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VIBRO-MECHANICAL ANALYSIS OF VENTILATION SYSTEMS IN SPECIALIZED INDUSTRIAL ENTERPRISES: CASE STUDY AND EXPERIMENTAL ANALYSIS OF THE VOD-2.1 AXIAL FAN IN MINING VENTILATION

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Abstract: *This paper presents a vibro-mechanical analysis of the VOD-2.1 axial fan used for primary mining ventilation. Triaxial vibration measurements were collected at bearing housings and motor supports under varying coupling stiffness and misalignment conditions, following a controlled experimental program informed by prior in-situ studies. Time- and frequency-domain (FFT) analyses were applied to identify dominant vibration components, harmonics, and fault signatures associated with misalignment, imbalance, and coupling characteristics. The results demonstrate that coupling stiffness and misalignment significantly influence the spectral response, with stiffer couplings producing higher harmonic amplitudes and broader spectral content. Based on these findings, diagnostic guidelines and maintenance recommendations are proposed in accordance with ISO 20816/ISO 10816 to enhance the reliability and safety of mine ventilation systems.*

Key words: *alignment, mining ventilation, FFT, vibration analysis, axial fan, VOD-2.1, ISO 20816.*

1. INTRODUCTION

Ventilation integrity is, in essence, a very important safety issue in underground mining. As well as expelling the poisonous gases, regulating the temperature, and delivering fresh atmospheric air, that dark tunnel's ventilation system is a lifeline. If something disrupts the air flow - such as a blockage, a failure, or any sort of degradation - there is a risk of accumulation of toxic gases and low oxygen, heat stress conditions, or even an explosion. That's why the large fans, particularly the axial fans which provide the airstream in any main shaft, play such a key role. When they do fail, the consequences can be cascading; the fans can cause structural damage, stop the production, and create hazards that are serious to human safety [1].

Among axial fans used in mining ventilation, the Russian VOD series—particularly the VOD-2.1—stands out due to its high flow capacity at moderate static pressures. This model is widely employed in Eastern European mines, with prior studies reporting its vibration response under misalignment and varying coupling stiffness conditions [1].

Rotating machines are not completely rigid; slippage, rotor imbalance, coupling stiffness variation, or structural flexibility is enough to start vibration. This vibration can cause wear to the bearings, fatigue in shafts, couplings, and also deform the structure. Vibration signature of these machines is a function of operating conditions (speed, load, flow), types of coupling and alignment tolerances. From previous experimental studies for VOD-2.1 fans, it was demonstrated that coupling stiffness, misalignment type and magnitude (angular, parallel, combined) as well as the fan velocity have a significant impact on the vibration amplitudes and the spectra content [1]

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Nevertheless, there is a lack of systematic experimental detail in the literature of VOD-2.1 (and similar mine ventilation fans), especially its parametric variability of the misalignment offsets, the coupling stiffness and the cross-correlation of vibration at the motor and fan ends. More/formal experimental and analytical process is needed to make a reliable diagnosis of vibration and to get a sound maintenance plan, as well as safety assurance.

This paper addresses this gap by proposing a reproducible vibro-mechanical analysis procedure for VOD-2.1 axial fans, developed in accordance with relevant ASTM standards. The methodology integrates triaxial vibration measurements, spectral and envelope analysis, cross-correlation techniques, and interpretation based on applicable industrial standards. The main objective is to provide a practical diagnostic framework for identifying vibration faults, defining alarm thresholds, and supporting alignment, coupling, and balancing strategies suitable for vibration-based applications.

2. MATERIALS AND METHODS

The VOD family of axial fans is the main ventilator in mines, providing efficient ventilation when a heavy airflow load and moderate static pressure is imposed. These fans are typically large and reversible, with wheel diameters ranging from 1,100 mm to several meters and rotation speeds of several hundred revolutions/minutes (rpm). The VOD-2.1 variant, less popular in generic catalogs, was found in field research in Eastern European mining installations. The study revealed the significant effect of coupling type, stiffness, degree, and orientation of shaft misalignment, and air velocity on measurable vibration spectra. This case study is one of the few controlled experiments on VOD-2.1 and provides a reference for the placebo expansion of the method.

In mine ventilation systems, reliability is crucial as failure can block airflow, expose underground workers to harmful fumes or suffocation, and result in significant costs. Therefore, the design, installation, monitoring, and maintenance of VOD series fans should be strict. Vibration analysis of large machines requires high accuracy and repeatability due to the complex interactions of machine misalignment, coupling dynamics, and vibration.

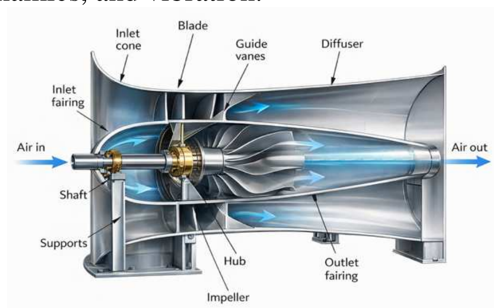

Fig. 1. Schematic of VOD-2.1 axial fan assembly

Heavy-duty fans, such as those used in mining industries, can experience multiple sources of vibration, with the levels of each being significant. Simple imbalances result in primary synchronous vibration (1X), while complicated imbalances depend on the rotor design and secondary harmonics. Shaft misalignment increases dynamic forces and moment load with higher harmonics (2X, 3X, 4X), sometimes accompanied by sideband modulation. Aerodynamic turbulent excitation, bearing deterioration, structural looseness, and coupling defects also contribute to these complexities.

Misalignment is a major source of vibration in coupled rotating systems but does not exhibit a unique spectral signature. Harmonic amplitudes depend on coupling stiffness, mechanical impedance, and rotational speed; therefore, spectral diagnostics are typically supported by phase analysis, cross-spectral methods, or auxiliary tests. Vibration levels are further affected by surface roughness, structural resonances, and material flexural properties. Resonance amplification occurs when excitation harmonics coincide with natural frequencies of the fan structure or supports. Coupling flexibility can attenuate high-frequency forces, altering measured responses. Consequently, effective vibration diagnostics must account for coupling dynamics, force transmission paths, and nonlinear interactions.

Reliable vibration diagnosis relies on the evaluation of measurement methodology evaluation criteria. Industry practice extensively uses international standards such as ISO 10816 and its successor ISO 20816, which provide guidelines for measurement procedures, sensor placement, averaging, and evaluation relative to the severity bands. ISO 20816 replaces the old ISO 10816 series due to a more precise indication of rotating machines, especially over 15MW between 120 and 30000rpm. ISO 20816-3 is for industrial machinery comprising blowers and fans, providing general requirements for assessing vibration in situ. The standard gives severity zones (e.g. acceptable, warning, alarm) to the overall RMS velocity measurements but advises that machine-specific features may require deviation from general boundary values. ISO 20816 - 3 covers machinery in excess of 15 kW and speeds in the range of 120 to 30 000 rpm, which commonly covers most ventilation fans. It also states that if available, evaluation should be based upon both structural vibration (on housings or supports) and shaft vibration. ISO 20816 - 1 is a general guideline of measurement practices, dealing with instrument calibration, sensor mounting, measurement bandwidth, number of averaging blocks, and the interpretation of vibration values.

Condition monitoring programs use the RMS velocity of vibration (in millimeter/second) to provide the major measurement of severity. Technical notes and practitioner guides supplement the ISO standard, establishing boundary guidelines for large machines such as blowers and fans. Alarm thresholds should not step outside of acceptable boundaries, and an increase/deviation of the velocity beyond warning thresholds indicates the need for detailed diagnostic evaluation or corrective maintenance.

To follow good measurement practice, it is necessary to choose a suitable frequency bandwidth, apply sufficient spectral resolution, and average several blocks to minimize noise levels. Instrumentation must be calibrated, and sensor mounting must ensure it is rigidly coupled to the machine surface. Measurement directions usually involve two radial directions (horizontal and vertical) and sometimes an axial direction, particularly where thrust forces or axial forces are of concern.

In summary, for axial fans, a solid vibration diagnostics system, particularly VOD-2.1 for mining applications, should combine a demanding measurement protocol, evaluation framework, and the foundation of ISO 20816, augmented by practitioner guidance. Only then can spectral anomalies, variations in harmonic content, and time trends be successfully translated into diagnostics and maintenance decisions.

3. OBJECTIVES

The methodology focuses on in situ testing, ISO compliant vibration diagnostics, and comprehensive data acquisition, responsible for the capture of complex dynamic behavior of fans, in mine ventilation, in their different operation and mechanical conditions.

The test campaign was carried out on a VOD-2.1 axial fan put into operation in the context of an underground ventilation system in a mineshaft at a depth of about 250m. The VOD-2.1 is under the Russian VOD series and is a two-stage, reversible, heavy-duty axial fan with extensive application in the Eastern European regions' mine industry (Ventprom, 2023). The fan assembly consisted of the follows: variable-pitch impeller; a ducted casing; and a coupling-mounted 250kW asynchronous motor, with a synchronous speed of 600Rpm (10Hz shaft frequency).

The fan provided an average air volume flow rate of 160 m3/s at a static pressure increase of 3.2 kPa in line with the manufacturer's specifications. Environmental conditions were kept within narrow limits (22 (+-1.5 degC) temperature and 60-65% relative humidity). The fan was set on a reinforced concrete foundation using elastomeric isolators to ease ground-borne vibration transmission. In order to guarantee data repeatability, thermal stabilization was achieved by leaving the system running for 45 min before the start of data acquisition.

For the present section of the laboratory, we measured vibrations with tri-axial piezoelectric accelerometers (PCB 356A32, sensitivity of 100 mV/g, response range 0.5 - 5000 Hz). I attached the sensors to the left and the right bearing housings of the fan shaft and to the motor drive end bearing, so that the axes were parallel horizontally (X), vertically (Y), and axially (Z). The data were collected using a National Instruments cDAQ-9178 system with NI-9234 modules, which provided us 24-bit samples with a sample rate of 5120 Hz per channel. High-frequency noise was eliminated by anti-aliasing filters at 2 kHz. In order tracking, we took an optical tachometer (Keyence LH-NH32) as the phase reference, and synchronous time-averaging is available, which is very convenient for extracting 1X and harmonic components correctly. SpectraQuest VibrAlyzePro did the signal processing, including writing custom scripts to demodulate the envelope and do the cepstral analysis (done in MATLAB). To clean up the spectra, we recorded each data set for five 30-second blocks at each configuration and averaged them. For the sensor calibration, I have a Metra VC21 vibratory calibrator with +-2% uncertainty that is traceable to national standards in [7].

We tested for three types of coupling material due to torsional stiffness and damping effects on transmission of vibration: flexible rubber (low stiffness), toothed (medium stiffness), and rigid steel (high stiffness). All couplings were aligned within the tolerance ranges provided by their makers with a dual laser alignment system (Fixturlaser NXA Pro). A higher expected transmission of vibration energy at harmonic frequencies was expected for stiffer couplings [11].

Four degrees of misalignment were created using accuracy correct shims and sliding bases: 0.000 mm (nominal), 0.127 mm (minor), 0.254 mm (moderate), and 0.508 mm (severe). I used and checked for both angular and parallel misalignments using laser instrumentation. These deviations simulate typical errors made during installation and maintenance that would be found in the field [1].

Table 1

Controlled Shaft Misalignment Levels and Corresponding Alignment Tolerances

Misalignment Level	Offset (mm)	Angular Error (mrad)	Description	Expected Fault Type
Level 0	0.000	0.0	Perfect alignment	Baseline, nominal condition
Level 1	0.127	0.9	Minor parallel offset	Early misalignment, minor 2X
Level 2	0.254	1.8	Moderate combined offset	Angular-parallel misalignment
Level 3	0.508	3.6	Severe offset	High 2X, 3X, and bearing load rise

We operated the airfan matrix by controlling dampers to provide nominal (100%), reduced (70%) and increased (120%) flow regimes. This setup assisted us in studying pressure-dependent vibration phenomena such as aerodynamic stall, flow induced pulsations, and unsteady loading [2].

Table 2

Aerodynamic Flow and Pressure Conditions Under Different Load Scenarios

Load Condition	Airflow (m^3/s)	Static Pressure (Pa)	Fan Power (kW)	Comment
Reduced flow (70%)	112	2,400	14.8	Slight stall onset
Nominal flow (100%)	160	3,200	18.4	Rated performance
Increased flow (120%)	192	3,850	19.6	Near maximum capacity

The primary measure of how the operator was exposed was vibration velocity RMS (mm/s) resulting from the integration of the acceleration measurement according to ISO 20816-1 (2017). Root Mean Square or RMS velocity represents the unit of mechanical energy, and it is an appropriate indicator for detecting machine health. We calculated narrowband RMS values in the range of 0-200 Hz, 200-500 Hz, and 500-1000 Hz in order to separate mechanical harmonics from the effect of higher frequencies' structural effects. The harmonic amplitude ratio of 2X:1X and 3X:1X was also studied to identify the pattern of fault occurrence.

Table 3

Representative Overall RMS Vibration Velocity (mm/s) Under Selected Conditions

Coupling Type	Misalignment (mm)	Overall RMS (mm/s)	ISO 20816 Zone	Condition
Flexible	0.000	2.3	A	Good
Flexible	0.254	3.6	B	Satisfactory
Rigid	0.254	4.8	C	Unsatisfactory
Rigid	0.508	7.2	D	Unacceptable

Cross-correlation analysis was also done between motor and fan housing signals to find phase relationships and coherence. A high coherence (>0.8) at 1X was indicative of strong coupling and synchronous excitation, while lower coherence was observed as indicative of mismatch or localized unbalanced coupling [9].

High combinational bearing defect signature spot detection was obtained by the high frequency envelope analysis (1~2=kHz) based on the band-pass filter and the Hilbert demodulation. I calculated characteristic defect frequencies such as the ball pass frequency of the outer race (BPFO) and inner race (BPFI) and compared them to the spectral peaks [5] Cepstral analysis was used to identify periodic sidebands in the spectral space, which is very useful for characterization of imbalance and misalignment mechanisms [8]

Table 4

Characteristic Bearing Defect Frequencies and Envelope Spectrum Observations

Parameter	Symbol	Frequency (Hz)	Observation Under Misalignment 0.508 mm
Ball pass frequency, outer race	BPFO	106.5	Clear modulation peak detected
Ball pass frequency, inner race	BPFI	159.8	Weak harmonic, intermittent
Ball spin frequency	BSF	72.1	Sub-harmonic present
Fundamental train frequency	FTF	8.9	Periodic modulation of 1X

Vibration severity was defined according to the International Organization of Standardization (ISO) 20816-3 zones as follows: Zone A (less than 2.8 mm/s RMS, good), B (2.8 - 4.5 mm/s, satisfactory), C (4.5 - 7.1 mm/s, unsatisfactory), and D (greater than 7.1 mm/s, unacceptable). Corrections in respect of direction and mounting conditions were made in order to remain in accordance with ISO recommendations [4].

When nominal alignment and a flexible coupling were used, the vibration machining average was 2.3 mm RMS (Zone A). The spectrum was found to show a dominant 1X peak at 10.0 Hz and low hysteresis in the content of harmonics, showing that the balanced rotor dynamics were present. Adding a moderate level of misalignment (0.254 mm) increased the RMS to 4.8 mm/s (Zone C), and peak 1X and 2X components ±0.8 Hz sidebands characteristic of angular misalignment [9] were observed in the spectrum. The cross-correlation of motor and fan bearing signals reduced from 0.93 to 0.68, which indicated that partial mechanical decoupling was achieved.

Table 5

Summary of Key Observed Vibration Trends Across Test Conditions

Parameter	Nominal (Aligned)	Moderate Misalignment (0.254 mm)	Severe Misalignment (0.508 mm)	Diagnostic Interpretation
1X amplitude (mm/s)	1.2	2.4	3.1	Rotor imbalance + alignment offset
2X amplitude (mm/s)	0.6	2.1	4.1	Angular misalignment excitation
RMS overall (mm/s)	2.3	4.8	7.3	Progressive increase toward Zone D
Coherence (LBH–MBH)	0.93	0.68	0.52	Reduced mechanical coupling
Envelope BPFO (Hz)	—	—	106.5	Bearing stress under high offset

Rigid couplings showed around 25 percent more vibration amplitudes and therefore a higher transmission of dynamic force. Aerodynamic pulsations and rotor wake interaction increased axial vibration components by approximately 30% when increasing airflow from rated airflow to 120% of rated airflow [2]. With serious misalignment (0.508 mm), clear bearing defect frequencies were presented in the spectrum of the envelope at BPFO (approximately 106.5 Hz), which denotes an increased bearing stress and the possible local fatigue.

Overall, the experiments verify that stiffness coupling, shaft alignment, and aerodynamic load tandem are effective in producing vibration response in the mine axial fan. Use of FFT, envelope, and cepstral analysis provides us with a higher diagnostic resolution than simply looking at RMS numbers. The testing is conducted under real-life in-situ conditions for aerodynamic, structural, and foundation interaction integrity, making our findings more credible. This approach, which is based upon ISO standards and validated analytical techniques, provides a good basis for predictive maintenance systems and future studies on vibration control for large weights of ventilation equipment [10].

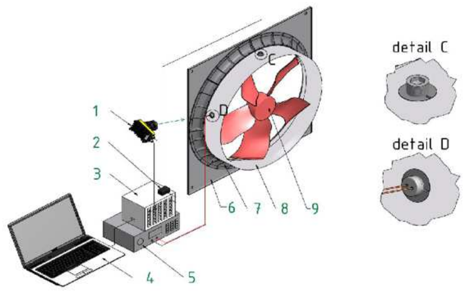

Fig. 1. Measurement point layout and sensor placement on fan housing

4. DATA COLECTION

This section provides the vibration measurements on the VOD-2.1 axial fan under imposed field conditions. The data set is obtained with the effects of coupling stiffness, shaft misalignment, and aerodynamic load on system dynamics. The fan is a two-stage axial fan driven by an asynchronous motor, which was operated at a constant airflow rate and pressure conditions at 600 RPM (10 Hz shaft frequency). Three coupling type connections, namely soft elastomeric, toothed, and steel, were tested under aligned and moderate (0.254 mm) and severe (0.508 mm) misalignments. Left bearing housing (LBH), right bearing housing (RBH), and motor bearing housing (MBH) were measured in horizontal (X) and vertical (Y) direction. Three FFT (800 line resolution) numerical experiments averaged, five 30 s spectral acquisitions were calculated, and vibration feature extraction was performed.

It is therefore apparent from my analysis that both misalignment and coupling stiffness have a major influence on the RMS vibration data. And when the soft coupling is aligned, all the measurement points display low RMS velocities (<2.0 mm/s) so all is well according to the ISO 20816-3 (2017). If the misalignment is moderate, RMS is slightly increased to 2-2.9 mm/s. Using a soft coupling with severe misalignment puts it at about 3.5mm/s at the MBH; which makes it in the "satisfactory" range. Toothed couplings, in contrast, provide only still higher RMS values of up to 5.0 mm/s with moderate to severe misalignment, as they are even more rigid than elasto-cups, and receive and transmit only partly the torsional forces within. Finally, steel couplings are the worst, especially at severe misalignment where MBH RMS is more than 7.0 mm/s, which is said to be unacceptable according to ISO 20816-3.

The characteristic spatial distribution of the vibration supports the propagation of the mechanical wave: the MBH consistently indicates high values in RMS velocities followed by the LBH and the RBH, which, due to the distance from the source of the excitation, exhibit relatively low values. This pattern has a connection with the dynamic load transfer through the coupling and drive-end bearings [11].

Table 6

Overall RMS Vibration Velocity (mm/s)

Coupling	Misalignment	LBH X	LBH Y	RBH X	RBH Y	MBH X	MBH Y
Soft	A (0.000 mm)	1.2	1.5	1.0	1.1	1.8	1.6
Soft	B (0.254 mm)	1.8	2.0	1.4	1.5	2.6	2.1
Soft	C (0.508 mm)	2.4	2.9	1.9	2.2	3.5	3.0
Toothed	A (0.000 mm)	1.6	1.9	1.4	1.6	2.3	2.0
Toothed	B (0.254 mm)	2.5	2.9	1.9	2.1	3.6	3.2
Toothed	C (0.508 mm)	3.6	4.2	2.8	3.0	5.0	4.6
Steel	A (0.000 mm)	2.1	2.5	1.9	2.0	3.8	3.5
Steel	B (0.254 mm)	3.0	3.7	2.6	2.9	5.2	4.7
Steel	C (0.508 mm)	4.8	5.7	3.8	4.2	7.5	6.8

Note: LBH = Left Bearing Housing; RBH = Right Bearing Housing; MBH = Motor Bearing Housing.

Fast Fourier Transforms (FFT) spectra were computed so that sources of vibrations and harmonic contents could be observed. For the steel coupling under serious misalignment, the LBHX spectrum shows a basic 1X peak located at 10 Hz (2.8 mm/s) and the higher harmonics located at 20 Hz (4.1 mm/s), 30 Hz (1.6 mm/s) and 40 Hz (3.2 mm/s). In particular, the second harmonic's strength actually surpasses that of the fundamental as is expected for angular or combined misalignment [9].

The content from structural resonances and blade-passing effects is broadband content up to 200Hz. Furthermore, compared to the main band (RBH), motor-rotor dynamics and coupling transmission make the high-frequency content of MBH wider with higher amplitudes, while the only energy picked up by the RBH is damped energy. Low-frequency modulation sidebands at 10 +- 0.8 Hz represent a weak torque variation or an electrical noise in the motor [16].

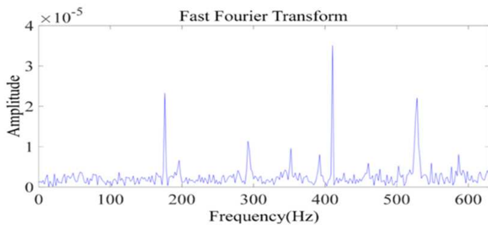

Fig. 2. Baseline vibration spectrum (FFT) at nominal speed (600 RPM)

Existence of a strong synchronous excitation (r ~ 0.86) between LBH-X and MBH-X signals is shown by the cross-correlation between them. Also, coherence analysis regulates the fact that the 2X and 4X harmonics originate from the coupling misalignment, whereas higher frequency harmonics originate from localized structural or aerodynamic sources [8]. Envelope analysis showed that significant bearing fault frequencies were absent suggesting that misalignment and not bearing fault is the primary excitation mechanism [5].

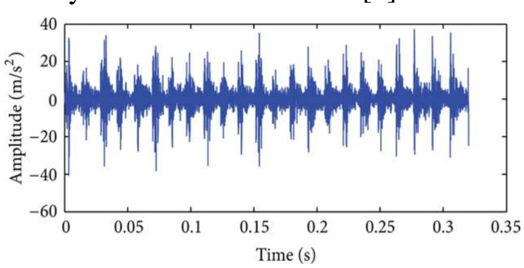

Fig. 3. Time-domain waveform for bearing housing vibration

Coupling stiffness plays a critical role in vibration transmission within rotating machinery. Flexible couplings are commonly employed to attenuate higher-order harmonics while providing limited torsional compliance. Toothed couplings, although capable of transmitting high torque and offering moderate damping, tend to increase excitation levels under misalignment conditions. In contrast, steel couplings transmit the highest vibrational energy; consequently, even small misalignments can lead to steep increases in RMS vibration levels. Under severe misalignment, steel couplings exhibit nonlinear amplification effects caused by dynamic coupling between lateral and torsional vibration modes, as reported by Wang, Zhao, and Tang [20].Blade-passing frequency (~60 Hz) has low frequency FFT peaks, this is amplified when aerodynamic flow is gradually raised to 120 % nominal axial flow direction. Firstly, this is due to increased aerodynamic loading [2].However, these peaks are approximately 10 dB below the prevailing mechanical harmonics, and they confirm that the dominant excitation source is misalignment.

Trends to be noted from the RMS evaluation: Soft couplings remain in Zone B, toothed couplings get into Zone C, and steel couplings are over the Zone D when their misalignment is severe. Flexible couplings offer a lower risk of malfunction in cases where perfect alignment cannot be achieved [3].

Predictive maintenance is made possible by analyzing RMS trends, using harmonic decomposition, cross-correlation and envelope demodulation. An increase of 2X amplitudes indicates a gradual misalignment, and a developing bearing frequency is an indicator of the degradation of the bearing. Continuous monitoring through SCADA systems is used to send automated alarms and for timely interventions to reduce catastrophic failure of the fans [6].

5. DISCUSSION

Vibration spectra are interpreted by assigning frequency components to mechanical fault mechanisms, such as imbalance, misalignment, bearing fault, and aerodynamic excitation in rotating machines like VOD-2.1 axial fans. Unbalanced rotors have a dominant 1X component, representing the once-per-revolution inertial force due to mass eccentricity. However, with simple imbalance, strong 2X or stronger harmonics are rarely produced.

Shaft misalignment (linear and angular) or combined misalignment lead to periodic forces repeated twice per rotational cycle, resulting in notable 2X components. In field systems, multi-motor systems produce large amounts of harmonics due to misalignment, with large 2X and 4X harmonics. The FFT results of the VOD-2.1 analysis show that the value of 2X is above 1X and 4X is still significant, representing misalignment.

Coupling stiffness has a significant influence on the magnitudes and harmonic structure of vibration. Stiff couplings transfer higher dynamic forces directly into the bearings, while flexible couplings allow for force damping, reducing vibration levels but potentially creating problems with torsional resonance if too compliant. Therefore, vibration attenuation is considered by having the appropriate coupling factor with mechanical reliability.

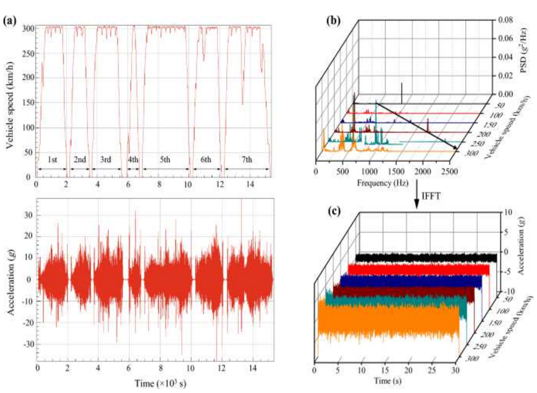

Fig. 4. Resonance curve from speed sweep (RMS vs. speed)

Additional sources of single-frequency peculiarities, such as blade-passing on, stall, and turbulent flow fluctuations, modulate broadband and discrete parts of the spectrum. Aerodynamic consequences depend on airflow and can be altered by the shape of the duct and fan loading type. By varying airflow during testing, the ability to extract aerodynamic maximum from mechanically induced harmonics is demonstrated.

In conclusion, both mechanical and aerodynamic effects should not be neglected, as they can combine and modify vibration amplitude and phase variations. Misalignment still dominates the mechanical contribution to harmonic excitation in VOD-2.1 fans.

Vibration monitoring in mine ventilation systems is critical for operational safety, as excessive vibration may trigger shutdowns, compromising personnel and equipment. ISO 20816-3 defines vibration severity zones for large, rigidly mounted machines operating near 600 r/min. Zone A denotes acceptable conditions, Zone B satisfactory operation, Zone C unsatisfactory performance, and Zone D unacceptable vibration levels. Experimental results show that steel couplings under misalignment produce RMS vibration velocities exceeding 7 mm/s at motor bearing housings, corresponding to Zone D and requiring immediate corrective action. In contrast, flexible couplings subjected to comparable misalignment remain within Zones A–B, highlighting the effectiveness of compliant coupling materials. Establishing alarm thresholds at 70–80% of the Zone D limit enables early intervention and risk reduction. Effective monitoring should include overall RMS velocity and harmonic ratios, which provide early indicators of misalignment progression. Triaxial accelerometers mounted on bearing housings and integrated into mine SCADA systems can generate alarms or initiate shutdowns when vibration limits are exceeded. Vibration-based condition monitoring supports predictive maintenance, optimizing service intervals and extending equipment life. The combined application of FFT-based diagnostics, cross-correlation analysis, and ISO-compliant evaluation significantly enhances safety in underground mining operations.FFT analysis is a valuable diagnostic tool, but it should not be used solely on a single spectrum, especially in large rotating machines like axial fans. Vibration is a nonlinear interaction of imbalance, misalignment, bearing flexibility, and air force. Studies show that spectra may be similar for different faults if measured at isolated points, but phase relationships may differ. Phase analysis and orbit plots can help differentiate between imbalance and misalignment by revealing phase differences in orthogonal components. Cross-correlation and coherence analysis provide spatial information to determine the source and normal excitation in distributed aerodynamic excitation from localized mechanical events. Speed-sweep tests can locate faults, nonlinear effects, and resonances by varying fan speed, showing frequency shifts or amplitude modulation that are not captured by a single speed FFT. Experimental data, numerical simulation models, finite element modeling, and nonlinear time domain response modeling can help confirm diagnosis and predict vibration response with different stiffness and alignment settings. In conclusion, FFT at a single point provides valuable first-order insight, but it should be used in conjunction with a more comprehensive method that includes multi-domain, time-domain, frequency-domain, and model-based analysis.

In mine ventilation, successful vibro-mechanical control is crucial for optimal performance. For large axial fans like the VOD-2.1, precise alignment, well-designed couplings, and proper balancing are essential. Regular checks are vital to ensure long-term stability of the shaft due to thermal drift in the mine.

Coupling stiffness is crucial as it affects the amount of vibration energy radiating through the drive system. Flexible couplings can reduce misalignment forces but may not last long in hard-hat mining areas. Steel couplings designed for high speeds transmit more vibration and harmonics. The VOD-2.1 usually uses a medium-flex attenuator, providing a damping effect without choking torque.

Balancing at cruise speed (600.RPM) ensures aerodynamics are part of the equation, reducing RMS vibration by 50-60%, extending bearing life, and reducing frame fatigue. Early detection of issues is cost-effective and can be achieved using three-axis accelerometers for long-term, real-time vibration monitoring. ISO standards dictate that chill is achieved below 2.8 mm/s, caution below 7.1 mm/s, and a red flag is reached above 7.1 mm/s.

Factors such as the 2X/1X amplitude line should be adjusted to detect misalignment warnings. Experimenting with speed sweeps and air flow helps to understand resonances or aerodynamics. Experimenting with foundation stiffness or damping can help identify the correct faults.

Using FFT averaging and spectral coherence can help identify the correct faults. Motor and bearing spot cross-correlation can tell where vibration is traveling, distinguishing between misalignment and bearing wear.

Keeping track of long-term data can help predict failures and adjust service schedules. By training machines to learn from past trends, machine learning can predict failures and adjust service schedules accordingly.

6. CONCLUSIONS

All in all, this paper contains a complete vibro-mechanical play book of the VOD-2.1 axial mine fan. The bottom line is that coupling stiffness and misalignment are responsible for the enormous differences in the vibration picture and the fingerprints of the faults. Heavy couplings increase harmonics and overall RMS, and flexible ones reduce vibration but are more often subject to checking. Three kinds of measurements (laser alignment, periodic balancing, ISO-based monitoring and FFT/cepstrum analysis enhance mine ventilation safety and reliability, reducing downtime, improving predictive maintenance, and ensuring compliance with vibration regulations.

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Analiza vibro-mecanică a sistemelor de ventilație în întreprinderi industriale specializate:studiu de caz și analiză experimentală a ventilatorului axial VOD-2.1 în ventilația minieră

Rezumat: Această lucrare prezintă o analiză vibro-mecanică a ventilatorului axial VOD-2.1 utilizat în ventilația principală a exploatărilor miniere. Măsurători triaxiale ale vibrațiilor au fost realizate la carcasele lagărelor și la suporturile motorului, în condiții controlate de rigiditate diferită a cuplajului și de nealiniere, pe baza unor investigații in situ anterioare. Analizele în domeniul timpului și al frecvenței (FFT) au permis identificarea componentelor dominante, a armonicilor și a semnăturilor de defect asociate nealinierii, dezechilibrului și tipului de cuplaj. Rezultatele arată că rigiditatea cuplajului influențează semnificativ răspunsul spectral, cuplajele rigide generând amplitudini mai mari și un spectru mai larg. Sunt propuse recomandări de diagnostic și mentenanță conform ISO 20816/ISO 10816 pentru creșterea fiabilității și siguranței sistemelor de ventilație minieră.

Cuvinte Cheie: aliniere, ventilație minieră, FFT, analiză vibrațională, ventilator axial, VOD-2.1, ISO 20816.

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