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Investigation of Dynamic Characteristics of Vibrating Components and Structure Through Vibration Measurement and Operational Modal Analysis

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Abstract: Smooth plant production is utmost expected by plant owners since any unplanned shutdown is very much undesirable as it may incur loss of company's profits and other repercussions. Critical operational assets such as rotating machineries and their components, e.g., motor, pump, compressor, and turbines are crucial to be monitored in ensuring continuous operation of the plant. Therefore, vibration being one of the elements in predictive maintenance is much needed. This paper addresses the monitoring and findings for predictive maintenance on high vibration of a make-up gas compressor in one of the oil & gas and petrochemical plant in Malaysia. The conventional vibration measurement using vibration analyzer was utilized to identify the severity of the machine vibrations through the overall vibration value obtained on the motor and compressor. A specialized vibration measurement technique, namely, Operational Modal Analysis (OMA) was further deployed to verify the presence of resonance on the equipment by obtaining its dynamic characteristics, i.e., natural frequency and mode shape. The findings showed clear indication of resonance occurrence as similar frequency (112 Hz) was identified from both vibration measurement and operational modal analysis.

Keywords: Operational modal analysis, vibration measurement, dynamic characteristics, harmonic excitation

1. Introduction

Condition-based monitoring (CBM) is one of the crucial aspects in any plant's operation as this helps to avoid unscheduled plant outages, maintain plant efficiency and reliability of the machines. Predictive maintenance, being one of the key elements in monitoring the asset's health condition, is undeniably a convincing approach to determine the required service interval of a machine before it starts to fail, and maintenance of rotating machineries is essential to ensure accurate fault diagnosis of the machine [1]. Vibration perspective has also been around the research domains in the work of asset's health monitoring. The nature of plants' components being subjected to extreme dynamic stresses has put out a firm association of these systems with vibration studies, with the furtherance of the subject of study ranging from intact, damaged, and failed condition. For instance, vibration analysis has been opted in past study combined with failure analysis as the objective to detect failure mechanism of a compressor blade [2]. One of the main goals of vibration

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analysis includes characterization of system vibration, as well as yielding the severity of the phenomenon. The severity of the vibration is assessed by performing vibration measurement and is normally measured on bearings, bearing pedestals, or housing of industrial machines when the measurements are made at site or in-situ [3].

Continuous vibration monitoring of critical assets in a scheduled manner is important to see the vibrations pattern and trending as early fault detection can be predicted through vibration spectrum and time-waveform. As such, vibration related studies in the mechanical industries have contributed to a few massive diagnostic inputs into the knowledge boundary. A vibration analysis has been conducted on an outbound pipeline of the Yongchang pressure station through field vibration test and the finding was compared with numerical model [4]. Few actionable results were found which implied that the source of abnormal vibration was concluded rooting from fluid pressure fluctuation in the pipeline. The proportional relation between the gas transmission volume and the pipeline vibration was also established from the results, along with comprehensive proposal of appropriate schemes in reducing the vibration problems. Another vibration analysis was opted to study on a screw compressor outlet piping system due to premature failures experienced [5]. From the works, the numerical results denoted that the vibration problem may be accredited from the pressure nonuniformity of the gas inside the pipes, low overall stiffness of the piping system and the first-order structural resonance occurred on the thermowells. [6] also utilized vibration properties to investigate operating failures in turbocharger centrifugal compressor by simulating damage scenarios of the blades component, aiming to establish a quantitative comparison system between vibration analysis and acoustic analysis. Vibration studies hold a significant role in the engineering field, simultaneously providing a great area of discipline to be explored. The rigidity of vibration monitoring hence needs to be understood first prior to advanced practical application. In certain conditions, basic vibration monitoring could not provide sufficient information in confirming events such as resonance. Thus, advanced vibration measurement methods such as Operational Modal Analysis (OMA) shall be implemented to further verify and validate the results.

The preference of OMA has significantly progressed in the last years due to the way it eases the difficult configuration of performing modal analysis during online or operating condition. [7]. Operational or also known as output-only modal analysis is an advanced method in obtaining the dynamic characteristics of a mechanical structure using the surrounding and normal operating condition as the unmeasured input [8]. Operational condition testing is highly favored in the mechanical engineering industries where the structures are usually complex and huge in size, posing incompatibility for experimental modal analysis. Due to this issue, operating condition measurement is viable using OMA because OMA treats the excitation as free and natural while utilizing only the output signals from the structure to produce its modal information. OMA method is also widely used in the field of Structural Health Monitoring (SHM) where damage detection is monitored from the ambient vibration testing [9]. In this viewpoint, OMA was opted for having the capability to measure the whole structure simultaneously instead of localized portion of the structure. In Saidin et al., OMA was used in conjunction with FE method as the procedure to verify the test data. The maximum percentage difference recorded between OMA natural frequencies and FEM natural frequencies of the test bridge has implied good agreement between the two methods which further support OMA as an accurate dynamic properties' processor.

OMA offers a few computation algorithms with their own advantages and disadvantages. (See [10]). Frequency Domain Decomposition (FDD), Enhanced Frequency Domain Decomposition (EFDD), and Stochastic Subspace Identification (SSI) are among the different algorithms being applied in modal determination using OMA. In this case, only FDD is being applied. FDD is the most straight forward technique in OMA application. Also known as the non-parametric method, FDD is the extension of the Basic Frequency Domain (BFD) technique [11] which from the spectral densities calculated, the modes of vibrations are obtained where the modal parameters are estimated directly from the signal processing calculations [12].

Due to the presence of harmonic forces, which complicate the estimation process, it is challenging to measure the dynamics characteristics of rotating equipment. In order to decompose the necessary modal parameters, the influence of harmonic components must be effectively eliminated. This paper focuses on a method for determining the modal parameters of rotating equipment and its connecting structure to enable early detection and resolution of problems, particularly in the case of resonance. The method presented is capable of identifying the required modal parameters while the system is in operation, which improves on the commonly used method in industry that requires the system to be offline or stop operation. This study is, hence, important to verify the presence of resonance of the make-up gas compressor unit where both measurement techniques are to be employed for comparison and verification. These outcomes are expected to be helpful for the predictive maintenance purpose.

2. Methodology

To investigate the vibration of the compressor, two means of modal parameter extraction were implemented. The first one was vibration measurement (VM) using Adash vibration analyzer. This equipment comes in handy in the work of vibration measurement and vibration analysis for detecting early warnings related to bearing conditions for industrial applications. Then, operational modal analysis (OMA) was conducted to post-process the vibration data obtained from separate measurement under OMA setup. In the setup of OMA, a set of accessories were assembled. The setup condition of both methods will be detailed out in this section.

2.1 Vibration Measurement (VM)

Vibration measurement was carried out using Adash A4500 VA5 Pro vibration analyser to acquire data on the bearing of each of the rotating parts, i.e., motor and compressor. The accelerometer used is CTC brand (AC102-1A) with a sensitivity of 100mV/g as shown in (Fig. 1). The setup of vibration measurement for this project were as such; f_{min} = 2 Hz, f_{max} = 1000 Hz, averaging = linear, sampling = 4096, and average = 4. The vibration measurement was performed on both motor and compressor side in which the measurement points were in the horizontal, vertical, and axial directions at the bearing locations. The reciprocating compressor was analysed in accordance with ISO 10816-8 (Mechanical Vibration - Evaluations of machine vibration by measurement on non-rotating parts - Reciprocating Compressor System) as shown in Table 4, while the acceptance severity range for the motor was compared against the OEM standard (Table 5).

Fig. 1 - Adash A4500 VA5 Pro vibration analyzer with CTC accelerometer

The vibration was measured at both drive end (DE) and non-drive end (NDE) of the machines as depicted in Fig. 2 and Fig. 3 respectively.

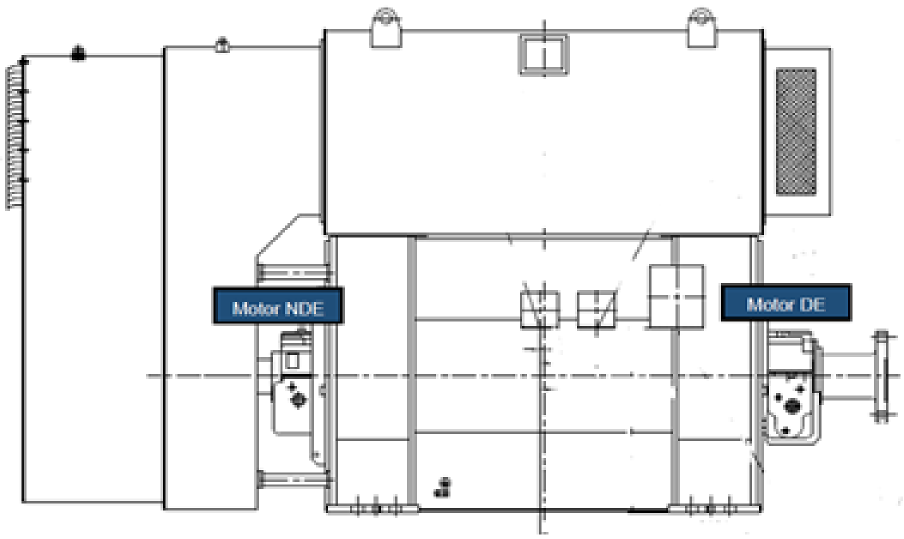


Fig. 2 - Motor configuration

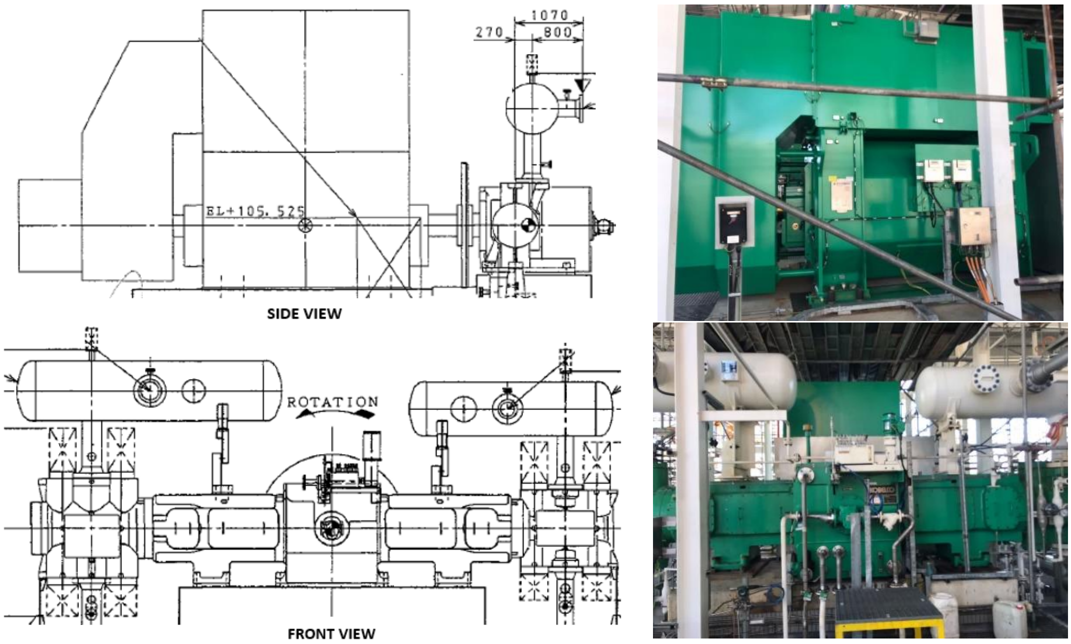


Fig. 3 - Motor-compressor configuration

2.2 Operational Modal Analysis (OMA)

The equipment and tools used for OMA measurement were laptop, multichannel analyser as shown in Fig. 4, and other accessories such as cables, accelerometers, and mounting clips. The accelerometer used for OMA measurement was Bruel & Kjaer Type 4508. It has a frequency range of 0.3 Hz - 8000 Hz and uses a clip adhesive mounting. The pre-processing part was done on OMA Pulse software while the post-processing part was done using ARTeMIS software for modal identification. Two (2) sets of OMA measurement were taken which covered different boundaries of the system:

- OMA Measurement 1: Motor Endplate NDE
- OMA Measurement 2: Motor Endplate DE

Fig. 4 - Bruel & Kjaer Type 3560-D

Twenty (20) accelerometers with 20 degrees of freedoms (DOFs) were used for both measurements. Only one data set was taken for each measurement, thus, each measurement was taken simultaneously, and no reference accelerometer was required. The setup for OMA for each measurement is shown in Table 1 and Table 2 below. Due to restriction of bringing a camera at site, no images of measurement setup at site were captured.

From Table 1and Table 2, the DOF (degree of freedom) was assigned in such a way that the X-axis in OMA follows the Axial direction in vibration measurement. The reason being is because the highest vibration measurement was found in the Axial direction which will be discussed further in the next section. Operational modal analysis was performed with two (2) measurements at different planes, namely, motor endplate NDE and motor end-plate DE. Fig. 5 and Fig. 6 show the geometry for each plane.

Table 1 - OMA setup for Measurement 1

Channel	Node	DOF
1	1	+X
2	2	+X
3	3	+X
4	4	+X
5	5	+X
6	6	+X
7	7	+X
8	8	+X
9	9	+X
10	10	+X
11	11	+X
12	12	+X
13	13	+X
14	14	+X
15	15	+X
16	16	+X
17	17	+X
18	18	+X
19	19	+X
20	20	+X

Table 2 - OMA setup for Measurement 2

Channel	Node	DOF
1	1	-X
2	2	-X
3	3	-X
4	4	-X
5	5	-X
6	6	-X
7	7	-X
8	8	-X
9	9	-X
10	10	-X
11	11	-X
12	12	-X
13	13	-X
14	14	-X
15	15	-X
16	16	-X
17	17	-X
18	18	-X
19	19	-X
20	20	-X

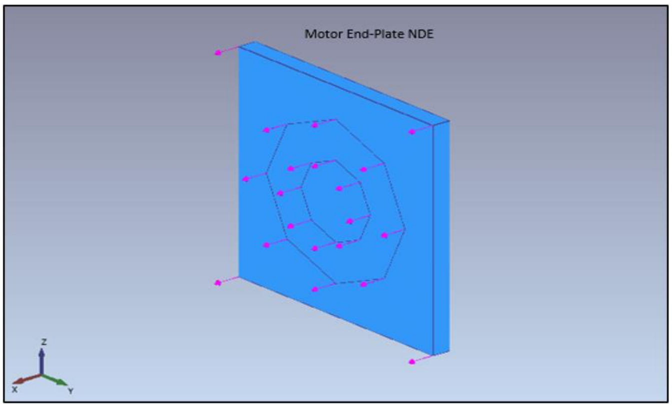


Fig. 5 - Geometry and DOF for Measurement 1

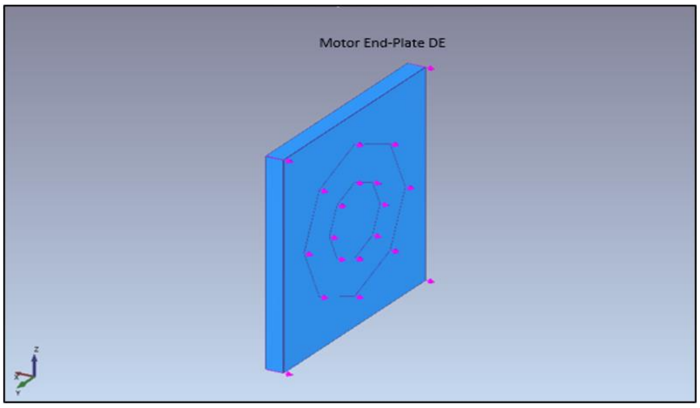


Fig. 6 - Geometry and DOF for Measurement 2

3. Results and Discussion

3.1 Vibration Measurement (VM)

From the overall vibration data in Table 3, it is shown that the overall vibration level for compressor side was within acceptable limit in accordance with ISO 10816-8 (Table 4). Meanwhile, the vibration level for motor side within danger limit is in accordance with OEM standard (Table 5) in which the highest value was recorded at both Motor Drive End (DE) and Non-Drive End (NDE) Axial direction (bearing casing). The vibration measurement was taken during the compressor load in the range of 44% to 73%. This condition occurred as the compressor kept on increasing in load and was unable to kept maintain at a single load during the process of vibration data acquisition due to plant operations restriction. From Table 3, the overall vibration values in velocity (mm/s RMS) were used to indicate the severity of a machine, as per ISO 10816. The highest overall value obtained from the vibration measurement was 8.54 mm/s RMS at the motor DE (Axial) and 8.47 mm/s RMS at the motor NDE (Axial), respectively. The highest vibration reading was depicted at the Axial bearing casing of both motor DE and NDE, in which the highest peak was located at the frequency of 112 Hz (18X running speed) with overall measurement of 8.54 mm/s RMS (Fig. 7) and 8.47 mm/s RMS (Fig. 8), respectively. The velocity spectrum for both sides of motor DE and NDE Axial direction plotted similar highest frequency peak at 112 Hz which was 18X of synchronous speed respectively.

Table 3 - Overall vibration value

DATE			07/04/2022	ACCEPTANCE CRITERIA AS PER ISO 10816		
LOAD (%)			44% 73%			
THROUGHPUT (MB)			50			
Description	**Direction**	**Unit**	**1030hrs**	**Fair**	**Alarm**	**Danger**
MOTOR NDE	Horizontal (Bearing Casing)	mm/s RMS	7.66	<5.60	5.60	8.20
	Vertical	mm/s RMS	4.54	<5.60	5.60	8.20
	Axial (Bearing Casing)	mm/s RMS	8.47	<5.60	5.60	8.20
	Horizontal (Sensor Probe)	mm/s RMS	1.68	<5.60	5.60	8.20
	Axial (Sensor Probe)	mm/s RMS	6.10	<5.60	5.60	8.20
MOTOR DE	Horizontal (Bearing Casing)	mm/s RMS	6.94	<5.60	5.60	8.20
	Vertical	mm/s RMS	2.63	<5.60	5.60	8.20
	Axial (Bearing Casing)	mm/s RMS	8.54	<5.60	5.60	8.20
	Horizontal (Sensor Probe)	mm/s RMS	2.30	<5.60	5.60	8.20
	Axial (Sensor Probe)	mm/s RMS	7.31	<5.60	5.60	8.20
COMPRESSOR DE	Horizontal	mm/s RMS	5.04	5.30	8.00	12.00
	Vertical	mm/s RMS	2.63	5.30	8.00	12.00
	Axial	mm/s RMS	2.37	5.30	8.00	12.00
COMPRESSOR NDE	Horizontal	mm/s RMS	4.06	5.30	8.00	12.00
	Vertical	mm/s RMS	3.38	5.30	8.00	12.00
	Axial	mm/s RMS	3.28	5.30	8.00	12.00

Table 4 - Compressor Acceptance Level in accordance with ISO 10816-8 [8]moto

Compressor system part	r.m.s vibration velocity values for horizontal compressors mm/s			r.m.s vibration velocity values for vertical compressors mm/s		
	Evaluation zone boundary			Evaluation zone boundary		
	A/B	B/C	C/D	A/B	B/C	C/D
Foundation	2,0	3,0	4,5	2,0	3,0	4,5
Frame (top)	5,3	8,0	12,0	5,3	8,0	12,0
Cylinder (lateral)	8,7	13,0	19,5	10,7	16,0	24,0
Cylinder (rod)	10,7	16,0	24,0	8,7	13,0	19,5
Dampers	12,7	19,0	28,5	12,7	19,0	28,5
Piping	12,7	19,0	28,5	12,7	19,0	28,5

Note:
For piping value above evaluation zone boundary C/D, see Table 1, Note 3.

Table 5 - Motor acceptance level in accordance with OEM standard

ACCEPTANCE CRITERIA		
FAIR	ALARM	DANGER
Up to 5.60	5.60	8.20

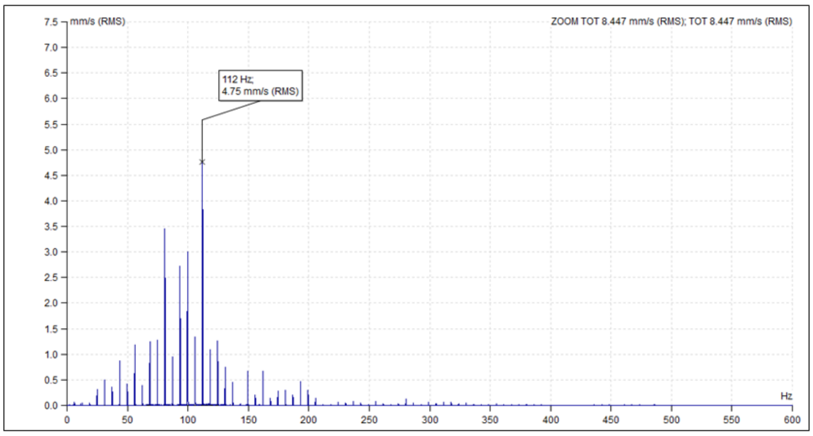


Fig. 7 - Velocity spectrum of Motor NDE Axial - highest peak plotted at 112 Hz

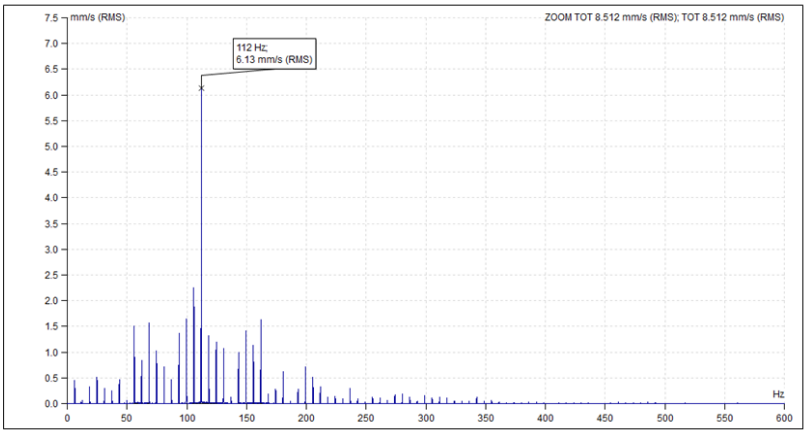


Fig. 8 - Velocity spectrum of Motor DE Axial - highest peak plotted at 112 Hz

The acceleration spectrum for both motors DE and NDE plotted the harmonics Rotor Bar Frequency (RBF) and sideband 2xLF (100 Hz) with low amplitude as shown in Fig. 9 and Fig. 10. The RBF was plotted at low amplitude, which indicates irregularities but not the severity of the motor.

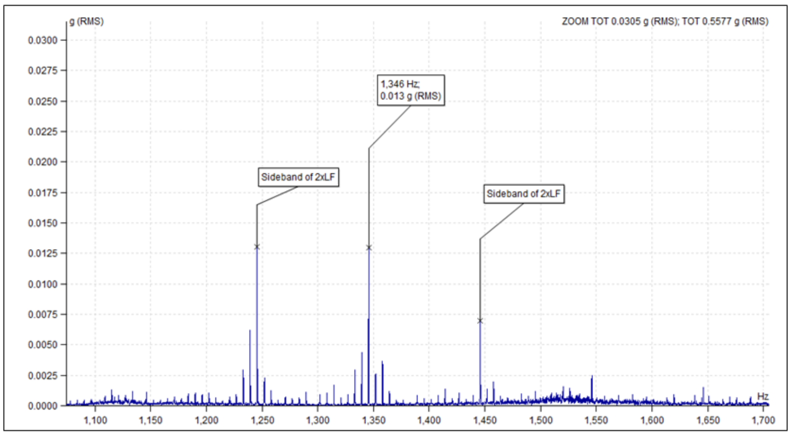


Fig. 9 - Acceleration spectrum of Motor NDE Axial - Rotor Bar Frequency (RBF) with sideband of 2xLF

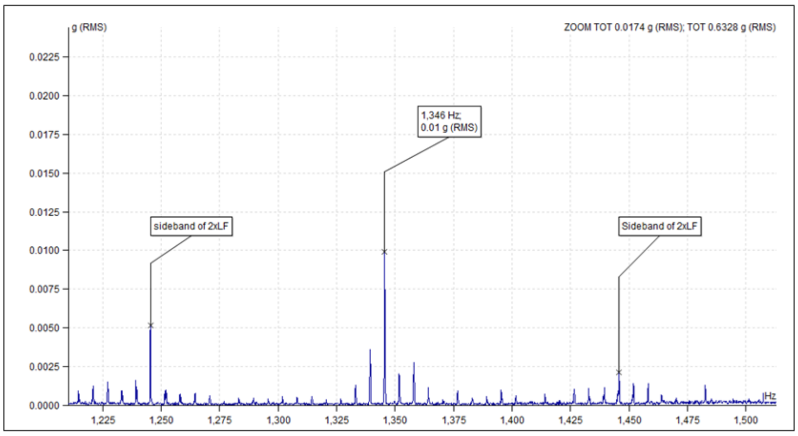


Fig. 10 - Acceleration spectrum of Motor DE Axial - Rotor Bar Frequency (RBF) with sideband of 2xLF

3.2 Operational Modal Analysis (OMA)

Fig. 11 and Fig. 12 show the SVD (Singular Value Decomposition) plot for each measurement. The usual evaluation for the data quality was done by calculating the signal to noise (S/N) ratio. The signal to noise ratio (S/N) plays an important role to check the quality of modal data where it compares the level of desired signal to the background noise. This desired signal will provide meaningful information of the modal data as compared to the unwanted signal. In this study, the value for S/N was observed from SVD plot for Measurement 1 and Measurement 2, showing ratio of 80 dB and 70 dB. The values obtained indicate a good S/N which is around 70 dB (according to [13]).

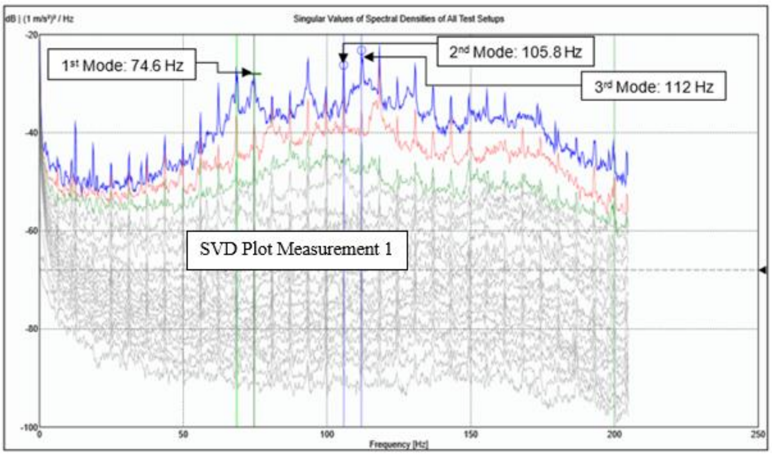


Fig. 11 - SVD plot for Measurement 1

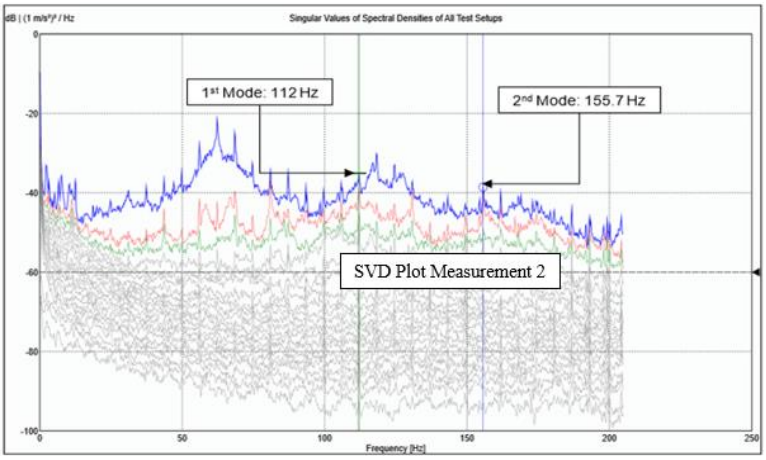
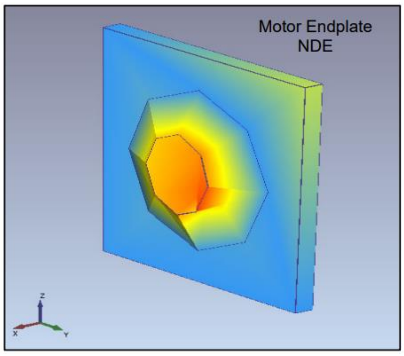
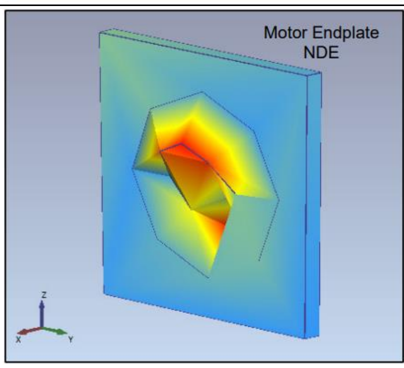


Fig. 12 - SVD plot for Measurement 2

Frequencies from Fig. 11 and Fig. 12 were found by using peak picking method on the FDD algorithms. The first mode was found at the frequency of 74.6 Hz, the second one at 105.8 Hz and the last one at 112 Hz on SVD plot from Measurement 1 (Motor Endplate NDE). On measurement 2 (Motor Endplate DE), the modes were extracted from 112 Hz and 155.7 Hz. From OMA measurement, it will provide multiple range of natural frequencies and mode shapes depending on the frequency range that has been set. In this case, several modes and natural frequencies were obtained as shown in both figures, and it shows that 112 Hz was the most dominant in both OMA measurements (Measurement 1 and Measurement 2). This matches with the dominant frequency obtained in vibration spectrum. Thus, this indicates the resonance occurrence. As mentioned in methodology where the measurement was performed simultaneously using 20 accelerometers in one data set, we believe that the system has a greater response at 112 Hz as compared to other natural frequencies found from OMA as illustrated in the following Table 6 and Table 7.

Table 6 - Dynamic characteristics for Measurement 1

No. of Mode	Natural Frequency (Hz)	Mode Shape
1st mode	74.60	
2nd mode	105.80	

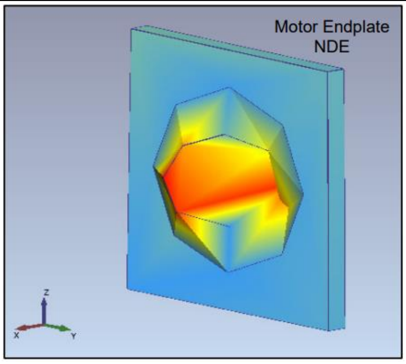
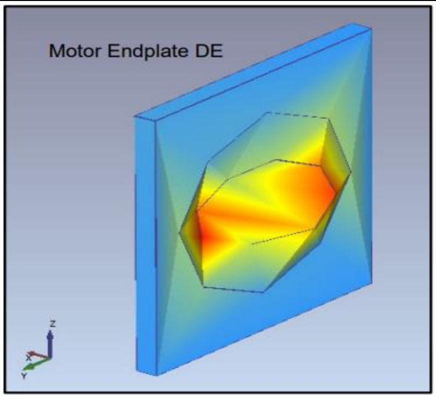
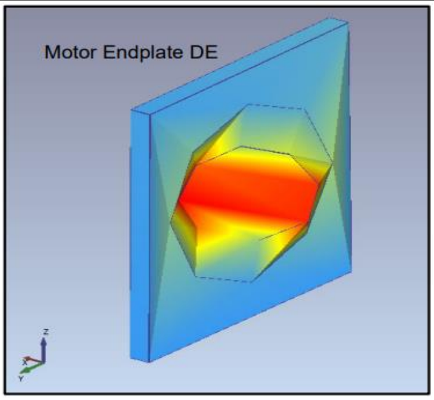
3rd mode	112.00	

Table 7 - Dynamic characteristics for Measurement 2

No. of Mode	Natural Frequency (Hz)	Mode Shape
1st mode	112.00	
2nd mode	155.70	

From the mode shape acquired from the OMA analysis, the load from compressor had excited the resonant frequency of 112 Hz of the whole motor - compressor assembly, but not at one particular point. This is because OMA measurement was performed simultaneously at each motor endplate (NDE & DE). As per findings from OMA for Measurement 1 and Measurement 2, the resonance can clearly be seen at both motor endplates NDE and DE side. It is suspected that these are the weakest areas as compared to the other components of the system at the frequency of 112 Hz. The sources of vibration are suspected to come from the compressor load, which causes the components of the motor - compressor assembly become weak at 112 Hz to resonate, hence causing the torsional movement. The torsional movement would trigger the torsional mode of the system.

Nonetheless, there were several limitations to this study. The torsional movement suspected of causing the torsional mode was not verified, and a detailed torsional analysis is required to obtain more valid information for verification and validation purposes. A frequency-domain operational deflection shape (ODS) could also be a great course of action to add to this research. This will be advantageous in obtaining the deflection of the system at an interest of specific forcing frequency, which in this case is the 112 Hz. Thus, substantiate with the dynamic characteristics obtained from OMA measurement.

4. Conclusion

The overall vibration value from vibration measurement is a good indicator and very reliable for predictive maintenance, while OMA, on the other hand, is successfully conducted to verify the presence of resonance. Both measurement techniques share the consistency in providing similar frequency (112 Hz) which indicates the resonance occurrence. Preference of OMA is also supported in various ways of this paper to support its reliability in this industry. However, future studies are proposed to improve the verification by using other computation algorithms and methods. SSI is deemed to be one of the famous algorithms in modal parameters computation. Hence, future research can be conducted by focusing on this method. Besides, FE method is also common as data validation in vibration analysis. An appropriate scope addition shall be served by extending verification mode using numerical model in the future.

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