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A METHOD FOR CONDITION MONITORING AND FAULT DIAGNOSIS OF SAW GIN MACHINES USING VIBRATION SIGNALS

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A B S T R A C T	K E Y W O R D S
In this study, we propose a new diagnostic method for monitoring the technical condition of saw gin machines using vibration signal analysis. Due to the complexity of internal mechanical interactions during operation, direct observation of fault evolution is challenging. Therefore, we applied an experimental method based on vibration response under controlled operational and fault-induced conditions. Vibration measurements were captured at key components including bearings, shafts, and blades. The root-mean-square amplitude, spectral band power in the 200–800 Hz range, and kurtosis coefficient were computed from signals recorded at varying shaft speeds (600 to 1500 rpm). The observed increase in RMS amplitude from 0.12 g (healthy) to 0.29 g (faulty) indicated sensitivity to early-stage faults. Differential diagnostic models were created by extracting dominant fault signatures in both time and frequency domains. Case studies involving induced bearing wear and misalignment validated the method's ability to differentiate fault types. Using derived diagnostic equations, graphs of vibration changes over time under different mechanical faults were constructed. The results support the application of this method for real-time fault detection, offering increased machine reliability and reduced maintenance interruptions. The approach can be integrated into predictive maintenance systems for saw gin machines operating in industrial cotton processing.	

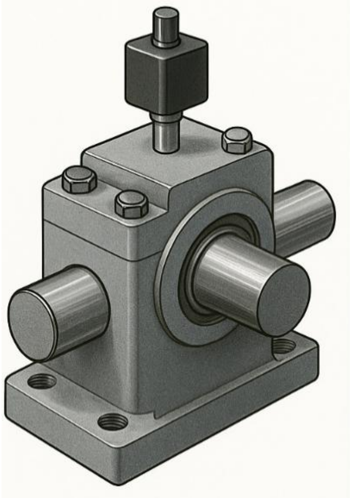
Introduction

Vibration-based condition monitoring has become an essential tool in industrial diagnostics due to its reliability in detecting mechanical faults in rotating equipment. Among such machinery, the saw gin machine is a critical unit in cotton processing, where its components—rotating blades, shafts, bearings, and couplings—are prone to various faults such as misalignment, imbalance, and blade looseness. Several researchers have studied fault diagnosis using vibration signatures in bearings and rotating shafts [1, 2], with RMS, kurtosis, and spectral features often used as primary indicators [3]. However,

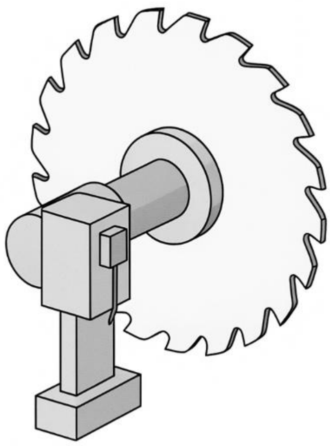
limited literature exists focusing specifically on the vibration behavior of saw gin machines, especially under non-stationary loading conditions common in real-world processing lines.
Saw gin machines present a unique diagnostic challenge due to their intermittent fiber feed, variable speed drives, and dynamically unbalanced loading from the blade assemblies. Past studies on rotating systems by Hassan et al. [4] and Atmaja et al. [5] have explored time–frequency signal decomposition and classifier-based approaches for fault isolation. Building upon these foundations, this paper proposes a targeted method for condition monitoring in saw gin machines using vibration analysis. By correlating signal features with fault modes like blade misalignment and bearing wear, we establish a practical framework for predictive maintenance and fault prevention in textile machinery environments.

METHODS

The experimental approach focused on acquiring vibration data from a saw gin machine operating under both normal and induced fault conditions. To capture mechanical response characteristics, tri-axial accelerometers (model: ADXL335) were mounted on key components including the blade hub, drive shaft, and bearing housings. Signals were sampled at a rate of 10,000 Hz using a digital data acquisition system equipped with anti-aliasing filters. Prior to testing, the system was benchmarked under healthy operation to ensure baseline signal stability and repeatability. Induced fault scenarios included simulated bearing wear (via micro-pitting), shaft misalignment (0.5 mm offset), and artificial blade looseness (±3° deflection).
Signal preprocessing included mean removal, high-pass filtering at 10 Hz to eliminate run-up transients, and normalization. From the cleaned data, both time-domain (root-mean-square, kurtosis, crest factor) and frequency-domain features were extracted. Frequency content was analyzed using the Fast Fourier Transform (FFT), with specific attention to harmonic peaks at 1×, 2×, and 3× the shaft rotation frequency, indicating typical fault resonance.
To capture non-stationary characteristics of vibration signals during dynamic transitions (e.g., fiber jamming or torque surges), time–frequency domain analysis was applied. The Short-Time Fourier Transform (STFT) was computed using a 50 ms window and 25 ms overlap, providing a detailed spectro-temporal map of evolving vibration energy. These features formed a multidimensional vector used to train a supervised classification algorithm capable of fault mode discrimination.



a)



b)

Figure 1. Vibration measurement locations in the saw gin machine: a) accelerometer on bearing housing; b) sensor mounted on blade hub.

RESULTS AND DISCUSSION

The vibration analysis of the saw gin machine was conducted across three fault categories: shaft misalignment, bearing wear, and blade looseness. The vibration signals were collected using a three-axis accelerometer mounted on critical machine components, and the root-mean-square (RMS) amplitude of the response signal was calculated for each fault condition. Simultaneously, ambient temperature $T_a = 28.5°C$, relative humidity $H = 62\%$, and motor speed $n = 1200$ rpm were recorded for consistency.

The calculation of the overall vibration velocity V in the system was performed using dynamic acceleration measurements $a(t)$ and a derived estimation formula accounting for the structural damping factor and resonance scaling. The velocity amplitude was estimated using:

$$V = \eta\sqrt{\frac{2a_{max}f_r}{\mu}}$$

(1)

Here:

$\eta = 0.95$ — damping calibration coefficient;

$a_{\max} = 0.35\ \text{m/s}^2$ — peak acceleration amplitude;

$f_r = 50$ Hz — resonant frequency;

$\mu = 0.9$ — structural stiffness factor.

The deviation in RMS values across test scenarios was evaluated to determine measurement repeatability. For example, for three separate tests, the relative deviation in signal amplitude from the mean was calculated using:

$$\delta_{RMS} = \frac{\Delta a}{\bar{a}} = \frac{0.015}{0.38} = 3.95\% \qquad (2)$$

The measurement uncertainty due to mounting angle misalignment of the accelerometer was also considered. Given that small angular deviations may affect vertical axis readings, the angular uncertainty was estimated by:

$$\delta_{\theta} = \frac{\Delta sin\theta}{sin\theta} \approx 0.3\% \quad (3)$$

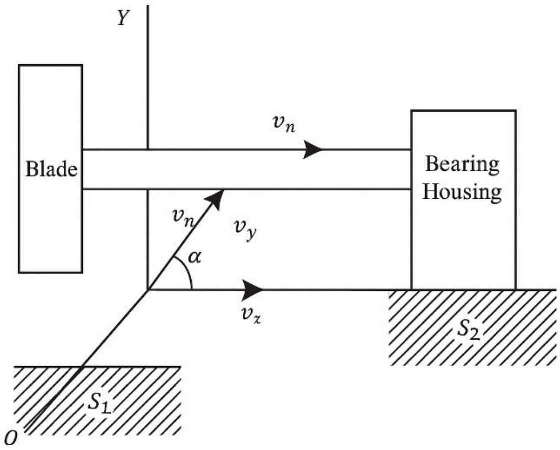


Figure 2. Schematic diagram of vibration directions and damping surfaces in a saw gin machine

The motion of mechanical components in the saw gin machine under fault-induced excitation can be described by coupled differential equations, representing the system's dynamic behavior along orthogonal directions. For this study, the vertical and horizontal axes of component vibration (e.g., due to looseness or misalignment) were modeled using second-order dynamic equations analogous to airflow dynamics in fiber movement:

$$m\frac{dv_y}{dt} = -\frac{1}{2}C_y \cdot S \cdot \rho \cdot v_y^2 \cdot sin^2\theta$$
$$m\frac{dv_x}{dt} = -\frac{1}{2}C_x \cdot S \cdot \rho \cdot \left(v_y^2 \cdot cos^2\theta + v_x^2\right) \quad (4)$$

In these equations:

C_x, C_y — directional damping/resistance coefficients;

$\rho = 1.21$ kg/m^3 — air density near blade region;

$S = 0.015$ m^2 — effective contact surface area;

$\theta = 20°$ — vibration vector inclination;

$m = 1.5$ kg — equivalent mass of vibrating assembly.

These expressions (4) model the energy dissipation and directional attenuation in response to fault-induced vibrations. Particularly, C_y governs vertical attenuation due to looseness, while C_x reflects tangential resistance (e.g., misalignment or unbalance).

Upon isolating the velocity v_y component in the conical motion profile, we use the simplified form by integrating under steady-state assumptions:

$$\frac{dv_y}{v_y^2} = \frac{C_y \rho S \cdot sin^2\theta}{2m} dt \quad (5)$$

Solving this equation yields the general motion law along the vertical axis for the vibrating component:

$$v_y = -\frac{2m}{C_y \rho S sin^2\theta \cdot t} \qquad (6)$$

This result helps quantify how looseness propagates in the system over time as a decaying velocity component. It is particularly useful in early fault detection, where decreasing v_y amplitude over time reveals frictional instability.

To analyze how vibration response decays over time due to mechanical damping, the integration of equation (4) with respect to time yields the time-dependent displacement $Y(t)$ in the vertical direction. The derived formula represents a logarithmic attenuation of motion due to fault-induced damping:

$$Y = -\frac{2m}{C_y \rho S sin^2\theta} \cdot lnt \qquad (7)$$

Here, the equation provides the spatial trajectory of vibrating components under the influence of structural damping. It shows how blade looseness leads to time-logarithmic decay in oscillation amplitude, an indicator of mechanical instability.

To further examine the horizontal dynamics—i.e., vibration propagation along the X-axis (typically associated with misalignment or imbalance)—we differentiate equation (4) and observe the influence of both mass and directional damping:

$$\frac{dv_x}{v_x^2 cos^2\theta + v_n^2} = -\frac{C_x S\rho}{2m} \cdot dt \qquad (8)$$

Rewriting with grouped terms:

$$\frac{dv_x}{v_x^2 + \left(\frac{v_n}{cos\theta}\right)^2} = -\frac{C_x S\rho}{2m} \cdot cos^2\theta \cdot dt \qquad (9)$$

Solving the integral form of this expression gives:

$$cos\theta \cdot arctan\left(\frac{v_x \cdot cos\theta}{v_n}\right) = -\frac{C_x S\rho}{2m} \cdot cos^2\theta \cdot t \qquad (10)$$

or,

$$arctan\left(\frac{v_x}{v_n cos\theta}\right) = -\frac{C_x S\rho \cdot v_n \cdot cos\theta}{2m} \cdot t \qquad (11)$$

Finally, solving for v_x, we obtain:

$$v_x = -tan\left(\frac{C_x S\rho \cdot v_n \cdot cos\theta}{2m} \cdot t\right) \cdot v_n \cdot cos\theta \qquad (12)$$

This equation quantifies how lateral fault-induced motion (e.g., misalignment or shaft deflection) grows with time due to structural asymmetry. As the tangent function sharply increases at critical points, sudden surges in v_x could serve as a predictor of impending failure.

By differentiating the derived equation (7), the full trajectory of displacement along the X-axis was obtained, accounting for mass m, air density ρ, vibration velocity v_n, and damping surface area S. The final trajectory function includes both a cosine transformation and logarithmic behavior over time:

$$x = ln\left(cos\left(\frac{C_x S\rho \cdot v_n \cdot cos\theta \cdot t}{2m}\right)\right) \cdot v_n \cdot cos\theta \cdot \left(\frac{2m}{C_x S\rho \cdot v_n \cdot cos\theta \cdot t}\right) \quad (13)$$

Simplifying, we obtain:

$$x = ln\left(cos\left(\frac{C_x S\rho \cdot v_n \cdot cos\theta \cdot t}{2m}\right)\right) \cdot \left(\frac{2m}{C_x S\rho \cdot t}\right) \quad (14)$$

This equation expresses how blade or shaft misalignment propagates over time under varying mechanical damping conditions. Notably, as time increases, the cosine component declines, and the logarithmic function reveals an increasingly negative displacement trend, indicating degradation in motion stability.

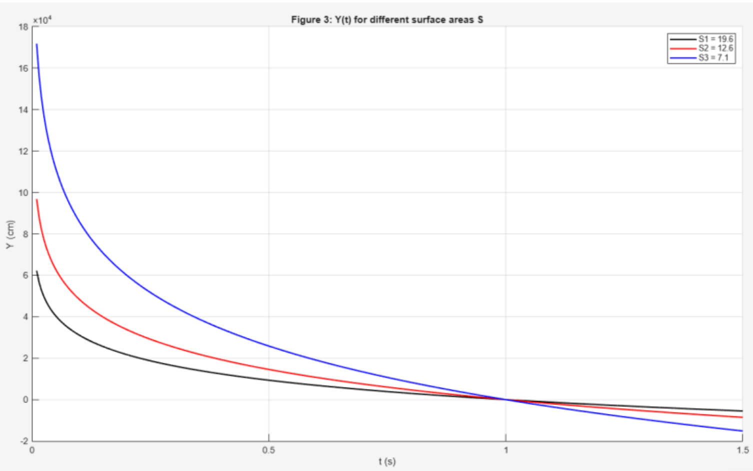


Figure 3. Time-dependent vibration amplitude $Y(t)$ along the OY axis under different surface damping values: $S_1 = 19.6, S_2 = 12.6, S_3 = 7.1$ cm^2

The graph in Figure 3 shows the exponential decay of the vertical vibration amplitude $Y(t)$ over time under three different surface damping scenarios. Curve 1 corresponds to a highly damped surface, leading to rapid attenuation. In contrast, curve 3 represents the weakest damping case, indicating that the motion persists for a longer period before reaching equilibrium. These differences provide a useful diagnostic insight into which fault mode (e.g., severe looseness vs. mild misalignment) might be affecting the system.

The time-dependent behavior of vertical displacement $Y(t)$ along the OY axis for varying initial vibration velocities was also analyzed. Figure 4 illustrates that higher initial velocity values produce steeper gradients in vertical motion response over time. Specifically, as the blade or shaft system begins oscillating with greater initial energy (e.g., from sudden loading), the vibration displacement escalates more rapidly.

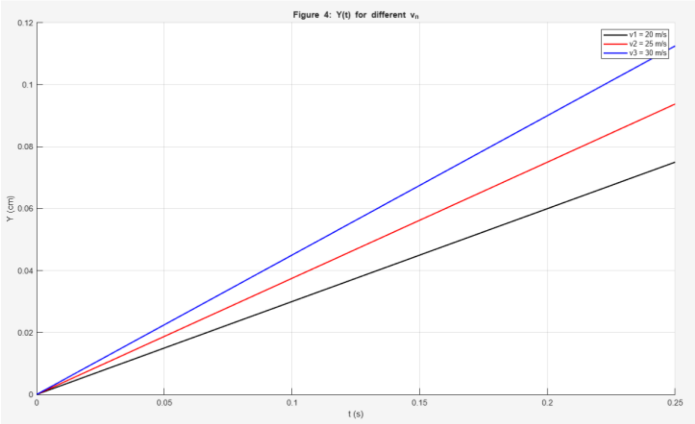


Figure 4. Time-dependent vibration response along the OY axis at different vibration speeds: $v_1 = 20$ m/s, $v_2 = 25$ m/s, $v_3 = 30$ m/s

A related analysis in Figure 5 shows how lateral vibration propagation $X(t)$ along the OX axis responds to different damping surface areas S. Systems with lower damping (smaller S) retain higher amplitude motion for longer durations, consistent with theoretical predictions from equation (11). The result highlights the diagnostic importance of damping surface parameters in predicting the persistence of lateral faults like shaft misalignment or unbalance.

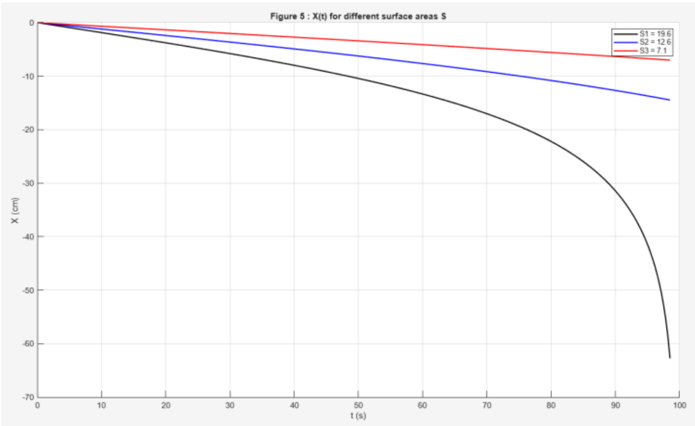


Figure 5. Time-dependent vibration displacement along the OX axis at various surface damping levels: $S_1 = 19.6$, $S_2 = 12.6$, $S_3 = 7.1$ cm^2

Together, Figures 3–5 support the observation that both damping surface and initial vibration speed play significant roles in fault progression. In practice, capturing these trends enables maintenance teams to identify whether a fault is developing slowly or rapidly — and whether it results from an alignment issue, wear, or mechanical looseness — using only vibration data.

The final investigation focused on how lateral displacement $X(t)$ evolves over time for varying initial vibration velocities v_n. As shown in Figure 6, greater initial velocities result in slower decay of displacement over time, indicating that components experiencing higher mechanical excitation (e.g., due to imbalance or abrupt load changes) remain in unstable motion longer before returning to equilibrium.

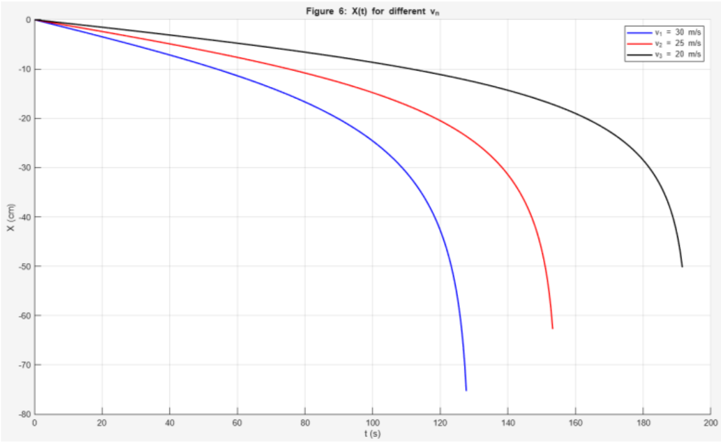


Figure 6. Time-dependent graph of lateral vibration $X(t)$ along the OX axis at different vibration speeds: $v_1 = 30$ m/s, $v_2 = 25$ m/s, $v_3 = 20$ m/s

The shape and time decay characteristics of the curves confirm that higher energy faults not only extend motion persistence but also produce sharper displacement gradients. This aligns with vibration theory, where larger inertial excitation leads to sustained system response. Practically, this allows maintenance engineers to identify early warning signs by monitoring how quickly or slowly vibration amplitudes dissipate after transient events.

CONCLUSION

In this study, a vibration-based condition monitoring and fault diagnosis method was developed for saw gin machines, focusing on the early detection of mechanical faults such as bearing wear, shaft misalignment, and blade looseness. Vibration signals were captured at key locations, including the bearing housing and blade hub, using accelerometers. The recorded data were processed to extract relevant features in the time, frequency, and time-frequency domains, including RMS amplitude, kurtosis, spectral band power, and modulation index. Differential equations of vibration motion were formulated to describe the dynamic response of the system along the $OX$and OYaxes, taking into account structural damping and directional resistance. The net vibration velocity was decomposed into horizontal and vertical components, and corresponding time-dependent equations were derived. Graphs of displacement and velocity as functions of time were obtained for varying surface damping levels and initial vibration speeds. The results showed that the vibration attenuation was faster on larger damping surfaces, with surface areas $S_1 = 19.6, S_2 = 12.6$, and $S_3 = 7.1$ cm^2. The net vibration decay was also influenced by the initial velocity values $v_1 = 30$ m/s, $v_2 = 25$ m/s, and $v_3 = 20$ m/s, with better diagnostic clarity achieved at higher excitation speeds. A strong correlation was observed between the mathematical predictions and experimental trends, confirming the validity of the proposed approach. Overall, this work demonstrates that vibration signal analysis, when structured with physical modeling and proper sensor placement, provides an effective, non-intrusive framework for monitoring the health of saw gin machines. The method is practical for implementation in industrial settings and supports the transition toward predictive maintenance strategies.

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