

Vibration Analysis of the Ammonia Compressor Motor unit #1 at PT. X

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Abstract—The ammonia compressor motor is a critical component in industrial refrigeration systems. This study aims to analyze the vibration condition of the ammonia compressor motor Unit #1 at PT. X based on ISO 10816-3 standard. Measurements were conducted on-site using Haliza 9.1 vibration analyzer at four rotational speed variations (600, 1200, 2100, and 3500 rpm) across six measurement points (DE and NDE in horizontal, vertical, and axial directions). The results show that vibration values increase significantly with speed, with the highest value reaching 10.797 mm/s at NDE Axial 3500 rpm. Evaluation based on ISO 10816-3 indicates critical conditions at high speeds, with two points in Zone D (dangerous) and four points in Zone C (not recommended). Frequency spectrum analysis identifies dominant vibration sources including mass unbalance, shaft misalignment, mechanical looseness, and bearing defects, particularly on the Non-Drive End side. The excitation force due to unbalance is estimated at 5.915 N with potential partial resonance between the system's natural frequency (111.5 Hz) and the 2nd harmonic (116.67 Hz). This study recommends stopping operation at high speeds, correcting unbalance and alignment, replacing NDE bearings, and implementing a periodic vibration analysis-based predictive maintenance program.

Keywords— Vibration analysis, Compressor motor, ISO 10816-3, Predictive maintenance, Bearing failure.

I. INTRODUCTION

The drive motor of the ammonia compressor system is one of the most critical components in an industrial refrigeration system, particularly in food-storage facilities such as those at PT. X. The screw-type ammonia compressor operates continuously to maintain stable refrigerant pressure and temperature. Unstable operation of the compressor drive motor can lead to production disturbances, increased energy consumption, and even potential catastrophic failure.

One of the primary parameters used to assess machine health is vibration. Excessive vibration may indicate early-stage faults such as unbalance, misalignment, looseness, or bearing defects. Therefore, vibration analysis is an essential part of a predictive maintenance program.

The three-phase electric motor driving the ammonia compressor Unit #1 at PT. X has a capacity of 291 kW, operates continuously, and plays a crucial role in the reliability of the refrigeration system. On-site vibration measurements are required to determine the motor's condition based on international standards. The vibration health assessment refers to ISO 10816-3, which is widely applied in industry for rotating machinery in Categories II–III. Based on these conditions, this study was conducted to analyze the vibration condition of the ammonia compressor motor Unit #1 through direct measurements and evaluation in accordance with ISO standards.

II. LITERATURE REVIEW

Industrial refrigeration systems widely use ammonia compressors due to their high thermodynamic efficiency and good reliability for large-scale cooling applications. The continuity of compressor operation heavily depends on the performance of its drive motor. Damage to the drive motor can cause production stoppages, increased operational costs, and potentially lead to damage to other refrigeration system

components. Therefore, monitoring the condition of rotating machinery is necessary to maintain system reliability.

One of the most effective machine condition monitoring methods is vibration analysis. Vibration monitoring can detect mechanical degradation at an early stage without the need to disassemble the machine. Periodic vibration monitoring enables early detection of damage, thus preventing sudden machine failure and supporting the implementation of predictive maintenance in industrial equipment.[1]

A. Basic Vibration Parameters

Mechanical vibration can be expressed in three main parameters: displacement, velocity, and acceleration. These three parameters are interrelated and each has sensitivity to specific types of damage.

1. Vibration Displacement

Displacement is the maximum deviation of a particle from its equilibrium position due to vibration. Simple harmonic vibration can be expressed in the form of sinusoidal displacement, where the displacement amplitude depends on the angular frequency of the vibration system[2]. Mathematically, simple harmonic vibration can be expressed as:

$$x(t) = X \sin(\omega t)$$

The displacement parameter is generally used for low-speed machines, but remains relevant as a theoretical basis in vibration analysis.

2. Vibration Velocity

Vibration velocity is the first derivative of displacement with respect to time and is expressed as:

$$v(t) = \frac{dx(t)}{dt} = \omega X \cos(\omega t)$$

Vibration velocity is the first derivative of displacement with respect to time and is used as the main parameter in evaluating industrial machine condition[1]. The Root Mean Square (RMS) value of vibration velocity is formulated as:

$$v_{RMS} = \frac{\omega X}{\sqrt{2}}$$

The RMS vibration velocity value represents the overall vibration energy and is used as the standard parameter in evaluating rotating machinery vibration [2][1] because it can represent the overall vibration energy. International standards such as ISO 10816-3 use this parameter as the main reference.

3. Vibration Acceleration

Vibration acceleration is the second derivative of displacement with respect to time and can be written as:

$$a(t) = -\omega^2 X \sin(\omega t)$$

The RMS value of vibration acceleration is:

$$a_{RMS} = \frac{\omega^2 X}{\sqrt{2}}$$

The vibration acceleration parameter is sensitive to high frequencies and is often used to detect early bearing damage [2].

B. Fundamental Harmonic Vibration Frequency

The fundamental vibration frequency is determined by the shaft rotational speed and becomes the main reference in FFT spectrum analysis [3]. The fundamental rotational frequency ($1 \times$ RPM) is calculated using the equation:

$$f_1 = \frac{n}{60}$$

In addition to the fundamental frequency, there are harmonic frequencies which are multiples of the fundamental frequency:

$$f_n = n \times f_1$$

Vibration harmonics are multiples of the fundamental frequency and are often used to identify types of disturbances such as misalignment and looseness [3]. Harmonic analysis is very important in identifying vibration sources, where the $1 \times$ RPM frequency is generally associated with unbalance, the $2 \times$ RPM frequency is associated with misalignment, and high harmonic frequencies can indicate looseness or structural instability.

C. Bearing Damage Frequencies

Damage to rolling elements and races in bearings produces characteristic bearing frequencies such as BPFO, BPFI, BSF, and FTF, which are used to identify the location and type of bearing damage through vibration spectrum analysis [2]. Identification of these frequencies in the vibration spectrum is used to determine the location and type of bearing damage at an early stage.

$$BPFO = \frac{N}{2} f_r \left(1 - \frac{d}{D} \cos \theta\right)$$

$$BPFI = \frac{N}{2} f_r \left(1 + \frac{d}{D} \cos \theta\right)$$

$$BSF = \frac{D}{2d} f_r \left(1 - \left(\frac{d}{D} \cos \theta\right)^2\right)$$

$$FTF = \frac{1}{2} f_r \left(1 - \frac{d}{D} \cos \theta\right)$$

D. Bearings Used in the Object

The bearings used in the ammonia compressor drive motor Unit #1 are deep groove ball bearings type 6314 C3 installed on the Drive End (DE) and Non-Drive End (NDE) sides. In vibration analysis, the geometric characteristics of the bearing greatly influence the determination of bearing damage frequencies such as BPFO, BPFI, BSF, and FTF.

TABLE 1. Specifications of bearing 6314 C3

Parameter	Value
Bearing Code	6314 C3
Bearing Type	Deep Groove Ball Bearing
Inner diameter (d)	70 mm
Outer diameter (D)	150 mm
Bearing width (B)	35 mm
Pitch diameter (Dp)	± 110 mm
Rolling element diameter (d)	± 25 mm
Number of balls (n)	8 pieces
Contact angle (θ)	0° (radial bearing)

E. Vibration Severity Evaluation Based on ISO 10816-3

Machine condition evaluation is carried out based on ISO 10816-3 standard. This standard classifies machine condition based on RMS velocity values measured on non-rotating parts, such as bearing housings.

Machine condition is divided into four zones:

- Zone A: very good condition
- Zone B: still acceptable for continuous operation
- Zone C: unsatisfactory condition, requires maintenance
- Zone D: dangerous condition, requires immediate action

Machines in zones A and B are still safe to operate, while zones C and D require maintenance actions to prevent more serious damage [4].

TABLE 2. ISO 10816-3 vibration standard

ISO 10816-3 vibration standard	Machine group 4		Machine group 3		Machine group 2		Machine group 1	
	Integral driver		External driver		Motors		Motors	
Velocity	Pumps > 15 kW				180 mm \leq H \leq 315 mm		315 mm \leq H	
mm/s rms	Radial, axial, mixed flow				Medium sized machines		Large machines	
in/sec rms					15 kW < P \leq 300 kW		300 kW < P \leq 50 MW	
11	0.44							
7.1	0.28							
4.5	0.18							
3.5	0.11							
2.8	0.07							
2.3	0.04							
1.4	0.03							
0.71	0.02							
Foundation	Rigid	Flexible	Rigid	Flexible	Rigid	Flexible	Rigid	Flexible

A New machine condition
B Unlimited long-term operation allowable

C Short-term operation allowable
D Vibration causes damage

III. METHODOLOGY

The research flow in this study was systematically arranged to ensure that the data collection, processing, and machine condition evaluation processes were carried out in a structured manner. The research flowchart is used to illustrate the stages of activities carried out, from problem identification to determining the diagnosis of the ammonia compressor motor condition. Each stage is interrelated, where the results

of one stage become the basis for implementing the next stage. With this flowchart, the research process can be understood more clearly and facilitates tracing the analytical methods used.

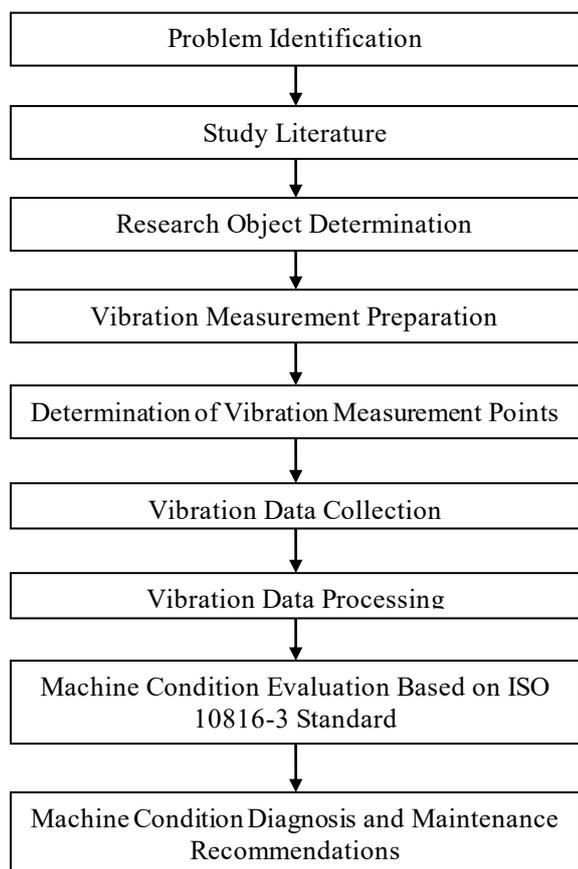


Fig 1. Vibration Analysis Flowchart

A. Problem Identification

Problem identification involves reviewing the importance of the ammonia compressor drive motor reliability in industrial refrigeration systems. At this stage, initial information is collected regarding operational disturbances, potential machine damage, and the need for condition monitoring using vibration analysis.

B. Study Literature

Literature study is conducted as a theoretical basis for carrying out the research. At this stage, various references related to rotating machinery vibration analysis, mechanical damage characteristics, vibration measurement methods, and machine condition evaluation standards are collected and reviewed. The references used include textbooks, scientific journals, international standards, and previous research relevant to condition monitoring of electric motors and industrial compressors.

The literature study focuses on understanding the basic principles of vibration, vibration parameters (displacement, velocity, and acceleration), frequency spectrum analysis methods using Fast Fourier Transform (FFT), and identification of damage types such as unbalance,

misalignment, looseness, and bearing damage. In addition, at this stage the ISO 10816-3 standard is also studied as a reference for determining the severity level of machine vibration based on RMS velocity values.

The results of the literature study are used as a foundation for determining measurement methods, determining data collection points, data processing techniques, and machine condition evaluation criteria. Thus, all stages of the research have a clear scientific basis and are in accordance with rotating machinery condition monitoring practices in industry.

C. Research Object Determination

The object of this research is a three-phase induction motor used as the drive motor for the screw-type ammonia compressor Unit #1 at PT. X. The motor operates continuously to support the industrial refrigeration system. The reliability of this motor is very important because any operational disturbance can directly affect the cooling process and production continuity.

The motor has an installed power of 291 kW and operates under continuous duty conditions. Due to long operating hours and mechanical loads from the compressor, the motor has the potential to experience mechanical degradation, especially in bearings, shaft alignment, and rotor balance. Therefore, machine condition monitoring through vibration analysis is needed to assess the health level of the machine.

D. Vibration Measurement Preparation

The preparation stage includes visual inspection of the motor condition, checking access to measurement points, and preparing equipment. Vibration measurements are carried out using a portable vibration analyzer equipped with an accelerometer sensor. The accelerometer sensor is magnetically mounted on the bearing housing surface to capture machine vibration signals. The measuring instrument can measure vibration parameters in the form of velocity (mm/s RMS) and record time domain signals for frequency spectrum analysis. Additionally, instrument parameter settings and calibration are performed to ensure accurate and accountable measurement results.

E. Determination of Vibration Measurement Point

At this stage, the sensor mounting locations are determined on the Drive End (DE) and Non-Drive End (NDE) bearing housings. Measurements are carried out in three directions: horizontal, vertical, and axial. The determination of measurement directions aims to detect various types of mechanical disturbances, where the radial direction is sensitive to unbalance and looseness, while the axial direction is used to detect misalignment.

F. Vibration Data Collection

Data collection is carried out on-site when the motor is operating under normal conditions. At each measurement point, the overall vibration value in the form of RMS velocity is recorded and the time domain vibration signal is captured. Measurements are repeated to ensure that the data obtained is representative of the machine condition.

Vibration data is collected when the compressor motor is operating under normal load conditions (on-site measurement). The measurement procedure includes the following steps:

1. Visual inspection of the motor and surrounding components.
2. Preparation and calibration of the vibration analyzer.
3. Mounting the accelerometer sensor at each measurement point.
4. Recording the overall vibration value in the form of RMS velocity.
5. Recording the time domain vibration signal
6. Capturing the FFT spectrum

Measurements are repeated at all six measurement points to obtain consistent and representative data.

G. Vibration Data Processing

The vibration data obtained is then processed to obtain RMS velocity values and frequency spectra using the Fast Fourier Transform (FFT) method. RMS values are used to determine the severity level of vibration, while frequency spectra are used to identify vibration sources based on the characteristic frequencies that appear.

H. Machine Condition Evaluation Based on ISO 10816-3

The RMS velocity values from the measurements are compared with the limits of the ISO 10816-3 standard. Based on this standard, the machine condition is classified into zones A, B, C, or D to determine whether the machine is still suitable for operation or requires maintenance action.

I. Machine Condition Diagnosis and Maintenance Recommendations

After the vibration severity level is known, machine condition diagnosis is carried out by linking the standard evaluation results and frequency spectrum analysis. Based on this diagnosis, maintenance action recommendations are then prepared, such as further inspection, realignment, balancing, or bearing replacement to prevent machine failure.

IV. RESULTS AND DISCUSSION

Vibration measurements were carried out using a Haliza 9.1 vibration analyzer under normal motor operating conditions with rotational speed variations of 600 rpm, 1200 rpm, 2100 rpm, and 3500 rpm under normal usage. The data collection parameters were velocity (mm/s) and overall RMS values in accordance with ISO 10816-3 standard.

A. Vibration Data Collection

Velocity Drive End (DE):

At the DE Horizontal point, vibration data collection produced a frequency graph from 0-1000 Hz, RMS values, and the motor rotational speed was 3500 rpm. The highest frequency point was at 47.48 Hz, with an overall RMS value of 6.382.

At the DE Vertical point, vibration data collection produced a frequency graph from 0-1000 Hz, RMS values, and the motor rotational speed was 3500 rpm. The highest

frequency point was at 23.74 Hz, with an overall RMS value of 6.399.

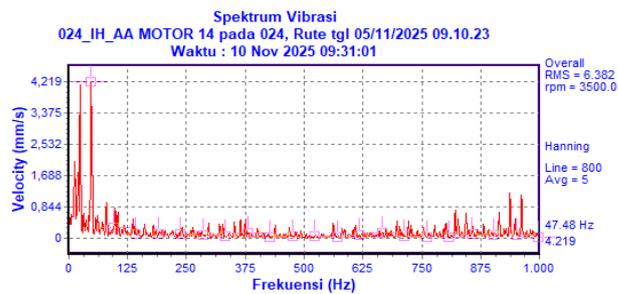


Fig. 1. DE Horizontal velocity vibration graph

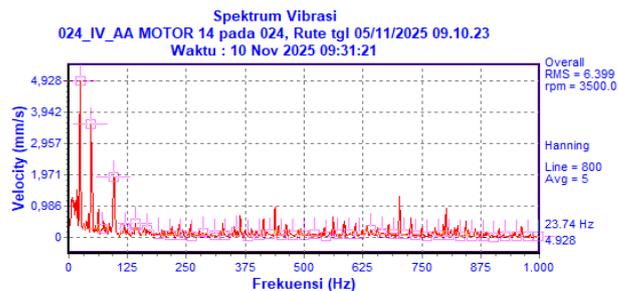


Fig. 2. DE Vertical velocity vibration graph

At the DE Axial point, vibration data collection produced a frequency graph from 0-1000 Hz, RMS values, and the motor rotational speed was 3500 rpm. The highest frequency point was at 23.74 Hz, with an overall RMS value of 4.732.

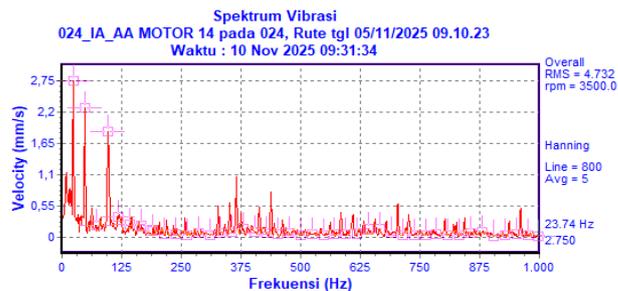


Fig. 3. DE Axial velocity vibration graph

Velocity Non-Drive End (NDE):

At the NDE Horizontal point, vibration data collection produced a frequency graph from 0-1000 Hz, RMS values, and the motor rotational speed was 3500 rpm. The highest frequency point was at 47.48 Hz, with an overall RMS value of 7.177.

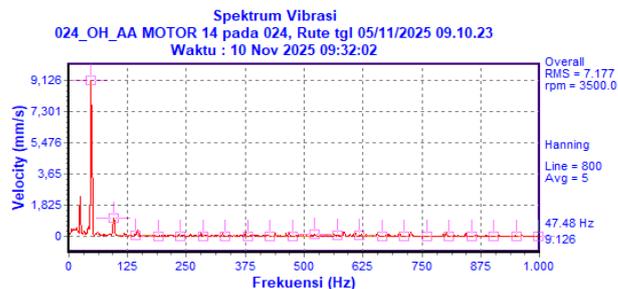


Fig. 4. NDE Horizontal velocity vibration graph

At the NDE Vertical point, vibration data collection produced a frequency graph from 0-1000 Hz, RMS values, and the motor rotational speed was 3500 rpm. The highest frequency point was at 47.48 Hz, with an overall RMS value of 5.525.

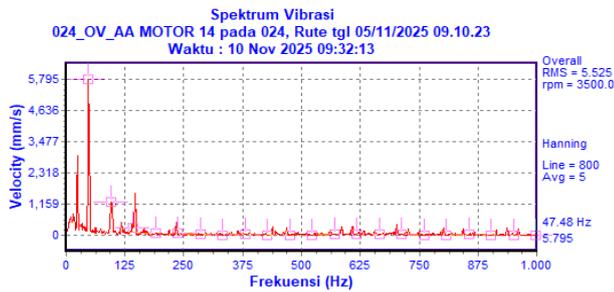


Fig. 5. NDE Vertical velocity vibration graph

At the NDE Axial point, vibration data collection produced a frequency graph from 0-1000 Hz, RMS values, and the motor rotational speed was 3500 rpm. The highest frequency point was at 23.74 Hz, with an overall RMS value of 10.997.

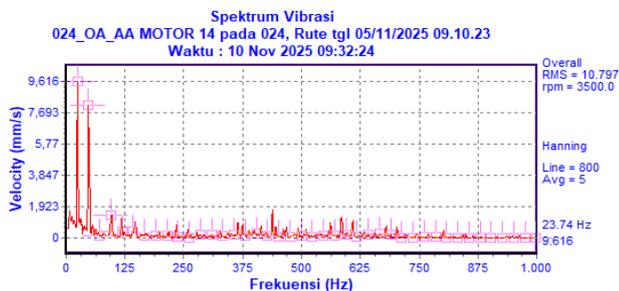


Fig. 6. NDE Axial velocity vibration graph

All measurement results with speed variations of 600 rpm, 1200 rpm, 2100 rpm, and 3500 rpm can be seen in Table 1 below.

TABLE 1. Motor vibration measurement results based on four speed variables

Position	Velocity (mm/s)			
	600	1200	2100	3500
DE-H	3.208	3.168	5.442	6.382
DE-V	3.126	3.161	3.620	6.399
DE-A	2.940	2.799	5.056	4.732
NDE-H	4.007	4.660	4.064	7.177
NDE-V	2.628	2.444	2.659	5.525
NDE-A	3.913	4.492	3.659	10.797

To see the differences in measurement results contained in Table 1, a graph of these measurement results was created, as can be seen in Figure 7.

From the graph generated in Figure 7, it can be concluded that motor vibration increases significantly at high speeds, with the NDE side being more susceptible to increased vibration, especially in the axial direction. This can serve as a basis for recommendations for further inspection of non-drive end components, as well as evaluation of system balance and alignment at high operational speeds.

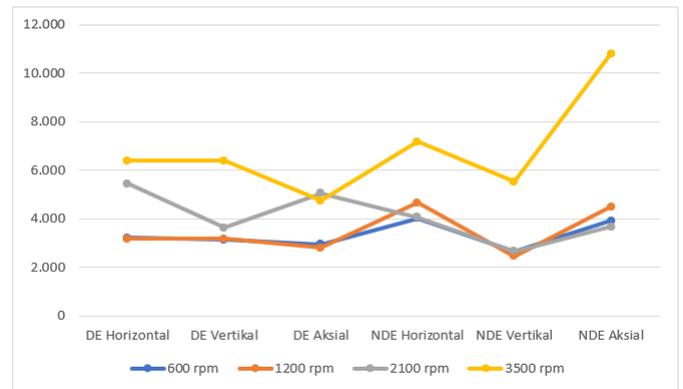


Fig. 7. Graph of velocity vibration measurement results

B. Vibration Data Analysis

1. Fundamental Harmonic Frequency Analysis

The fundamental harmonic frequency of the motor is calculated based on the motor rotational speed variables of 600 rpm, 1200 rpm, 2100 rpm, and 3500 rpm using the equation:

$$f_1 = \frac{n}{60}$$

$$f_2 = 2 \times f_1 \quad f_3 = 3 \times f_1$$

The above equation is calculated based on the motor rotational speed. From each motor rotational speed, the results are shown in Table 2 and the fundamental frequency amplitudes are shown in Table 3 below.

TABLE 2. Results of fundamental harmonic frequency equation calculations

Speed	Frequency (Hz)		
	f_1	f_2	f_3
600 rpm	10	20	30
1200 rpm	20	40	60
2100 rpm	35	75	105
3500 rpm	58.33	116.66	175

TABLE 3. Vibration threshold guidelines for fundamental harmonic frequencies

Frequency	Amplitude		
	Normal Range (mm/s)	Alarm Level (mm/s)	Critical Level (mm/s)
1× RPM	< 2,0	2,0 - 3,5	> 3,5
2× RPM	< 1,0	1,0 - 2,5	> 2,5
3× RPM	< 0,5	0,5 - 1,5	> 1,5

2. Bearing Characteristic Frequency Analysis

A recap of the bearing characteristic frequency calculation results can be seen in Table 4, and the vibration threshold guidelines for bearing damage frequencies can be seen in Table 5 below.

TABLE 4. Characteristic Frequencies of Bearing 6314 C3 Based on Speed Variations

Speed (rpm)	Frequency (Hz)			
	BPFO	BPMF	BSF	FTF
600	30.91	49.09	20.86	3.86
1200	61.82	98.18	41.73	7.73
2100	108.18	171.82	73.02	13.52
3500	180.4	286.4	121.9	22.5

TABLE 5. Vibration threshold guidelines for bearing damage frequencies

Kode	Amplitude		
	Normal Condition (mm/s)	Alarm Level (mm/s)	Critical Level (mm/s)
BPFO	< 0,3	0,3 - 0,8	> 0,8
BPFI	< 0,3	0,3 - 0,8	> 0,8
BSF	< 0,2	0,2 - 0,6	> 0,6
FTF	< 0,1	0,1 - 0,3	> 0,3

3. Frequency Spectrum Analysis

Frequency spectrum analysis was conducted to identify dominant frequency components at each measurement point. The following is an analytical table of the frequency spectrum compiled based on FFT data and speed variables from the measurements. After conducting frequency spectrum analysis on all measurement points (DE Horizontal, DE Vertical, DE Axial, NDE Horizontal, NDE Vertical, and NDE Axial) at the four speed variations (600 rpm, 1200 rpm, 2100 rpm, and 3500 rpm), a recapitulation table was then created to see a comprehensive comparison between the average amplitude of measurement results at each speed variation with limits consisting of three zones: Normal (good condition, no action required), Alarm (developing condition, needs monitoring and repair planning), and Critical (dangerous condition, immediate action required) as per Tables 3 and 5. The comparison with existing spectrum analysis results can be seen in Table 6 below.

TABLE 6. Complete recapitulation of frequency measurement result comparison

Frequency	Rpm	Average Amplitude (mm/s)	Status	% of Normal Limit
1× RPM	600	2,09	ALARM	+4,5%
	1200	2,35	ALARM	+17,5%
	2100	3,01	ALARM	+50,5%
	3500	4,41	CRITICAL	+120,5%
Increasing Trend 600→3500 = +111% (WORSENING)				
2× RPM	600	0,72	NORMAL	-28%
	1200	0,99	NORMAL	-1%
	2100	1,85	ALARM	+85%
	3500	3,08	CRITICAL	+208%
Increasing Trend 600→3500 = +327,8% (WORSENING)				
3× RPM	600	0,39	NORMAL	-22,0%
	1200	0,47	NORMAL	-6%
	2100	0,78	ALARM	+56%
	3500	1,40	ALARM	+180%
Increasing Trend 600→3500 = +259% (WORSENING)				
BPFO	600	0,20	NORMAL	-33,3%
	1200	0,32	ALARM	+6,7%
	2100	0,73	ALARM	+143,3%
	3500	1,05	CRITICAL	+250%
Increasing Trend 600→3500 = +425% (WORSENING)				
BPFI	600	0,12	NORMAL	-60%
	1200	0,19	NORMAL	-36,7%
	2100	0,43	ALARM	+43,3%
	3500	0,62	ALARM	+106,7%
Increasing Trend 600→3500 = +416,7% (WORSENING)				
BSF	600	0,25	ALARM	+25%
	1200	0,41	ALARM	+105%
	2100	1,02	CRITICAL	+410%
	3500	1,81	CRITICAL	+805%
Increasing Trend 600→3500 = +624% (WORSENING)				

Analysis of the amplitude increase trend based on damage type:

- Unbalance shows a 111% increase from 600 to 3500 rpm, indicating the presence of consistent unbalanced

mass. At 3500 rpm, the amplitude has exceeded the critical limit by 26%, indicating that unbalance has reached a dangerous level.

- Misalignment shows the most dramatic increase (328%) compared to other parameters. This indicates that misalignment between the motor and compressor worsens with increasing speed. At 3500 rpm, the 2× RPM amplitude has exceeded the critical limit by 23.2%, with very clear dominance in the NDE axial direction (5.42 mm/s). This pattern is consistent with severe parallel misalignment.
 - Looseness begins to be detected at 2100 rpm and worsens significantly at 3500 rpm. Although it has not reached the critical limit (1.5 mm/s), the value of 1.40 mm/s is very close to this limit. The dominant 3rd harmonic, especially at NDE Axial (2.48 mm/s), indicates mechanical looseness in the motor structure or components, possibly due to prolonged vibration loosening foundation bolts or bearing housings.
 - Outer race bearing damage is first detected at 1200 rpm and worsens consistently. At 3500 rpm, the BPFO amplitude has exceeded the critical limit by 31.3%. NDE Horizontal shows the highest value (1.85 mm/s), indicating that the NDE side outer race bearing is experiencing the most severe damage. This pattern is consistent with fatigue spalling on the outer race due to cyclic loading and possible misalignment exacerbating the damage.
 - Inner race bearing damage begins to be detected at 2100 rpm and worsens at 3500 rpm, but is still within the alarm limit (not yet critical). NDE Horizontal shows the highest value (0.92 mm/s at 3500 rpm). Inner race defects typically develop more slowly than outer race defects because the inner race rotates with the shaft and experiences more varied load distribution. Sidebands around BPFI in the spectrum confirm amplitude modulation due to shaft rotation.
 - BSF is the most critical damage indicator with the highest increase (624%) and has exceeded the critical limit since 2100 rpm. At 3500 rpm, the BSF amplitude reaches 1.81 mm/s, 805% above the normal limit and 201.7% above the critical limit. NDE Axial shows an extreme value of 3.85 mm/s (541.7% above the critical limit). This indicates severe spalling on the NDE side bearing rolling elements (balls). Rolling element damage is very dangerous because it can cause ball fragmentation, excessive heat generation, and sudden total bearing failure.
4. Calculation of Excitation Force Due to Unbalance
Given:
Rotor mass $m = 420 \text{ kg}$
Eccentricity $e = 0,1 \text{ mm} = 1 \times 10^{-4}$
Angular velocity $\omega = 2\pi f1 = 375,1 \text{ rad/s}$
 $F_u = m \cdot e \cdot \omega^2 = 420 \times (1 \times 10^{-4}) \times (375,1)^2 = 5909 \text{ N} = 5,91 \text{ kN}$
This force can cause significant vibration, especially if the excitation frequency approaches the system's natural frequency.
5. Evaluation Based on ISO 10816-3 Standard

The measurement results were then compared with these limits to determine the motor's health condition. The evaluation results can be seen in the table 7 below:

TABLE 7. Vibration level classification based on ISO 10816-3

Position	Velocity (mm/s)	Zone	Description
DE-H	6,382	C	Not recommended
DE-V	6,399	C	Not recommended
DE-A	4,732	C	Not recommended
NDE-H	7,177	D	Dangerous
NDE-V	5,525	C	Not recommended
NDE-A	10,797	D	Critical – Must stop

V. DISCUSSION

Vibration measurements on the ammonia compressor motor Unit #1 were carried out at four rotational speed variations: 600 rpm, 1200 rpm, 2100 rpm, and 3500 rpm. Measurements were taken at two main positions, namely Drive End (DE) and Non-Drive End (NDE), each with three measurement directions: Horizontal, Vertical, and Axial. The data obtained were RMS velocity values (mm/s) taken directly using a Haliza 9.1 vibration analyzer.

A. Vibration Analysis Data at 600 rpm Rotational Speed

At 600 rpm, the measured vibration values were still within relatively low limits. The highest vibration point occurred at NDE Horizontal with a value of 4.007 mm/s, while the lowest point was at NDE Vertical with a value of 2.628 mm/s. In general, all measurement points were still in Zone B according to ISO 10816-3, which means the condition is still permissible for normal operation.

B. Vibration Analysis Data at 1200 rpm Rotational Speed

At 1200 rpm, there was an increase in vibration values compared to 600 rpm. The highest values were recorded at NDE Horizontal at 4.660 mm/s and NDE Axial at 4.492 mm/s. Both of these points entered Zone C (not recommended), while the other points were still in Zone B.

C. Vibration Analysis Data at 2100 rpm Rotational Speed

At 2100 rpm, vibration increased further with the highest values at DE Horizontal at 5.442 mm/s and DE Axial at 5.056 mm/s. Both entered Zone C. Meanwhile, NDE Horizontal and NDE Vertical were still in Zone B, indicating that the drive end side was beginning to show signs of wear or imbalance.

D. Vibration Analysis Data at 3500 rpm Rotational Speed

At the highest operating speed (3500 rpm), vibration reached alarming values. NDE Horizontal reached 7.177 mm/s and NDE Axial reached 10.797 mm/s, both of which entered Zone D (dangerous). DE Horizontal and DE Vertical also entered Zone C. This indicates that the motor is in a critical condition and requires immediate action.

E. Identification of Vibration Sources

Based on the frequency spectrum analysis that has been carried out, the dominant vibration sources causing the significant increase in vibration values were identified. This identification was based on characteristic frequency patterns,

relative amplitudes, dominant vibration directions, and comparison with EASA threshold standards and ISO 10816-3.

1. Unbalance occurs when the rotor's center of mass does not coincide with its center of rotation. In this motor, unbalance was detected as early as 600 rpm with an amplitude of 2.09 mm/s, which had already entered the alarm zone (+4.5% above the normal limit). At 3500 rpm, the 1× RPM amplitude reached 4.41 mm/s, exceeding the critical limit (3.5 mm/s) by 26%.
2. Misalignment is a condition of misalignment between the motor shaft and the compressor shaft connected through a coupling. In this case, there is a very strong dominance of 2× RPM in the axial direction (especially NDE Axial: 5.42 mm/s). Based on the 2× RPM amplitude reaching 5.42 mm/s (442% above the normal limit, 116.8% above the critical limit), this condition is very severe and causes: Excessive axial load on the NDE bearing – The Non-Drive End bearing receives continuous axial force that should not exist in an induction motor with deep groove ball bearings (designed for dominant radial loads).
3. Mechanical looseness atau kelonggaran mekanis adalah Mechanical looseness is a condition where there is excessive clearance in structural components or rotating components. In this motor, the odd harmonic pattern that appears significantly at speeds of 2100-3500 rpm, especially on the NDE side, indicates developing structural looseness and bearing looseness. The mechanism of occurrence: Stage 1 (600-1200 rpm): Vibration due to unbalance and misalignment is still relatively low, not enough to loosen components. The 3× RPM amplitude is still below the alarm limit. Stage 2 (2100 rpm): Vibration increases sharply, periodic dynamic forces begin to loosen foundation bolts and bearing housings. The 3× RPM amplitude reaches 0.78 mm/s (alarm zone). Stage 3 (3500 rpm): Vibration is very high, the loosening effect creates a "positive feedback loop" – vibration loosens components, loose components amplify vibration. The 3× RPM amplitude reaches 1.40 mm/s, only 0.1 mm/s from the critical limit (1.5 mm/s).
4. Bearing Damage
 - a. BPFO (*Ball Pass Frequency Outer Race*)
In this motor, BPFO began to be detected at 1200 rpm (0.32 mm/s, alarm zone) and worsened progressively. An amplitude of 1.85 mm/s at NDE Horizontal at 3500 rpm indicates advanced spalling on the NDE side outer race bearing.
 - b. BPFI (*Ball Pass Frequency Inner Race*)
BPFI began to be detected at 2100 rpm (0.43 mm/s, alarm zone) and worsened at 3500 rpm (0.62 mm/s). Although it has not reached the critical limit (0.8 mm/s), the sharp increasing trend (417%) indicates that the inner race is also experiencing progressive damage.
 - c. BSF (*Ball Spin Frequency*)
At this frequency, BSF with amplitude > 0.6 mm/s is classified as critical – immediate bearing replacement required. The BSF amplitude of 3.85 mm/s at NDE Axial is 6.4 times the critical limit. This indicates very

severe spalling on the rolling elements of the NDE side 6314 C3 bearing. Spalling is material flaking due to rolling contact fatigue.

VI. CONCLUSION

Based on the vibration measurement and analysis results on the ammonia compressor motor Unit #1 at PT. X, it can be concluded that the motor vibration value increases significantly with increasing rotational speed. At 3500 rpm, the highest vibration value was recorded at the NDE Axial point at 10.797 mm/s, followed by NDE Horizontal at 7.177 mm/s. Evaluation based on the ISO 10816-3 standard shows that of the six measurement points at 3500 rpm operating speed, two points, namely NDE Horizontal and NDE Axial, are in Zone D which is dangerous and requires immediate action, while the other four points, namely DE Horizontal, DE Vertical, DE Axial, and NDE Vertical, are in Zone C which is not recommended for continuous operation and requires maintenance planning. The Non-Drive End side shows a more critical condition compared to the Drive End side.

Based on frequency spectrum analysis, the dominant vibration sources identified include unbalance with a $1 \times$ RPM amplitude reaching 4.41 mm/s or 26% above the critical limit and an excitation force of 1006.5 N, misalignment with the highest increase reaching 328% and a $2 \times$ RPM amplitude of 3.08 mm/s which is dominant in the NDE axial direction, mechanical looseness with a $3 \times$ RPM amplitude of 1.40 mm/s approaching the critical limit of 1.5 mm/s, and bearing damage, most critically indicated by BSF with an amplitude of 1.81 mm/s or 805% above the normal limit, indicating severe spalling on the NDE side bearing rolling elements. In addition, there is an indication of potential partial resonance between the system's natural frequency of 111.5 Hz and the 2nd harmonic of 116.67 Hz, which exacerbates the motor's vibration condition.

VII. FUTURE WORK

Based on the above conclusions, it is recommended to stop motor operation at 3500 rpm until repairs are carried out, replace the bearings on the Non-Drive End side type 6314 C3, and conduct visual inspections of coupling components and foundation bolts to identify mechanical looseness. For short-term repairs, it is necessary to perform rotor rebalancing to address unbalance, realign the motor and compressor shafts, especially to correct parallel misalignment, and check and tighten all foundation bolts and bearing housings.

For future research development, it is recommended to conduct thermal analysis by measuring bearing temperature simultaneously with vibration measurements to see the correlation between temperature rise and damage severity. Research can also be developed with motor current signature analysis to detect electrical disturbances such as rotor bar damage or air gap eccentricity that are not detected through conventional vibration analysis. Operational Deflection Shape testing is needed to visualize structural deformation patterns during operation and confirm potential resonance. Implementation of an online vibration monitoring system is

recommended for real-time early damage detection, as well as oil analysis to detect contaminants and wear particles as confirmation of bearing damage. Finally, a comparative study needs to be conducted by analyzing vibration on similar compressor units that are still in good condition as comparative data or baseline data.

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