10816 acceptable standards.

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1. Initial Vibration Results

At the outset of the analysis, the fan exhibited extremely high vibration levels across all four measurement points, indicative of critical operational conditions. These readings significantly exceeded ISO10816 acceptable limits for industrial machines, particularly given the rigid mounting of the fan. This severe condition necessitated an immediate and thorough mechanical inspection and alignment intervention and balancing. The vibration values were presented in table 1.

Table 1: Initial Vibration Results

Measurement point	Horizontal (mm/s)	Vertical (mm/s)	Axial (mm/s)
FDE	17.38	5.51	9.11
FNDE	15.06	9.43	6.45

These values confirmed the presence of severe unbalance, misalignment and possible structural or assembly defect as shown from the wave fronts in figure 4. The wave fronts exhibited high amplitude, predominantly at 1X frequency, which corresponds to the running speed of the fan (1475 RPM ≈ 24.6 Hz). This is a classic indicator of unbalance, as the unbalanced mass generates a centrifugal force synchronous with rotation. Additionally, axial wave

Measurement point	Horizontal (mm/s)	Vertical (mm/s)	Axial (mm/s)
FDE	17.38	5.51	9.11
FNDE	15.06	9.43	6.45

components were elevated, especially at the Fan Drive End (FDE), pointing to misalignment or shaft distortion as shown on figure 4c.

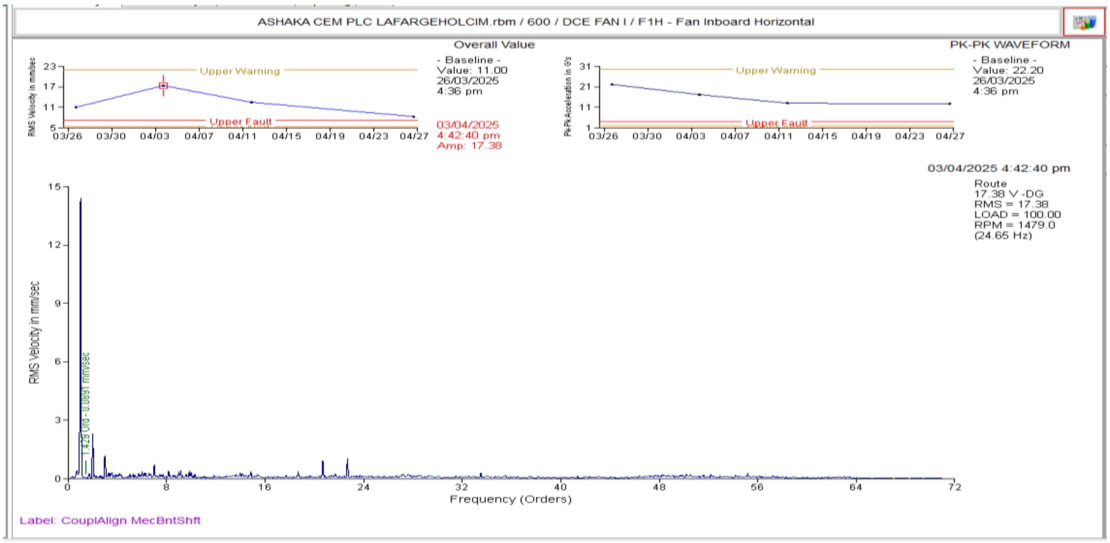


Figure 4a: Fan Drive End (FDE) – Horizontal vibration wave front

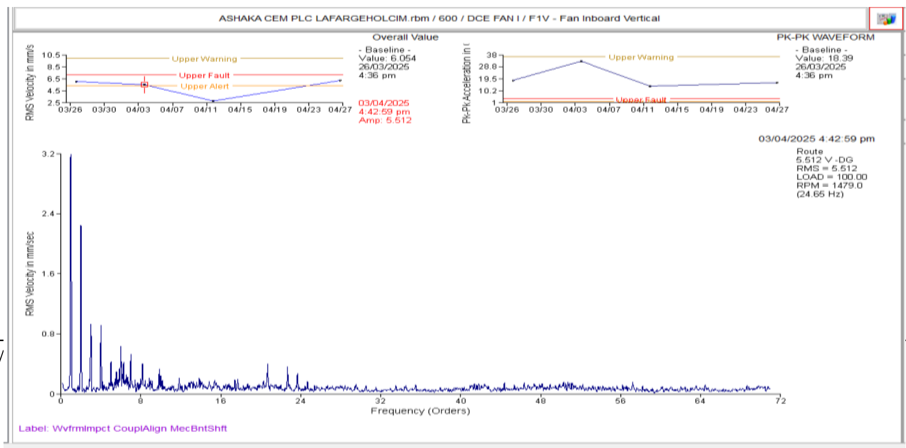


Figure 4b: Fan Drive End (FDE) – Vertical vibration wave front

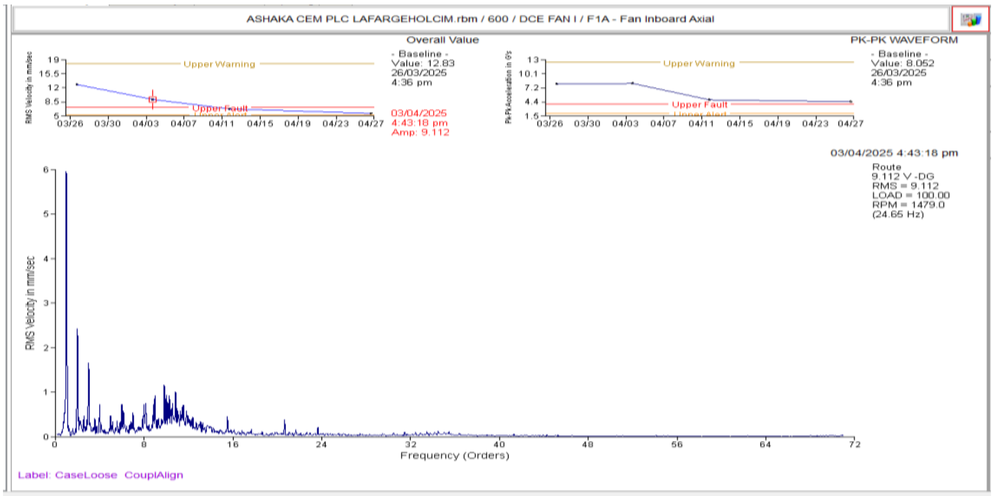


Figure 4c: Fan Drive End(FDE) – Axial vibration wave front

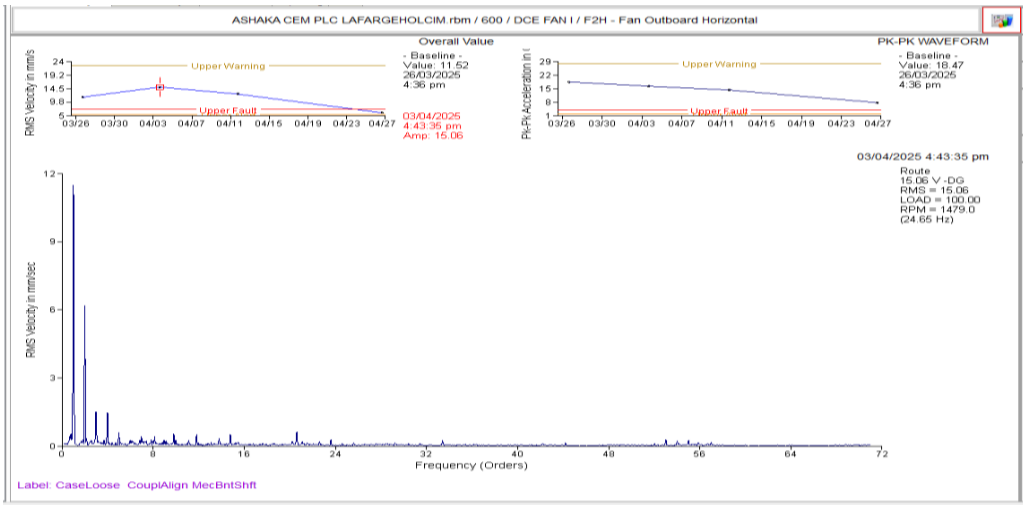


Figure 4d: Fan Non-Drive End (FNDE) – Horizontal vibration wave front

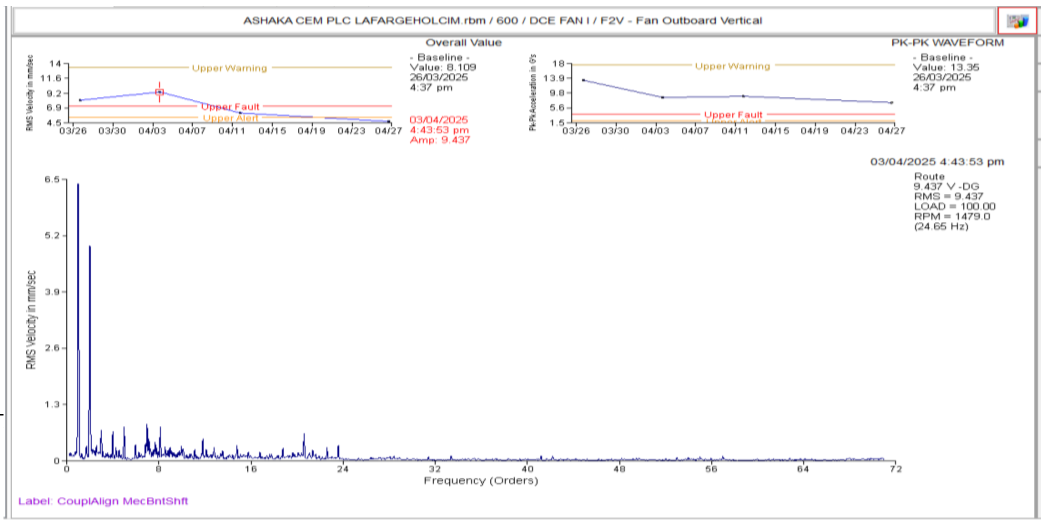


Figure 4e: Fan Non-Drive End (FNDE) – Vertical vibration wave front

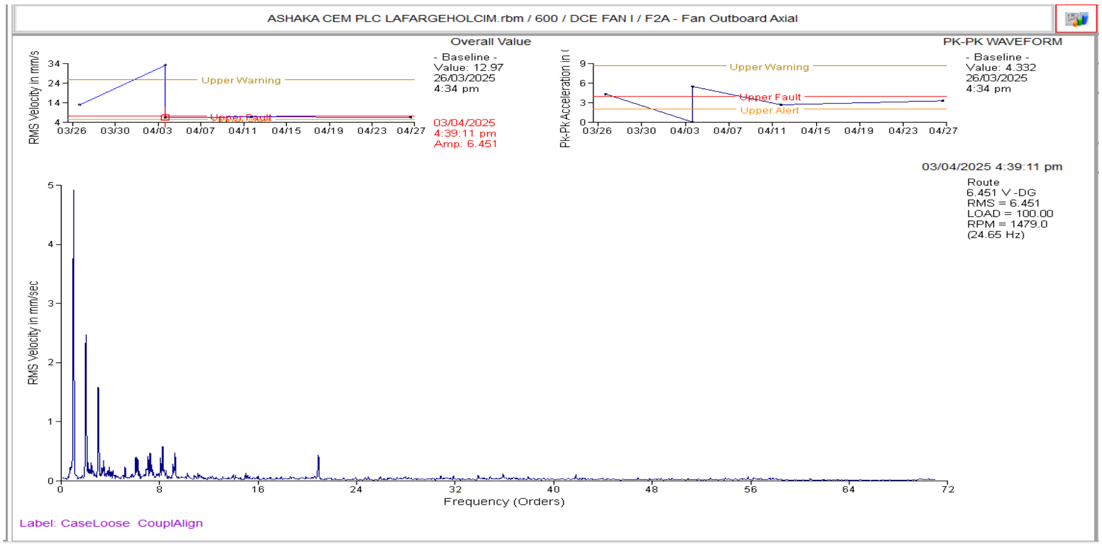


Figure 4f: Fan Non-Drive End (FNDE) – Axial vibration wave front

2. Post-Alignment and Mechanical Corrections

From the initial diagnosis, several critical mechanical faults were identified and addressed. These included an impeller shaft misalignment, which was contributing significantly to axial vibration (figure 4c), a bearing housing mismatch, and casing distortion(cracks). These issues were corrected through detailed inspection and reassembly, followed by alignment of the motor and fan shafts. The alignment process aimed to minimize angular and parallel offset between the rotating components, thereby reducing the stress on bearings and shafts. The vibration reading abtained was recorded in table 2 and the wave front in figure 5.

Table 2: Post-Alignment Vibration Readings

Measurement point	Horizontal (mm/s)	Vertical (mm/s)	Axial (mm/s)
FDE	12.48	2.79	6.75
FNDE	12.58	6.06	6.74

While the mechanical corrections and shaft alignment provided some relief, leading to a slight reduction in vibration amplitudes (e.g., horizontal vibration at FDE decreased from 17.38 mm/sec to 12.48 mm/sec, a 25.4 % reduction), the overall vibration levels remained considerably high and above ISO acceptable limits as shown on the wave fronts (figure 5a – 5f), confirming that correcting shaft misalignment was effective. However, the dominant 1X component remained, and overall amplitudes, especially in the horizontal direction, stayed high. This signaled persistent unbalance on the impeller, reinforcing the need for balancing, which the alignment process alone could not fully address.

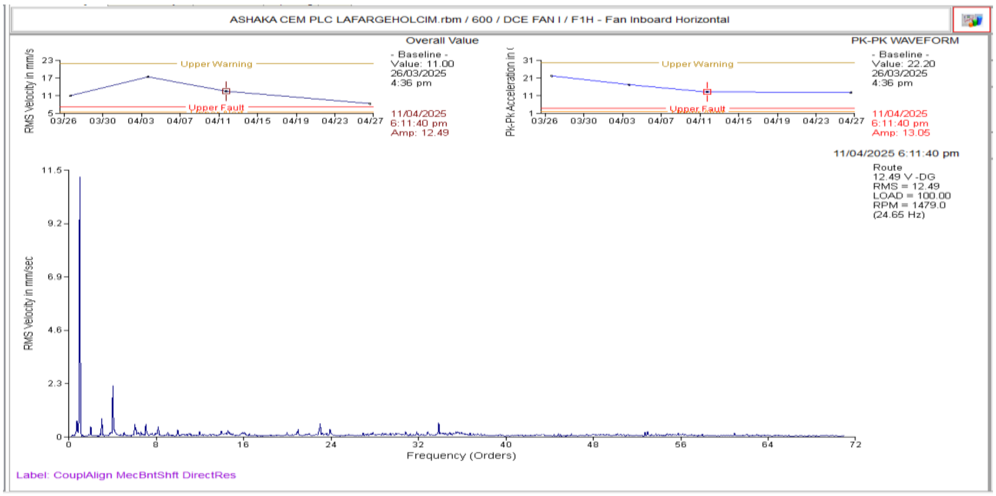


Figure 5a: Fan Drive End (FDE) – Horizontal vibration wave front

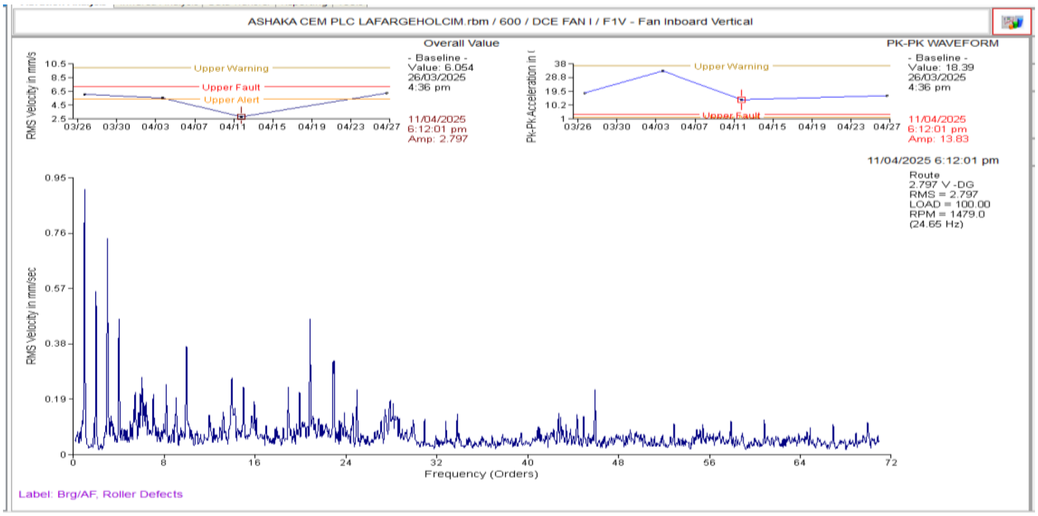


Figure 5b: Fan Drive End (FDE) – Vertical vibration wave front

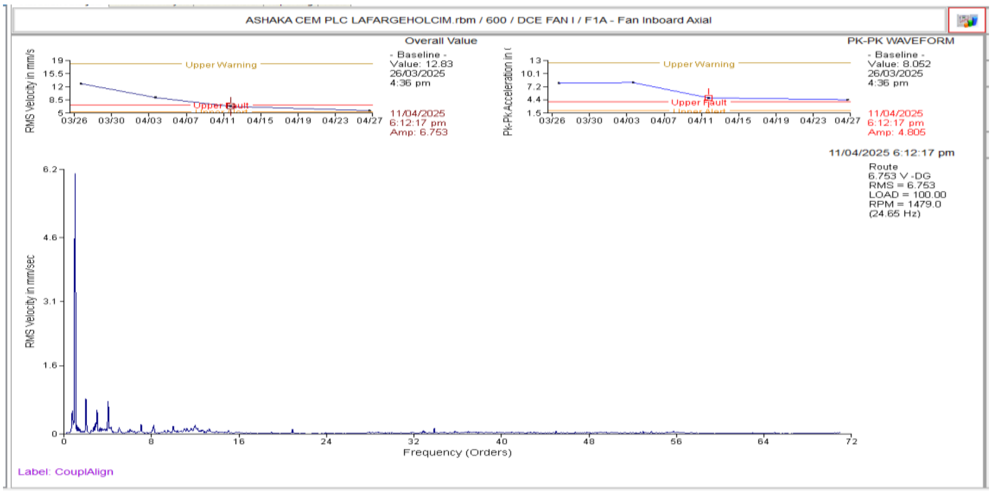


Figure 5c: Fan Drive End (FDE) – Axial vibration wave front

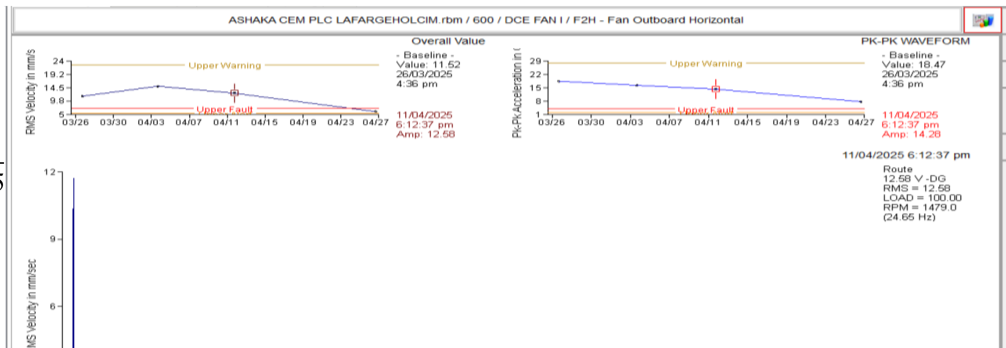


Figure 5d: Fan Non-Drive End (FNDE) – Horizontal vibration wave front

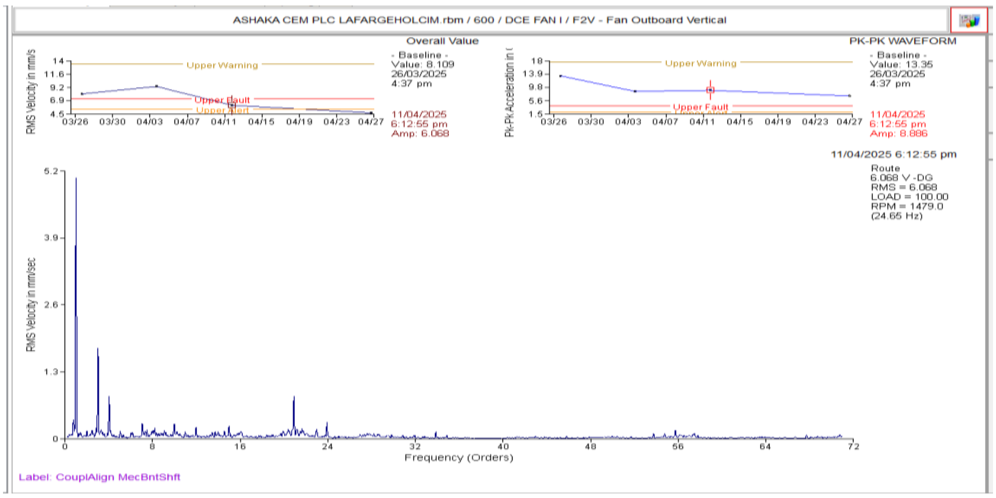


Figure 5e: Fan Non-Drive End (FNDE) – Vertical vibration wave front

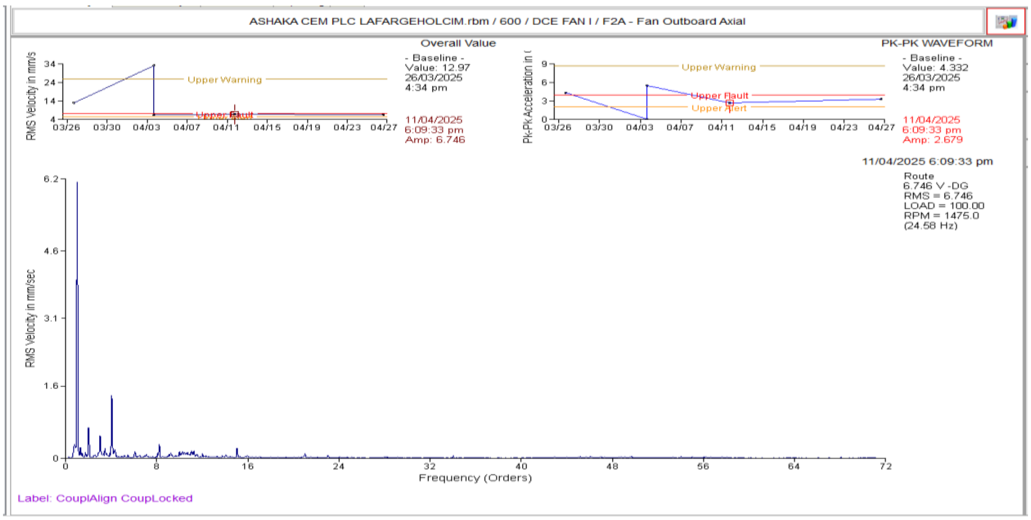


Figure 5f: Fan Non-Drive End (FNDE) – Axial vibration wave front

3. Post-Static Balancing.

Recognizing the persistent unbalance, a static balancing process was initiated following the comprehensive procedure outlined in Phase IV to reduce vibration amplitude observed at the fan. This process aims to eliminate the overall mass eccentricity of the impeller by adding or removing weight at a single plane on the impeller where the unbalance effect could be effectively neutralized. The procedure began with surface preparation and cleaning then manually rotating the impeller to determine the heavy spot which was marked as point 0° while Opposite this, was 180° identified as the light spot and therefore the most appropriate location for placing a trial counterweight.

A sequence of trial weights was tested for static correction. 5 g weight at 12 o' clock showed minimal effect. 17 g improved the settling balance but not satisfactorily. 20 g weight demonstrated the most significant correction during manual settling. The 20 g trial weight was mounted at 12 o' clock and the fan were reassembled for operational testing. Upon startup, the new vibration profile was measured across all four bearing locations using the SKF Quick Collect CMDT391 sensor and later confirmed with the CSI 2140 analyzer.

Table 3: Vibration After 20g Trial Weight

After applying the 20 g trial weight, the wave fronts showed a dramatic reduction in amplitude across all measurement points, particularly at FNDE. For instance, the horizontal vibration at FNDE was reduced from 12.58 mm/sec to 6.04 mm/sec, representing a 51 % reduction from the post-alignment state, confirming the effectiveness of the trial weight in mitigating the primary unbalance force as shown below. While the improvement was most pronounced in the horizontal plane (as expected for static unbalance), notable reductions were also observed in vertical and axial directions, indicating a more stable overall rotational system.

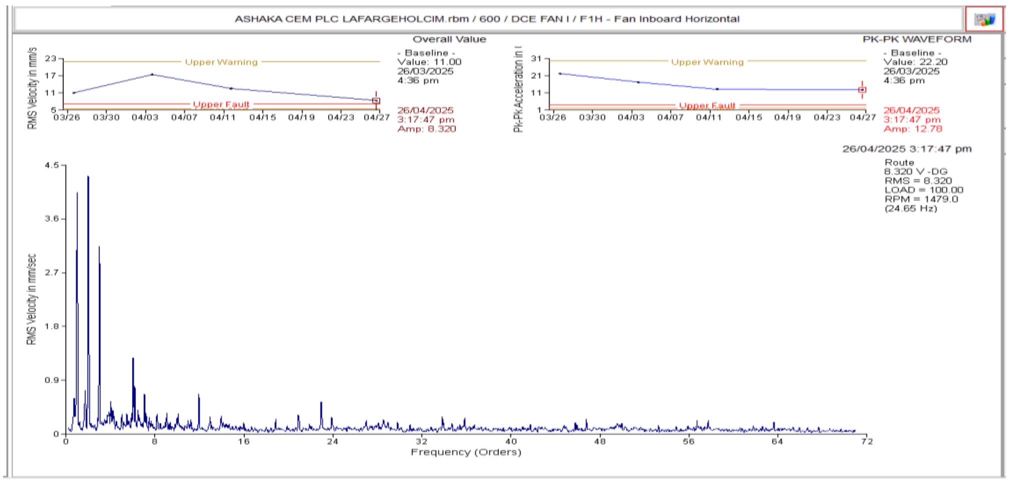


Figure 6a: Fan Drive End (FDE) – Horizontal vibration wave front

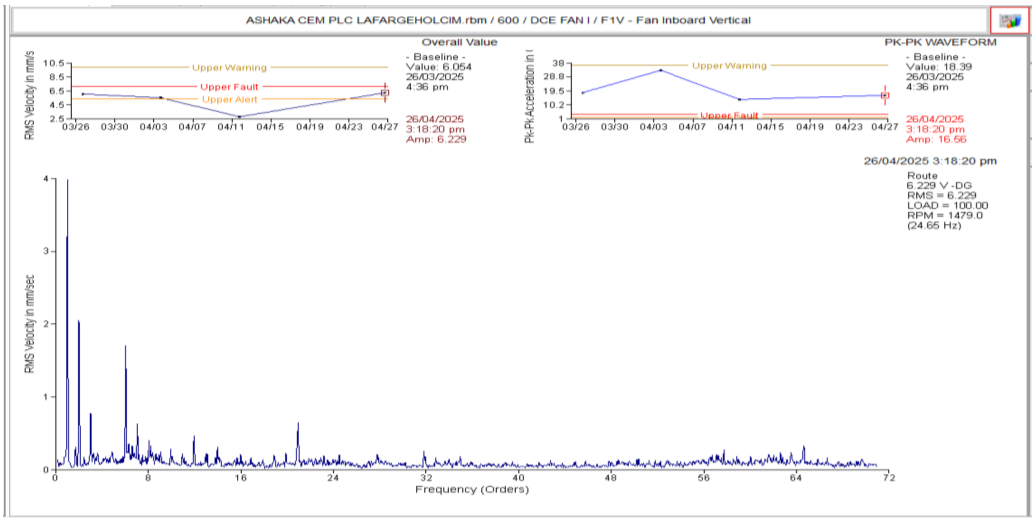
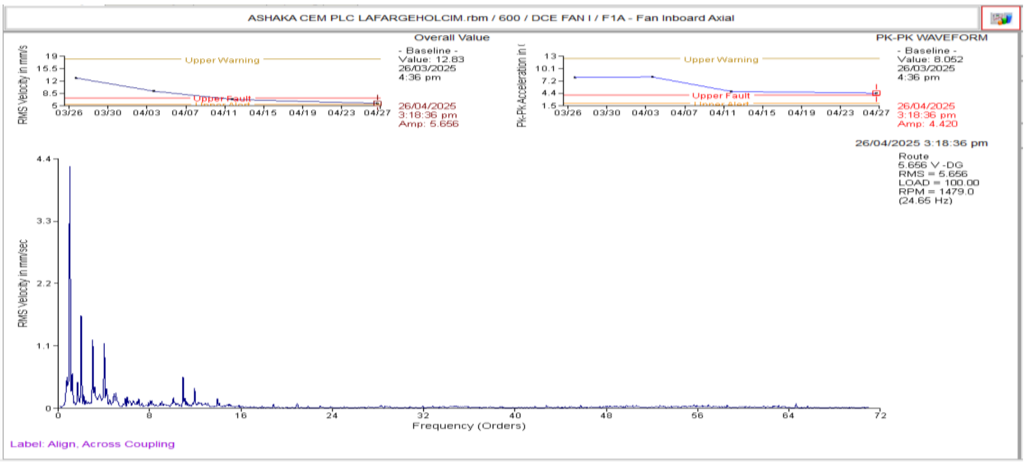


Figure 6b: Fan Drive End (FDE) – Vertical vibration wave front



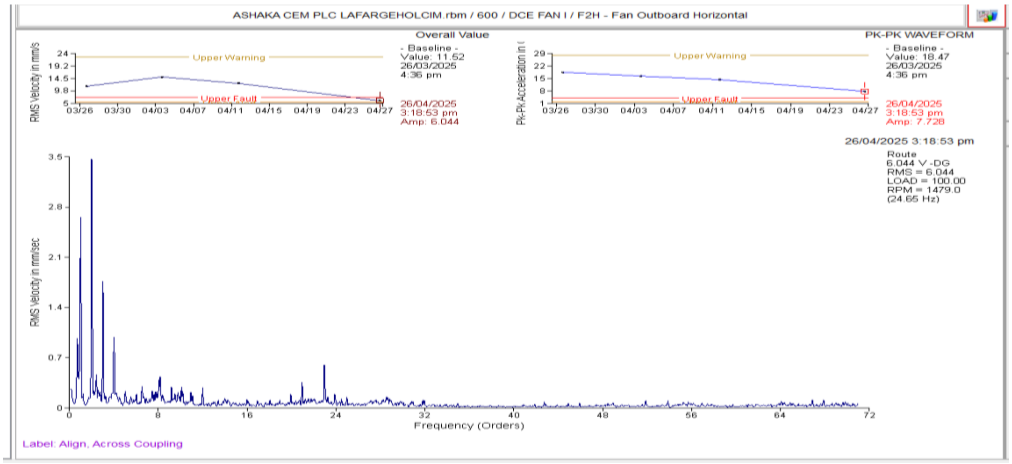


Figure 6d: Fan Non-Drive End (FNDE) – Horizontal vibration wave front

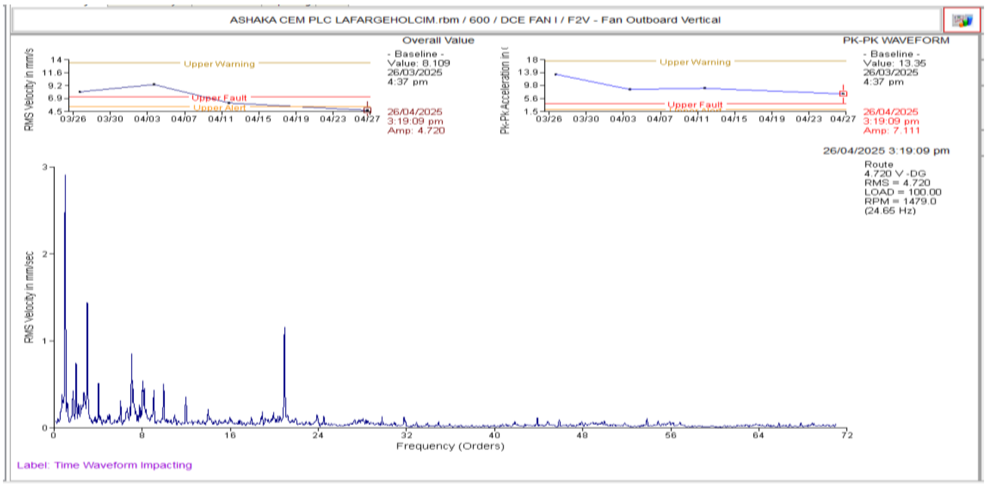


Figure 6e: Fan Non-Drive End (FNDE) – Vertical vibration wave front.

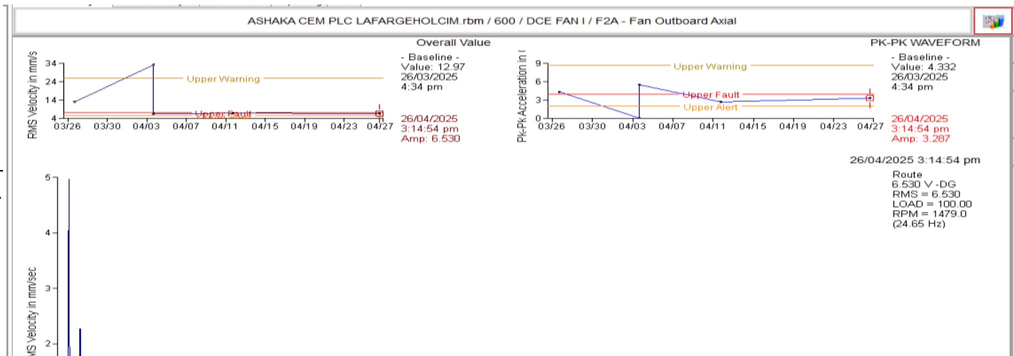


Figure 6f: Fan Non-Drive End (FNDE) – Axial vibration wave front

4. Final Balancing using the Correction Weigh

Measurement point	Horizontal (mm/s)	Vertical (mm/s)	Axial (mm/s)
FDE	3.91	3.35	3.08
FNDE	3.82	2.95	3.90

Based on the positive results from the 20 g trial weight, correction weight was determined using equation 1.

$$\text{Correction weight (W)} = \frac{\text{T. A}}{\text{A} - \text{B}}.$$
$$= \frac{20 \text{ x } 17.38}{17.38 - 7.32}$$
$$= \frac{347.6}{10.06}$$
$$= \mathbf{34.55\ g}$$

This calculated mass (the correction mass) was determined to be optimal for achieving the lowest possible residual unbalance. The 34.55 g was precisely welded at the same position (12 o'clock position). The position choice was validated by the significant drop in vibration after placing the trial weight on that same position. The vibration readings with the correction weight in position was presented in table 4.

Table 4: Final Vibration Readings

This clearly demonstrates the success of the final balancing. The vibration levels are now well within ISO 10816 and in good operating limits for this type of machinery. The final horizontal vibration at FDE is 3.91 mm/sec represents an 82% total reduction from the initial state of 17.38 mm/sec as summarized in figure 7. This indicates that the fan impeller is rotating with greatly reduced dynamic forces, leading to improved stability and a significant extension of its operational lifespan. The entire process, from initial fault identification via vibration analysis to mechanical corrections and manual static balancing, underscores the critical importance of a holistic approach to machinery maintenance. This case study effectively illustrates how systematic interventions can restore severely affected rotating equipment to optimal and safe operating conditions.

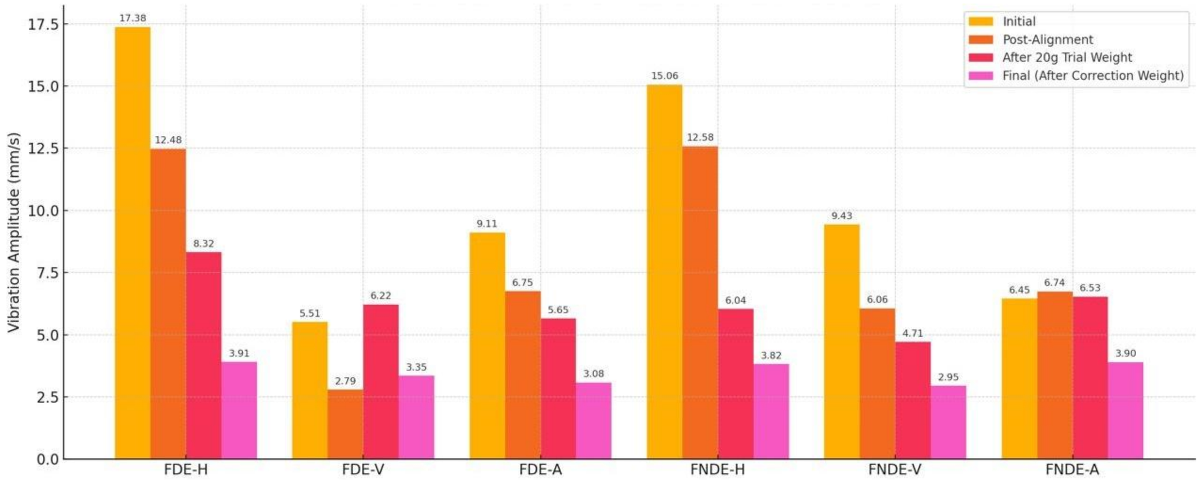


Figure 7: vibration measurement comparison before and after balancing

Initial Measurement: Vibration levels were critically high, with horizontal readings at FDE reaching 17.38 mm/s and at FNDE 15.06 mm/s—indicating severe unbalance and misalignment. Post-Alignment: Mechanical corrections such as shaft alignment and bolt tightening reduced vibration moderately, especially in axial directions. However, horizontal values remained well above ISO 10816 limits, confirming residual unbalance. After 20g Trial Weight: The application of a 20g trial weight significantly improved balance, especially at FNDE where horizontal vibration dropped to 6.04 mm/s (a 52% reduction from post-alignment). This confirmed that static unbalance was the dominant fault. Final (After Correction Weight - 34.55g): Final vibration readings showed substantial improvements across all directions and locations. The horizontal reading at FDE fell to 3.91 mm/s—a total reduction of 82% from the initial state. All values now fall within ISO 10816 acceptable limits.

IV. Conclusion

This study successfully investigated and resolved high vibration levels in a rigid-mounted centrifugal fan used in a cement plant. Initial vibration measurements revealed critical levels of unbalance and misalignment, with horizontal amplitudes as high as 17.38 mm/s at the fan drive end far exceeding ISO 10816 acceptable limits. Through thorough mechanical inspection and correction including shaft alignment, tightening of loose bolts, and addressing bearing housing mismatches vibration levels were moderately reduced. However, significant residual unbalance persisted. A structured static balancing procedure was then conducted using trial weights (5 g, 17 g, and 20 g), with the 20 g weight yielding the most stable response. Final balancing calculations led to the precise application of a 34.55 g correction weight at 12 o'clock.

This intervention resulted in a dramatic vibration reduction across all measurement points. For example, horizontal vibration at the fan drive end dropped from 17.38 mm/s to 3.91 mm/s, an 82% reduction bringing the fan's performance well within ISO compliance. This achievement demonstrates that with systematic diagnostics, proper mechanical corrections, and carefully executed static balancing, significant vibration issues can be resolved without sophisticated equipment. The case confirms the effectiveness of manual balancing methods in industrial environments where dynamic balancing may be impractical due to space, cost, or safety constraints.

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