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TURBOMACHINERY LABORATORY
TEXAS A&M ENGINEERING EXPERIMENT STATION

TEXAS A&M UNIVERSITY

THE IMPORTANCE OF STRUCTURAL MODAL ANALYSIS IN 2 POLES INDUCTION MOTORS FOR LNG APPLICATION

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Presenter/Author bios

Francesco Meucci
Is working as Electrical System Design Engineer at GE Oil&Gas, Florence, Italy.

He joined GE in 2012 through a technical leadership program gaining cross functional background on electrical and mechanical area of concern. From 2014 he's working on multiple projects with special focus on rotordynamic analyses and troubleshooting linked with electrical machines and structures.
He graduated in Electrical and Automation Engineering at University of Florence, Italy in 2011.

Francesco Capanni
Is the Senior Engineer for the Advanced Train Integration Team at GE Oil&Gas, Florence, Italy

In this role, being owner/contributor for several Design Practice, Design Template and Design Tools. He provides leadership and support for structural design without the Turbomachinery (TMS) and Downstream (DTS) organizations. His areas of expertise is structural design for On/Off Shore application with particular attention to the dynamic behavior of rotating machines supporting structures

Niccolò Spolveri
Is currently working as Lead Electrical System Design Engineer at GE Oil&Gas, Florence, Italy.

He joined GE O&G in 2012 as Lead Electrical System Engineer working in requisition projects. He is responsible for the integration of VSDS (Variable Speed Drive System) for complex projects mainly for LNG (Liquefaction Natural Gas) applications.
He got Master Degree in Electric Engineering at University of Bologna, Italy in 2009.

Stefano Rossin
Is Chief Consulting Engineer for Auxiliary systems at GT Team in GE Oil&Gas, Florence, Italy.

He lead projects and initiatives for Auxiliary System Engineering with broad scope and high impact to the business, he is responsible for assignments with long-term business implications, contributing to the overall strategy, providing consultation to engineering management and supplier organizations in structural dynamics, solid mechanics and fluid systems design

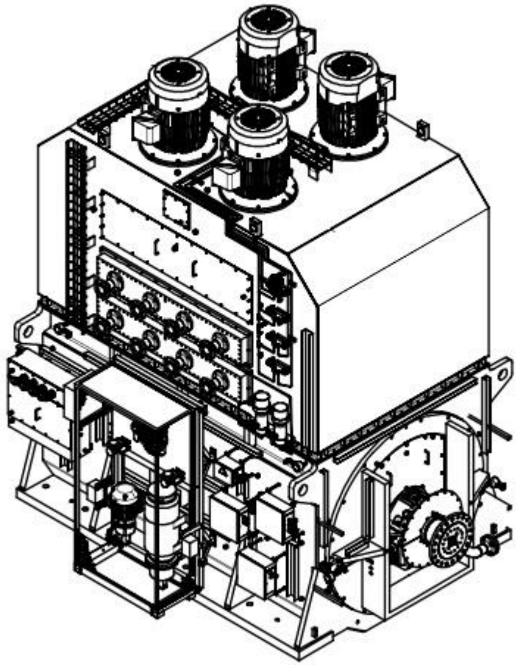
Abstract

High vibrations due to 2X electrical frequency excitation on bearing housings and frame occurred on a 20MW 2-Pole Induction Motors during acceptance tests in manufacturer workshop in stand alone configuration. Modal Analysis results in Factory Acceptance Tests (FAT) configuration were confirmed by Operating Deflection Shape (ODS) and Experimental Modal Analysis (EMA - Hammer test) leading to supports redesign. To mitigate the risk of high vibration in string test bench and customer site, validating also the present motor design, a further Modal Analysis were executed using the experimental data collected during the FAT to drive mode selection.

Motor General Data

- **Liquefaction Natural Gas Train**

  HD Gas Turbine + 3MCL + MCL + 2BCL + Helper Motor
- 2-pole Induction Motor
- Rated Power 20 MW
- Rated Speed 3600 rpm
- Motor fed by VSD (Variable Speed Driver)

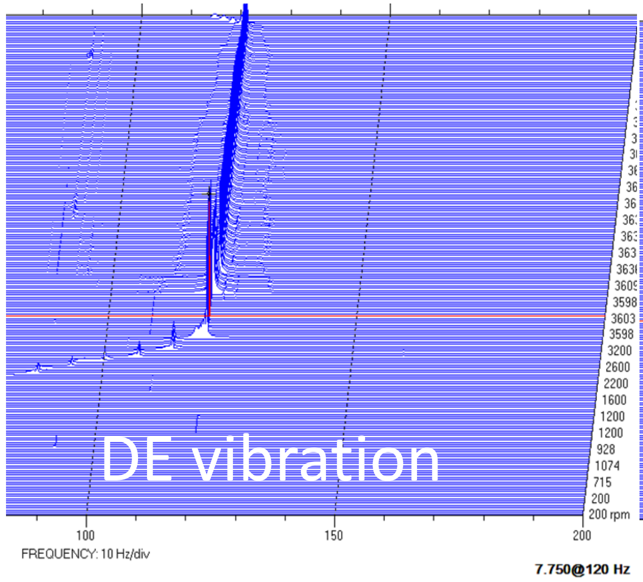


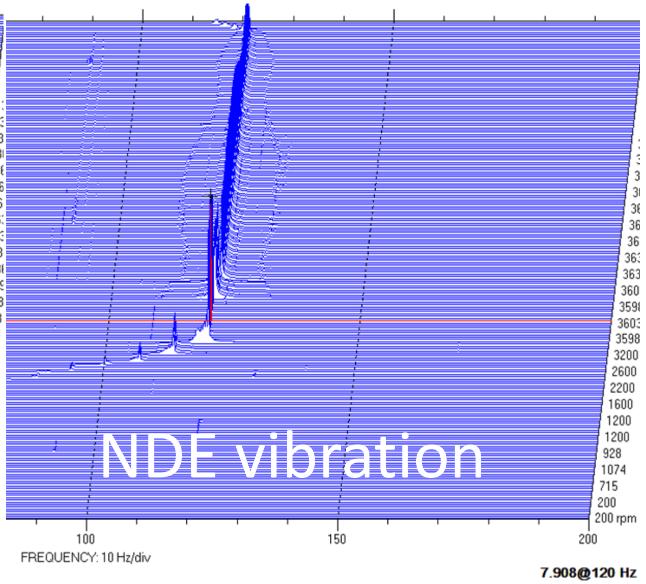
Problem Statement

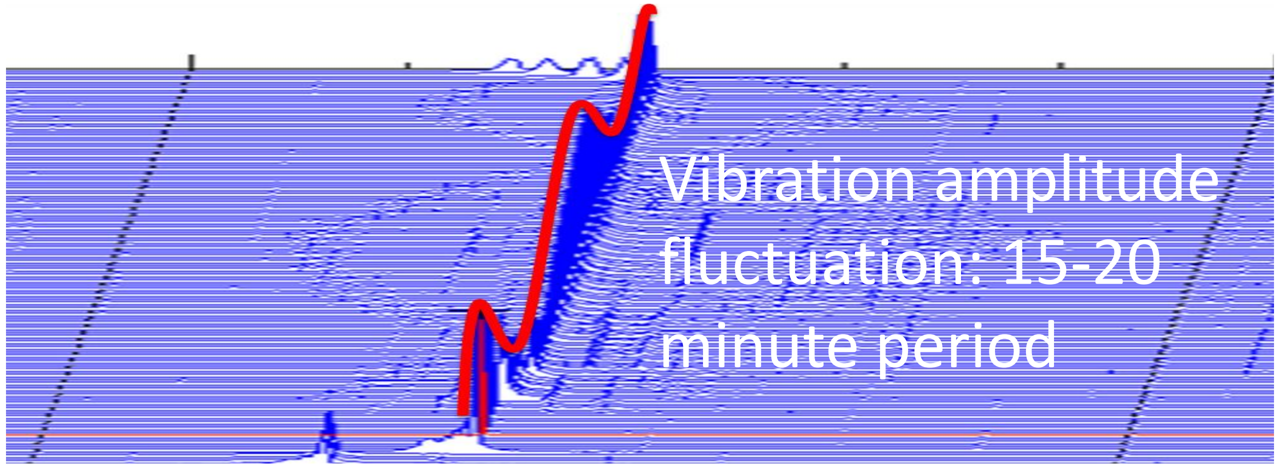
- Horizontal DE vibration probe: 7.75 mm/s RMS @ **120Hz**
- Horizontal NDE vibration probe: 7.91 mm/s RMS @ **120Hz**
- API541 acceptance criteria at factory acceptance test (FAT) was 1.8 mm/s RMS unfiltered;

<u>**Main finding:**</u>

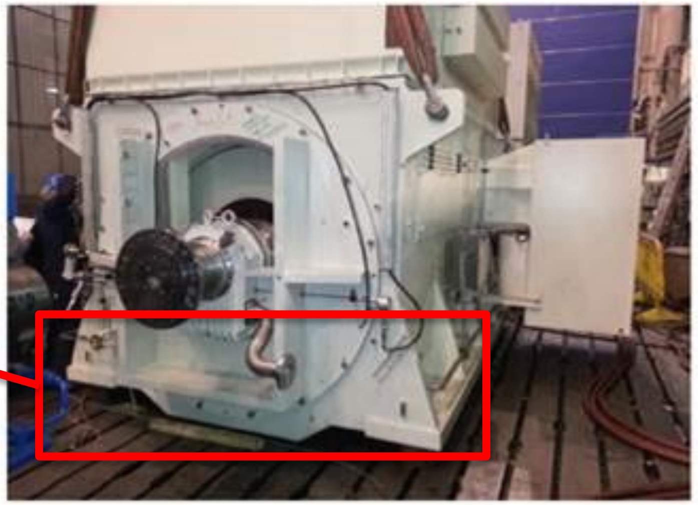
- Raised feet design: on test bench for FAT the motor was supported with **6 steel cylinders per side.**
- Waterfall @ 120Hz presents an amplitude fluctuation with a long period (about 15-20 min)







Raised feet Motor Design



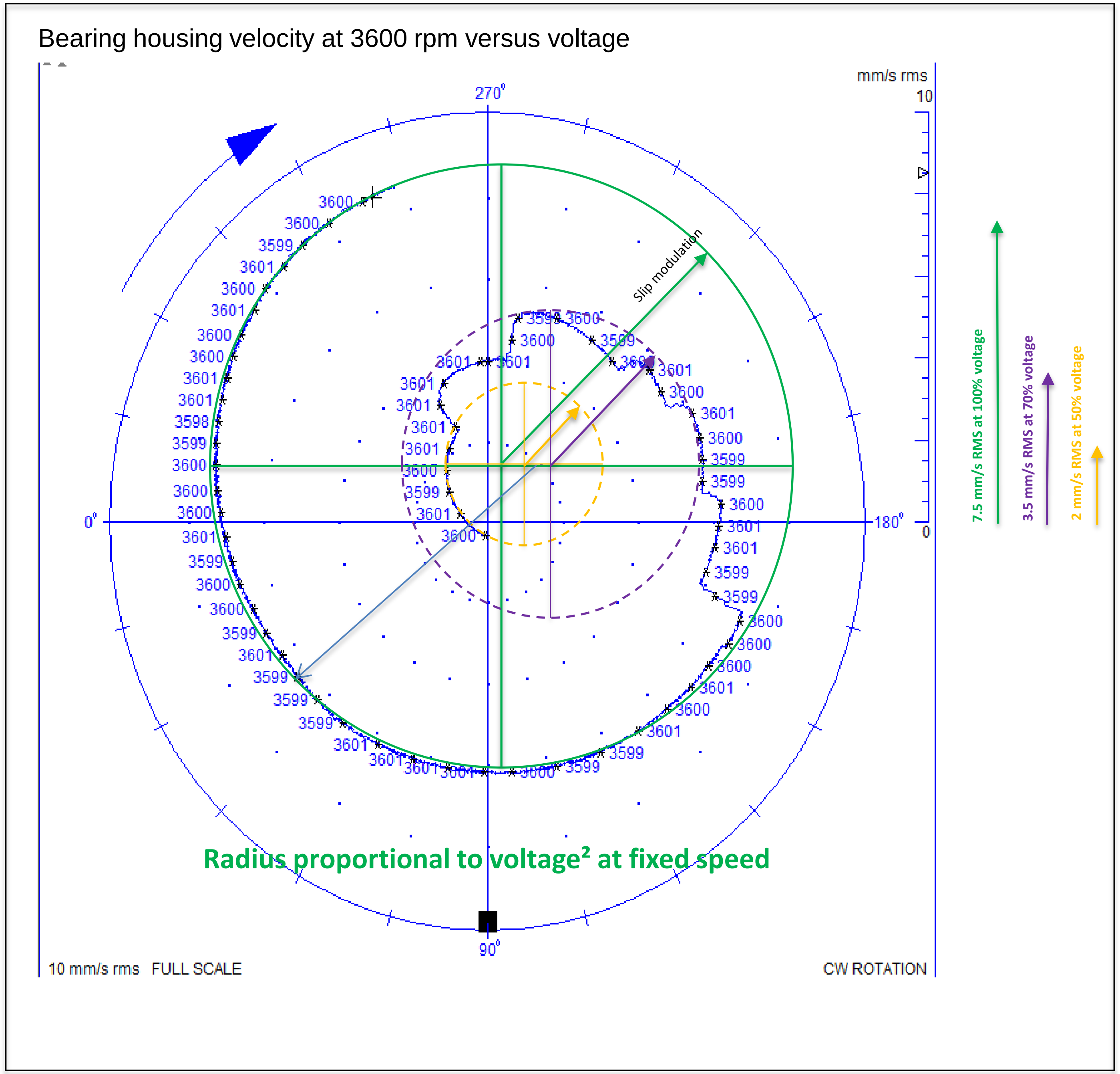
Analysis of the Problem

According previous manufacturer experience and available literature, high vibration with long period fluctuation have been correlated with the 2 times line frequency (electrical) vibration interaction with the 2 times rotational frequency (mechanical).

In such a case vibrations are very sensitive to:

- Flux amplitude;
- Motor's foot flatness (Soft-foot);
- **Frame and baseplate stiffness;**
- Amount of air gap between stator and rotor;
- Eccentricity of the rotor/stator;

FE Modal Analysis performed to investigate the impact of foundation stiffness on frame resonant frequencies.



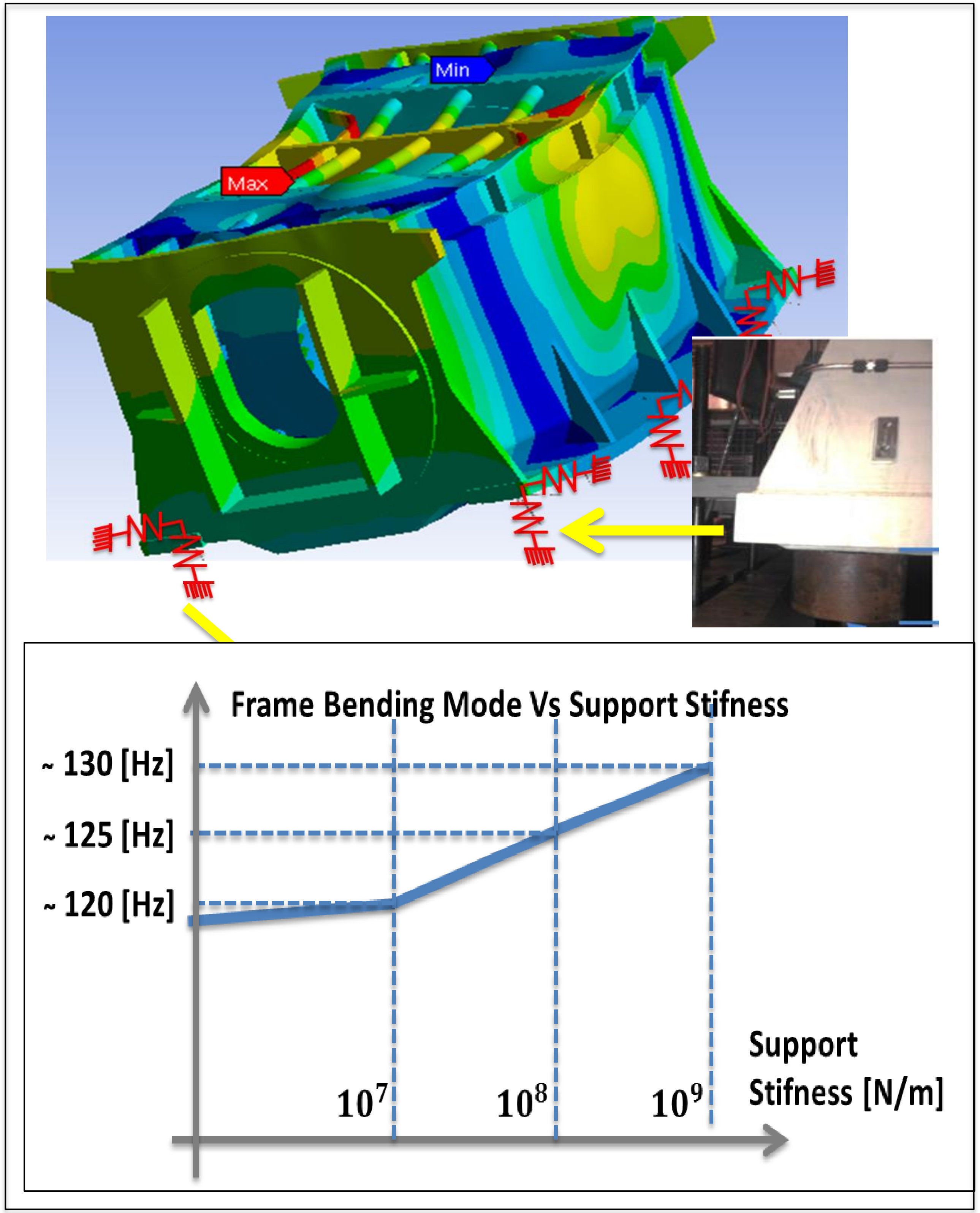
7.5	**mm/s**	**@ 100%**	**Flux (Voltage)**
3.5	**mm/s**	**@ 70%**	**Flux (Voltage)**
2	**mm/s**	**@ 50%**	**Flux (Voltage)**

Analysis of the Problem

Findings from FEA Anlysis:

- Motor frame resonance really sensitive to supports stiffness
- The motor frame resonance at 120 Hz is induced by a supports low horizontal stiffness (~10^7 N/m)

Redesign of the test bench, increasing stiffness in both horizontal and vertical direction should move the system resonant frequency far away from the 2X Excitation Source



Analysis of the Problem

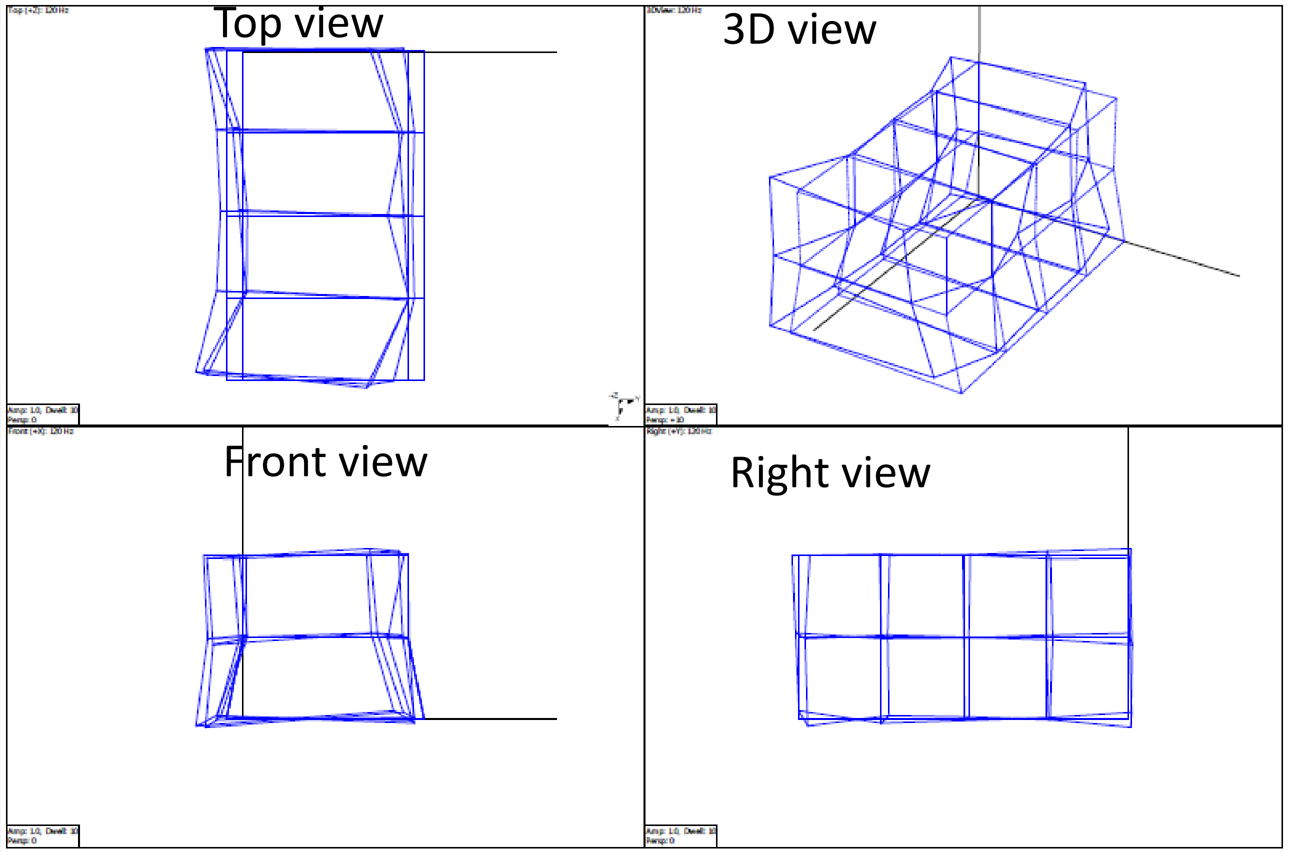
Findings from Vibration Measures:

- A mode @ 120 Hz was confirmed by hammer test results (EMA)
- ODS confirms the dynamic response of the motor is dominated by the mode @ 120 Hz.

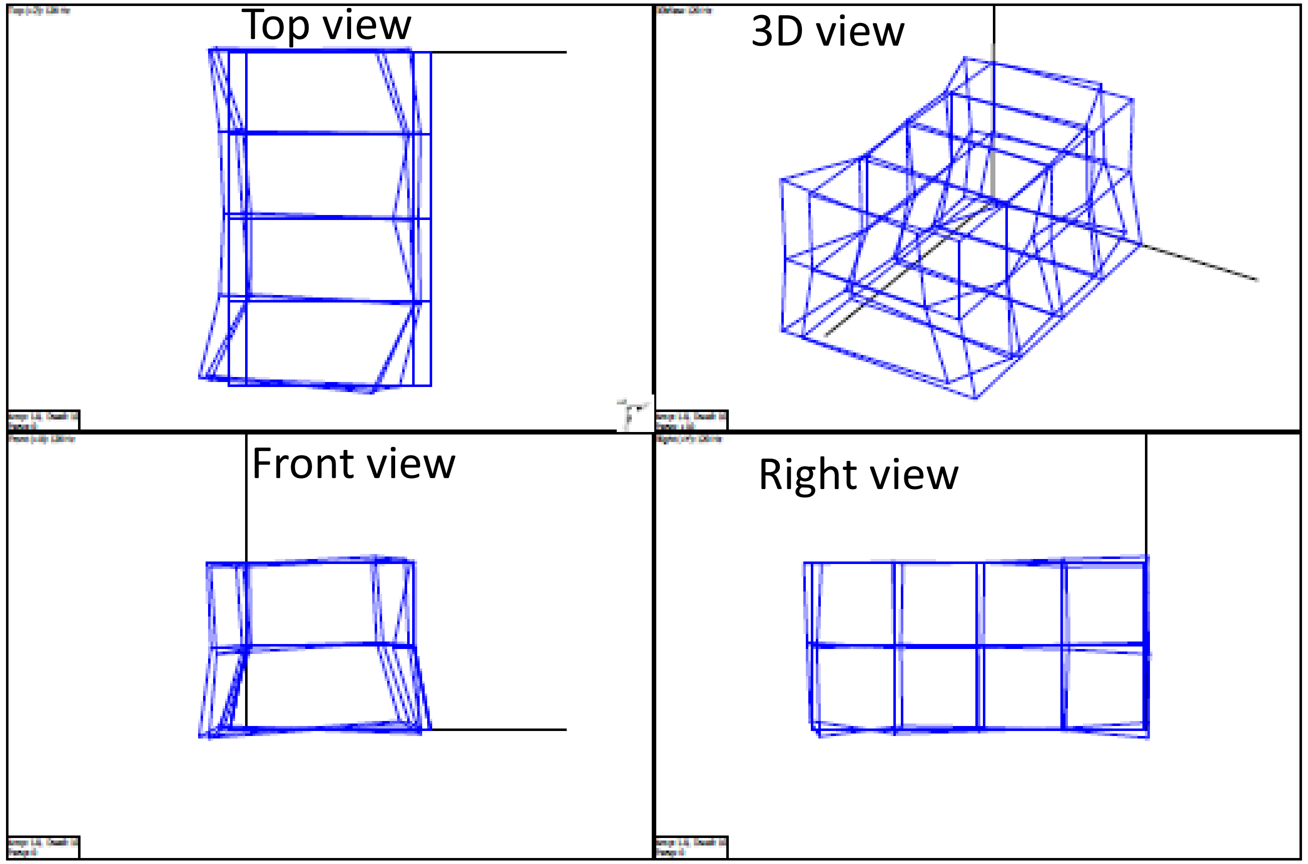
EMA results

Mode	Frequency (Hz)	Damping* (%)	Amplification factor Q
1	10.8	1.3	65
2	26.1	0.9	45
3	31.0	0.7	35
4	35.1	1.1	55
5	41.0	2.2	110
6	58.5	1.1	55
7	107	0.9	45
8	120	1.1	55

ODS Result – Deformed shape @ 120 Hz



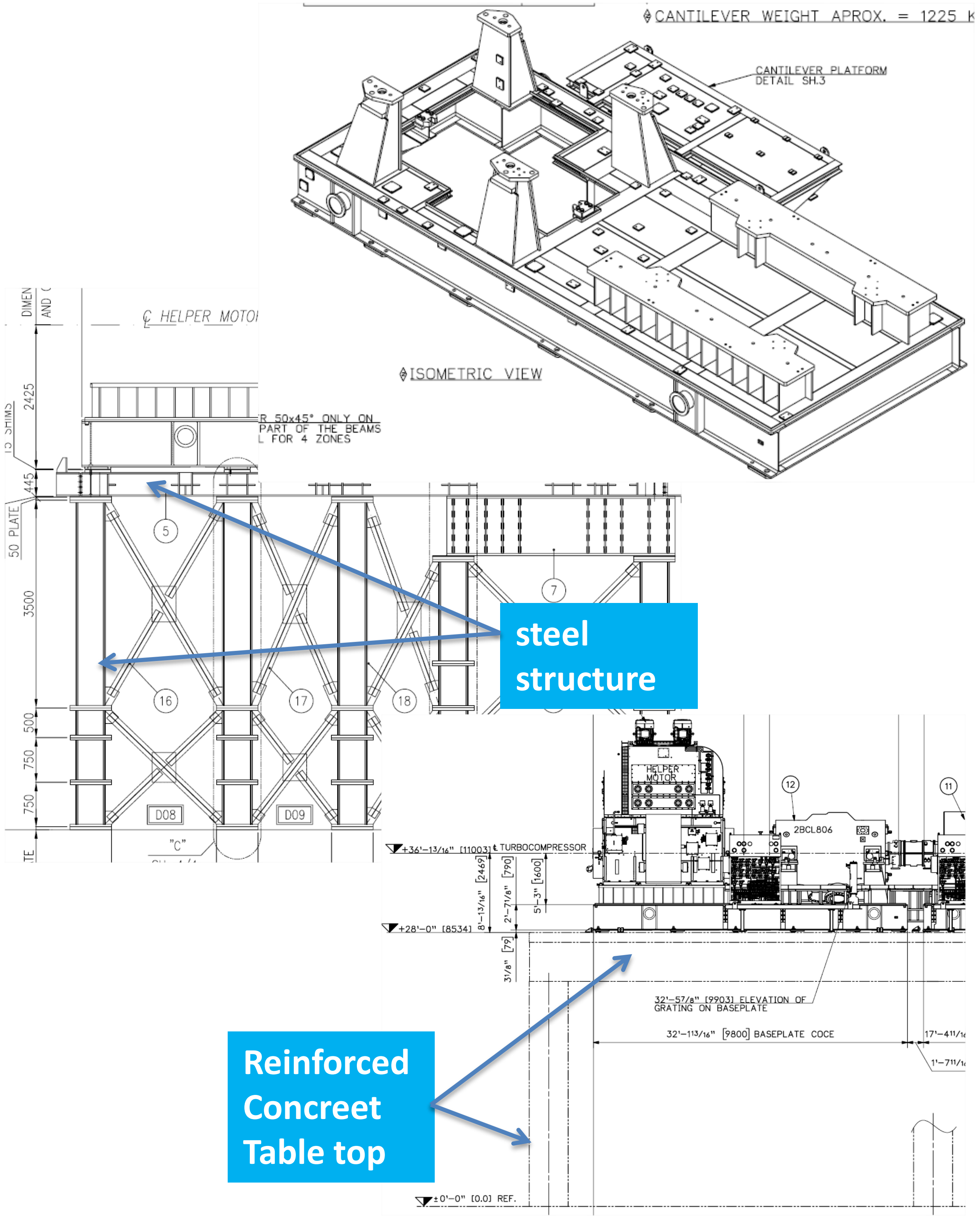
Bending mode @ 120 Hz



Analysis of the Problem

String Test and Customer Site have a different arrangement for motor feet supports and fixation:

- Motor mounted on a dedicated baseplate
- During string test: LNG train mounted on a elevated steel structure
- At site: LNG train installed on a reinforced concrete table top



Risk of high vibration to be investigated considering the above scenario through FE dynamic analysis

Analysis of the Problem

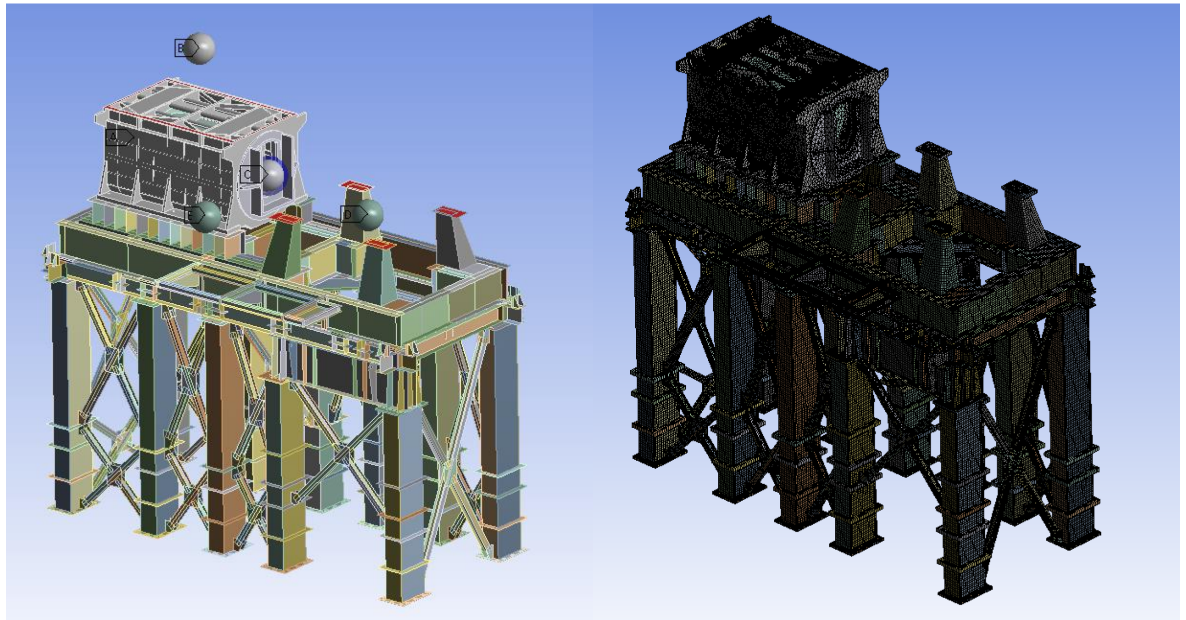
FEA on String Test configuration

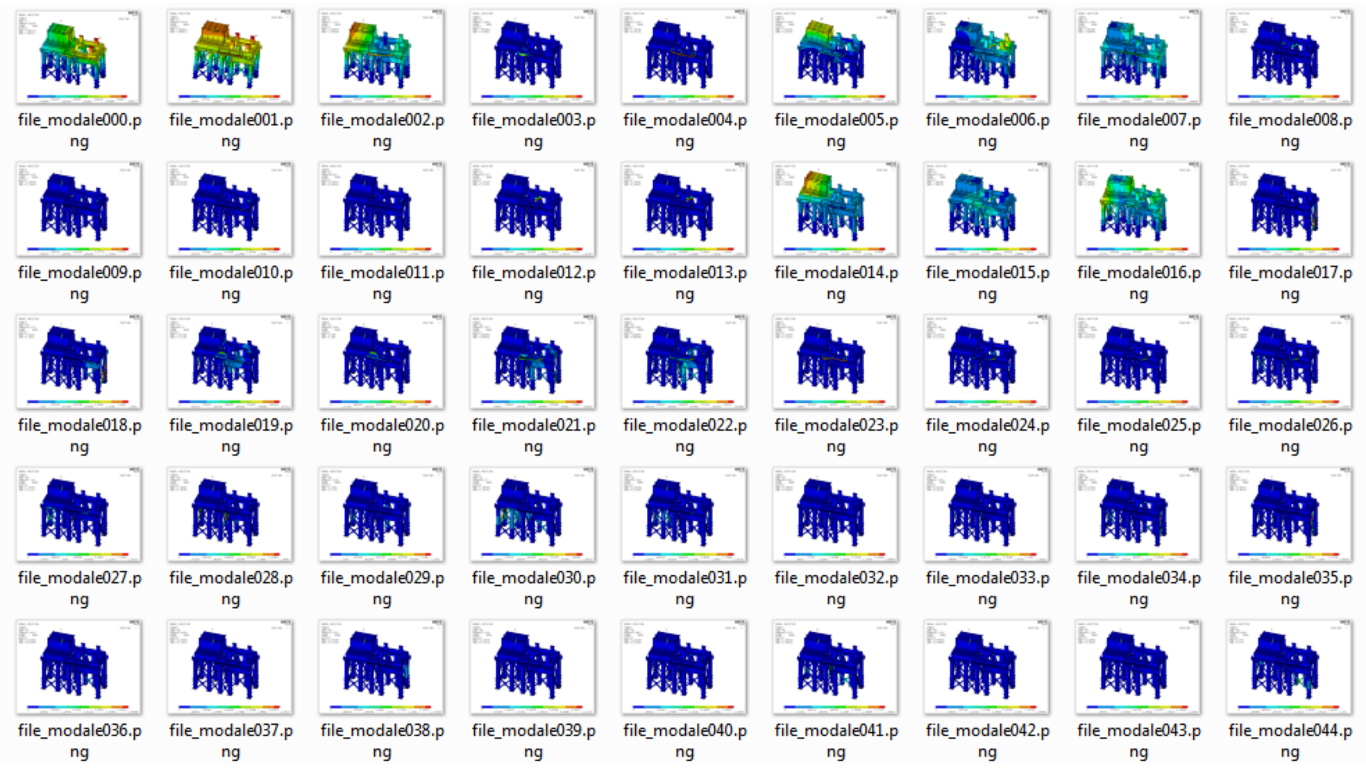
- For the whole structure (Motor + baseplate + String test bench) the modal analysis founds more than **235 Modes in the Frequency range 0-150 Hz**.

- Modes classification by a dedicated harmonic analysis was not possible because the *lack of detailed information** about the exciting force;

- Modes classification has been carried out with a different approach;

*Motor built by a supplier.





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Analysis of the Problem

Screening criteria:

1. Only the modes in proximity of Machine Operating speed Range (1X and 2X) have been considered with a separation margin of 15%

2. Modes have been classified as Local or Global elaborating the modal coordinates values. For each m mode with it's ϕ_{jm} DOF's translation, it is possible to calculate:

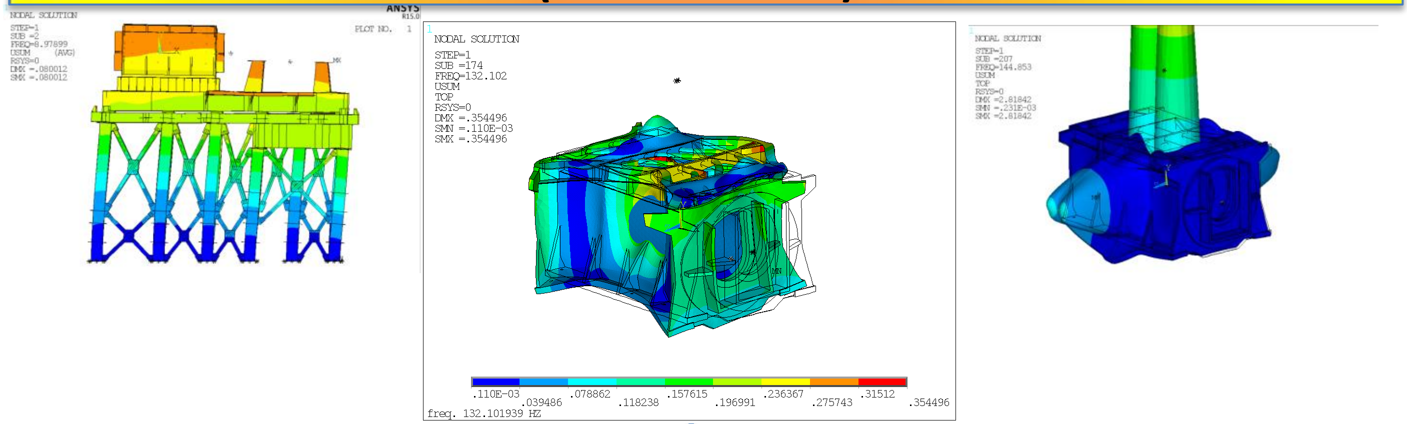
$$Global/Local\ Indicator_{m} = \frac{\sum_j |\phi_{jm}|}{N_j \cdot \max_j\{|\phi_{jm}|\}}$$

Where N_j is the number of DOF's considered and j is the j-th DOF.

235 modes
in the Frequency range 0-150 Hz.

117 modes
in the Frequency range of interest:
Machine Operating speed Range: 95-101% ±15%
48.5 - 70 Hz and 96 - 140 Hz.

Global Modes (hi)	Global/Local Modes (Intermediate)	Local Modes (Low)



21 modes
identified as Casing modes

Analysis of the Problem

21 modes

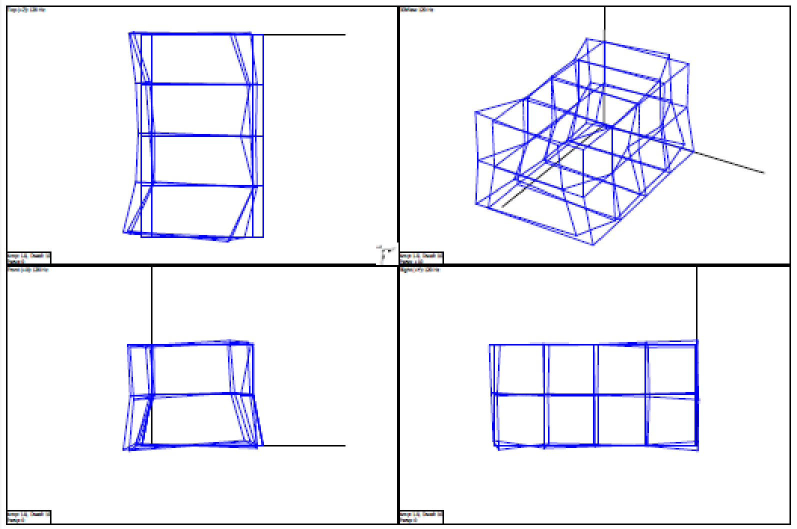
Classified as per displacement in the transversal direction

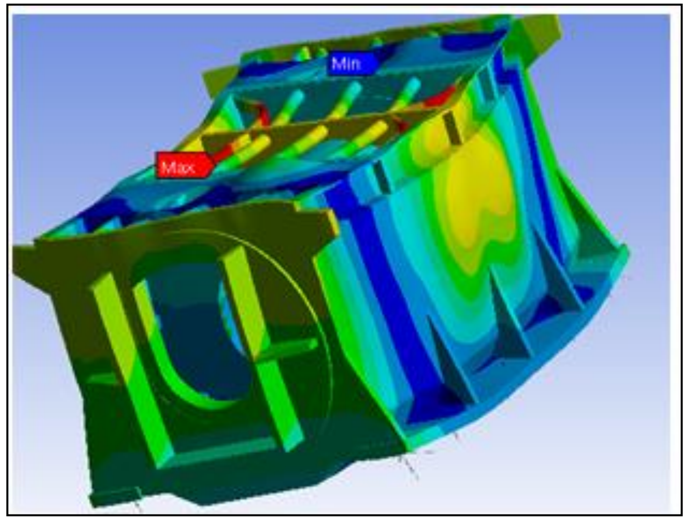
Screening criteria:

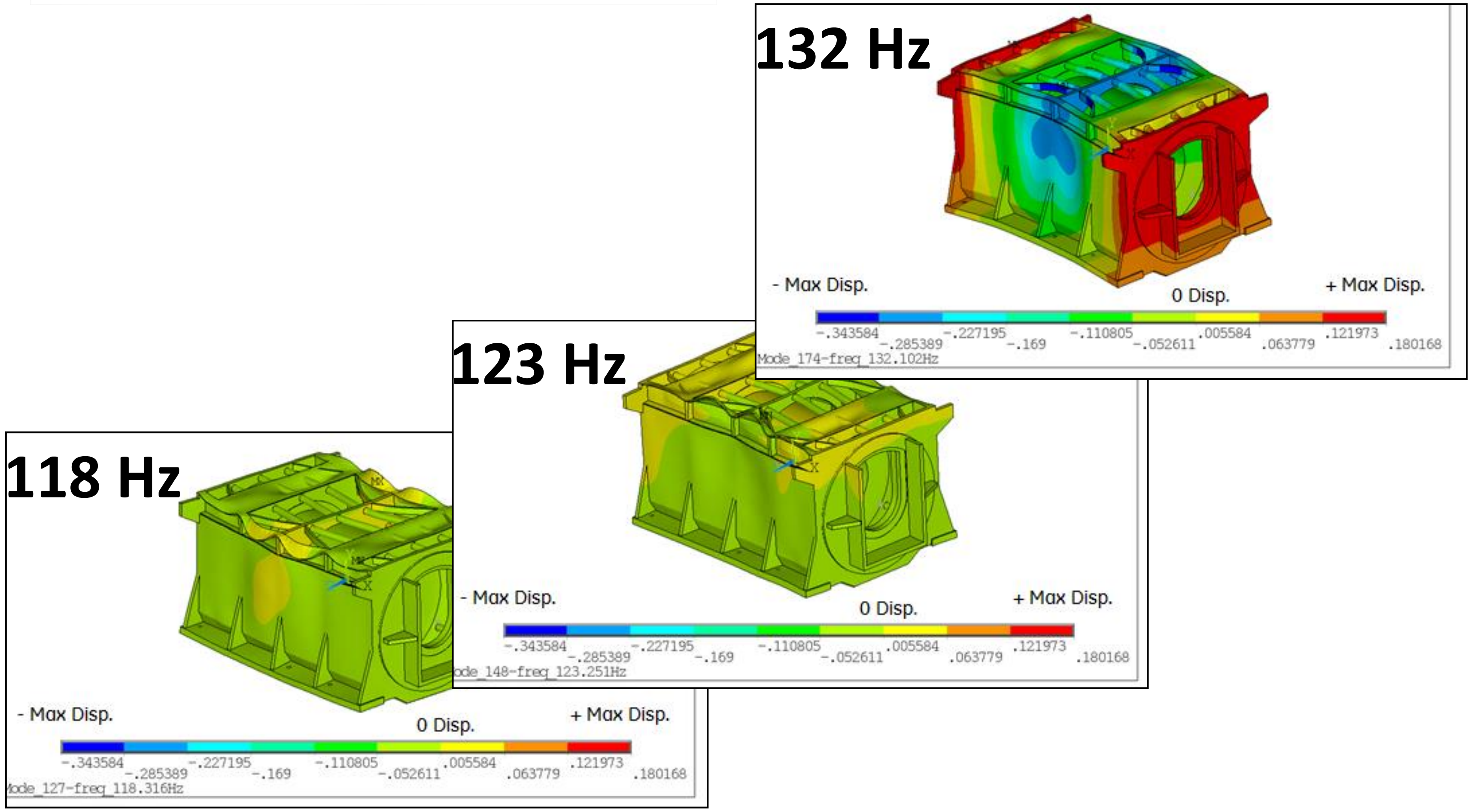
Considering the EMA and ODS results **a Horizontal bending modes is dominating the dynamic response** when the casing is excited by the 2x excitation.

1. The 21 modes were plotted considering only the **transversal direction**.

2. The 21 modes were classified as per the **maximum displacement** measured considering the fact that all the mode shapes are scaled in FE tool by the mass matrix.





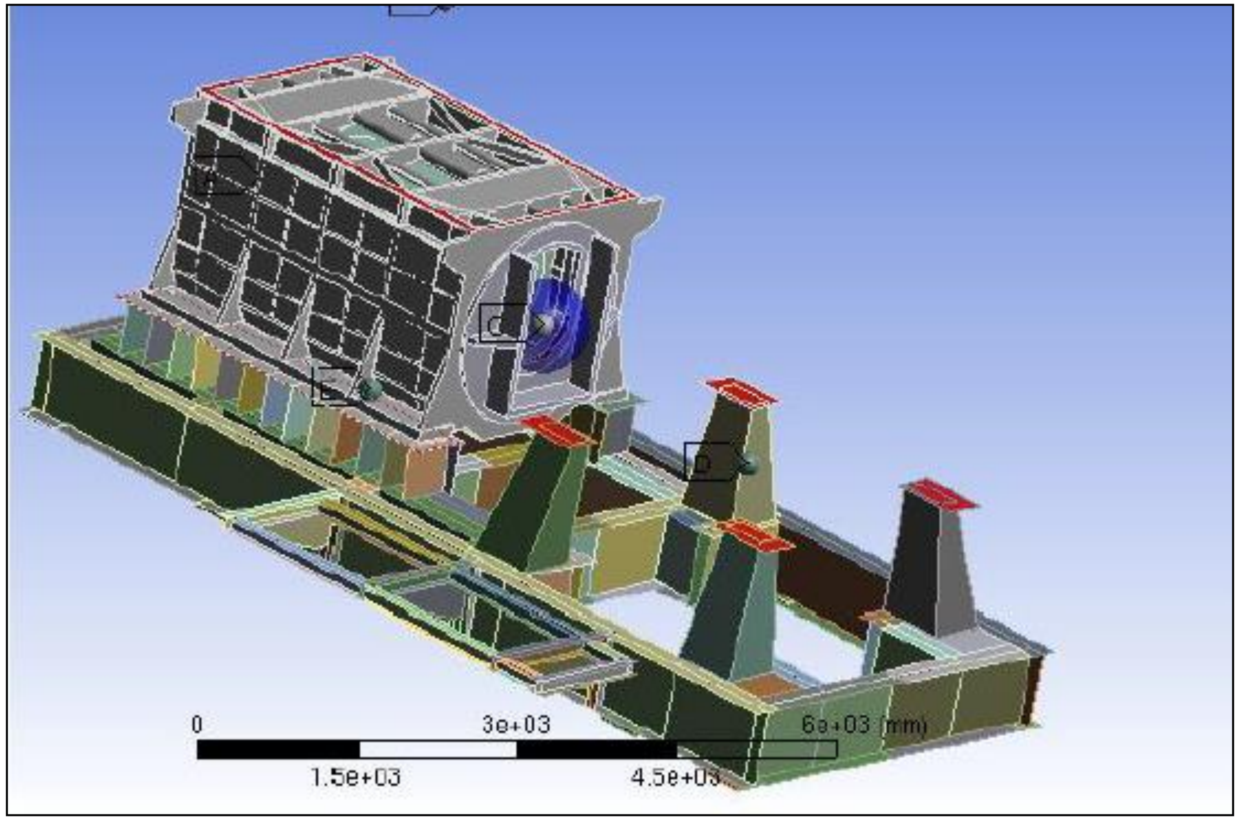


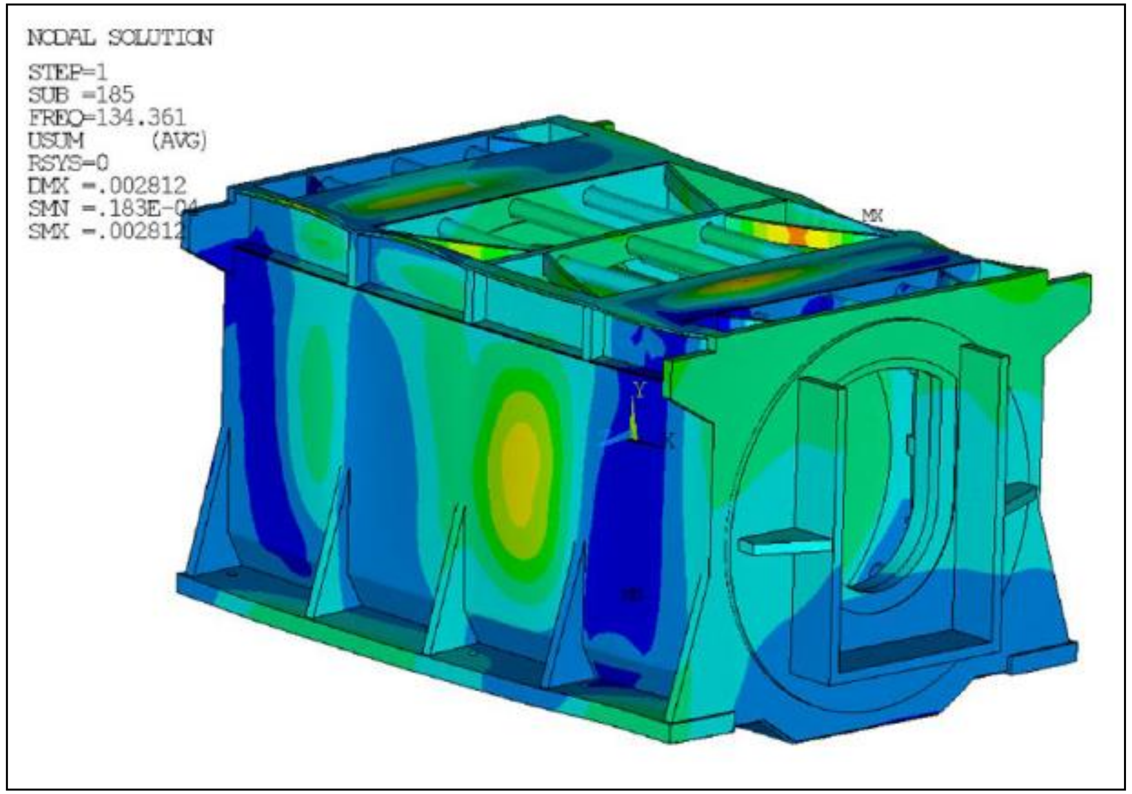
Analysis of the Problem

FEA on Site Configuration

- In comparison with the String test bench case according to customer data the reinforced concrete foundation at site introduces an higher stiffness below the baseplates sub-sole plates.

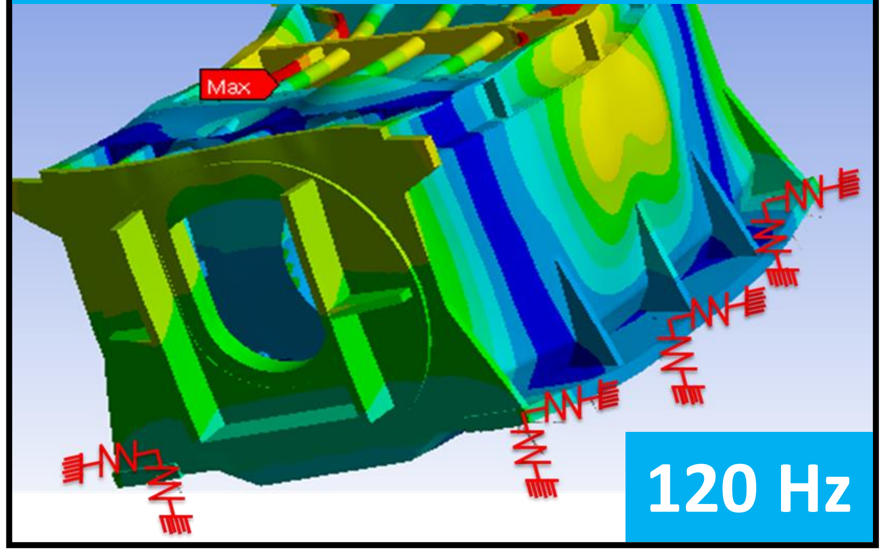
- FEA in the operating range of interest pointed out a similar dynamic behavior.





Conclusion and Recommendation

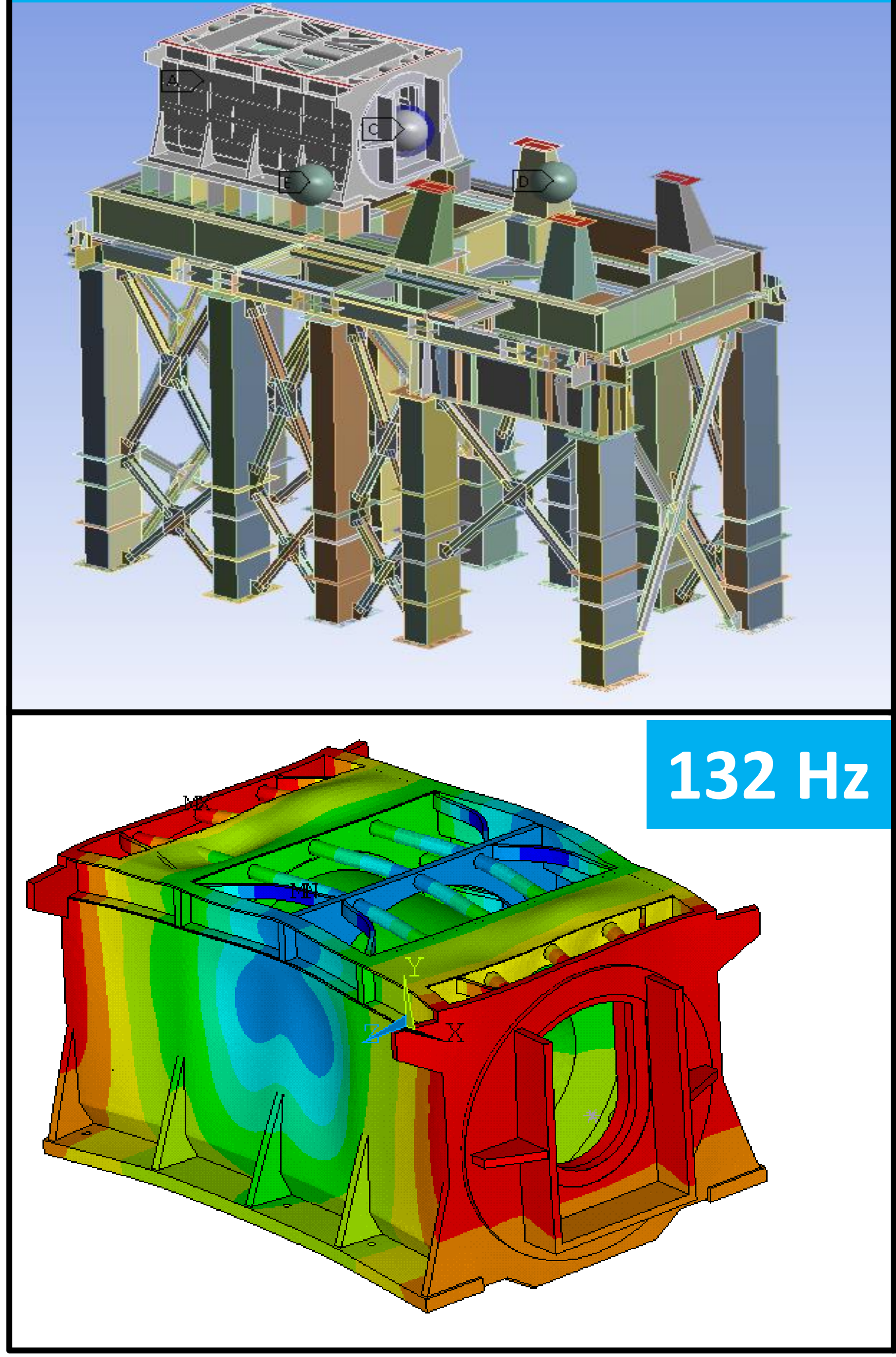
FAT Configuration



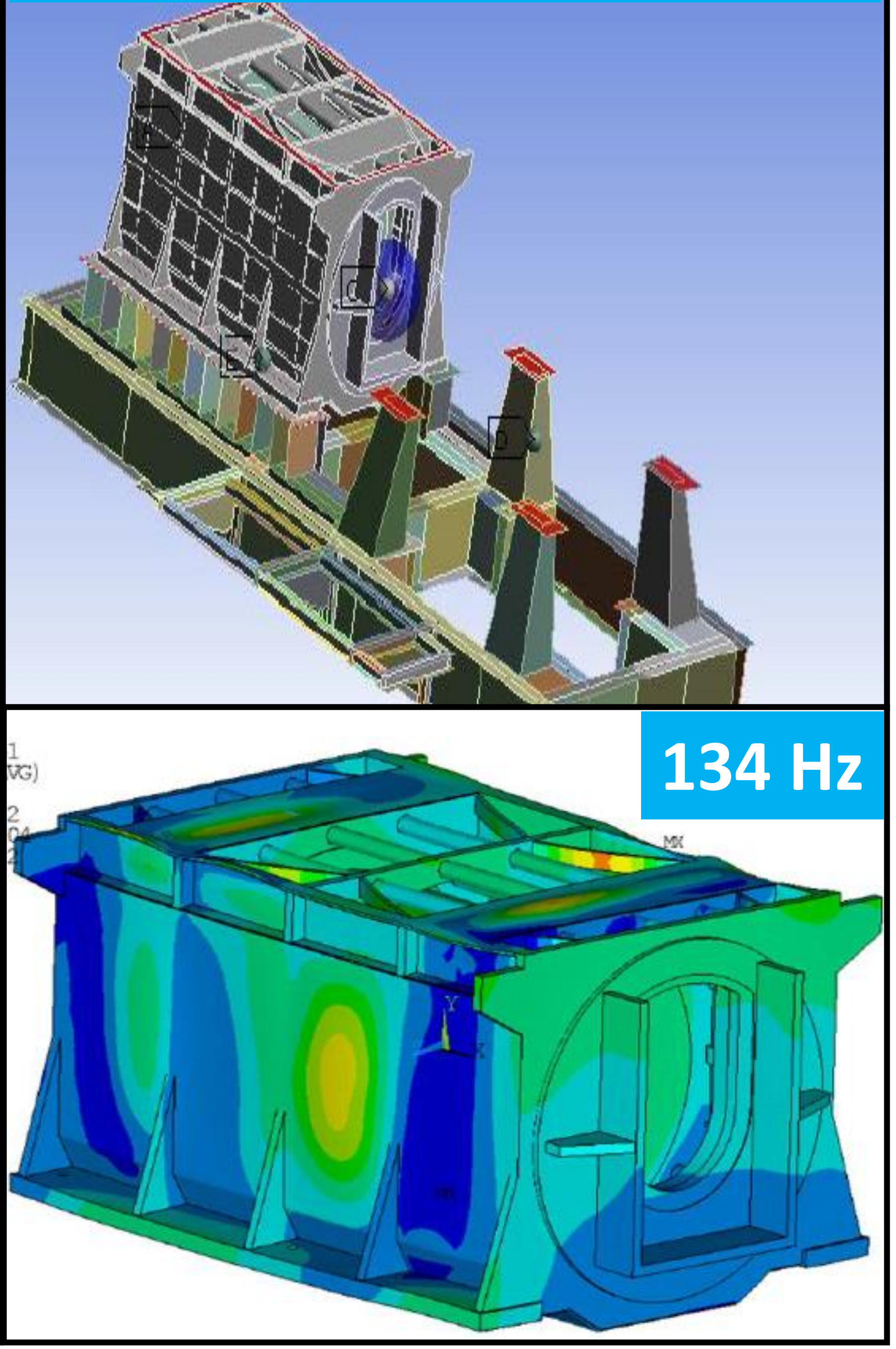
High vibration recorded during the motor FAT have been induced by the typical 2x exciting force for 2-poles Asynchronous motor and amplified by a not proper supports stiffness.

If the motor is well supported as in in String Test and Site Configurations vibration amplitude is expected to be within the limit

String Test Configuration



Site Configuration

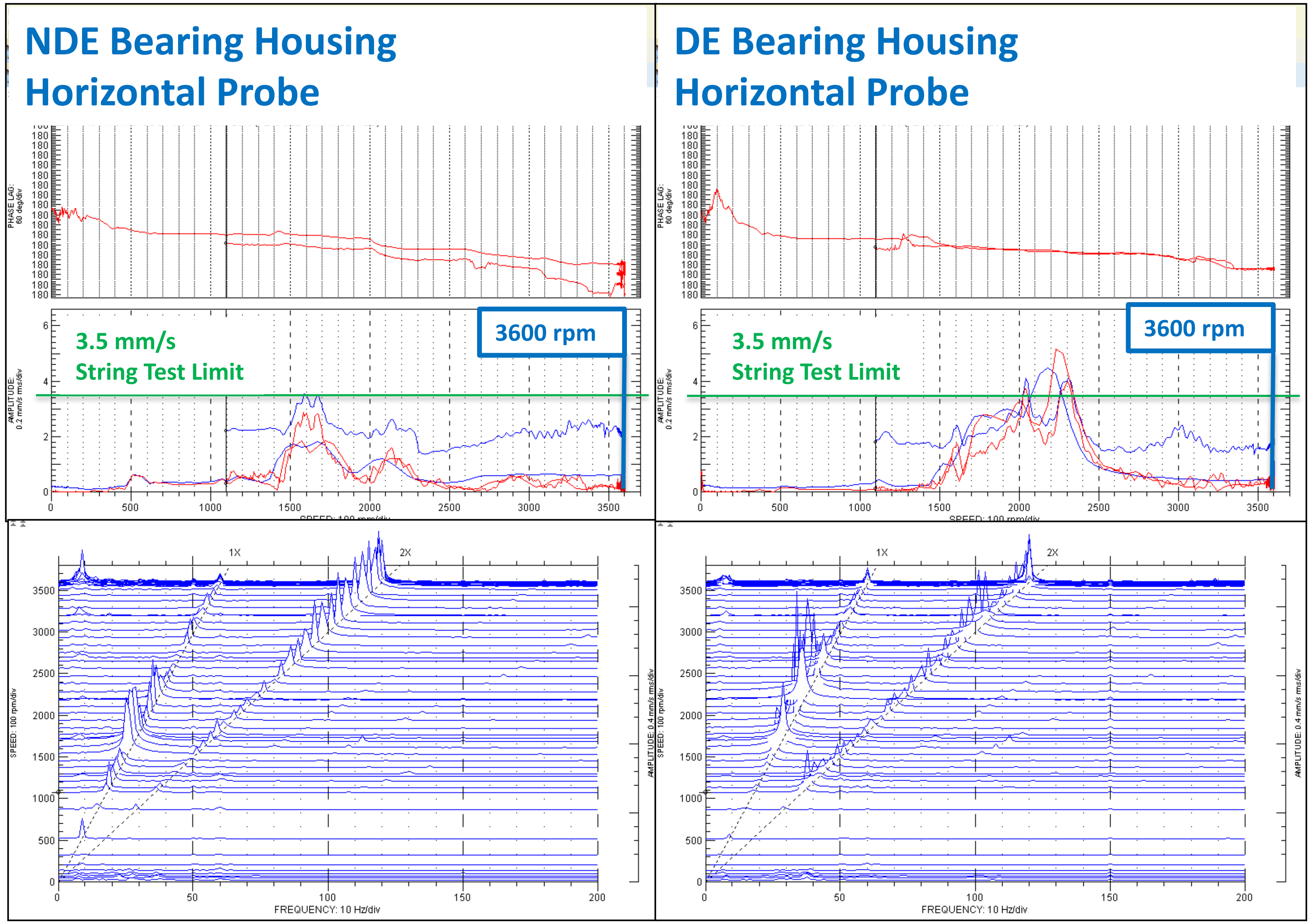


Conclusion and Recommendation

- Assessment of vibration risk in string configuration and at site was addressed by means of dedicated FEA.

- A "qualitative" method of modes classification allowed to demonstrate the low risk even if it was not possible to execute a dangerous mode selection with an harmonic analysis.

- The classification method is applicable only when experimental measurements (ODS and EMA) are available.



Readings during the String Test show total vibrations at operating speed below limits, in compliance with ISO standards, applied for this string test.

Conclusion and Recommendation

2-poles Asynchronous motor dynamic response was found to be strongly dependent from its supporting structure configuration and features. Great expectation on installation size and weight is nowadays increasing and supporting structure shall become much lighter and consequently more flexible. In this scenario, the importance of a machines and elastic supporting structure combined analysis is pointed out and new design strategy shall be developed to avoid unwanted vibration during machine operation.

Suggested best practice are :

- Motor design shall be fulfilled with dedicated finite element model including not only the motor casing but also the relevant supporting structures.

- Identify a standard rule to model the excitation force to perform a reliable harmonic analysis and allow a robust mode classification

Back Up

Amplitude Fluctuation

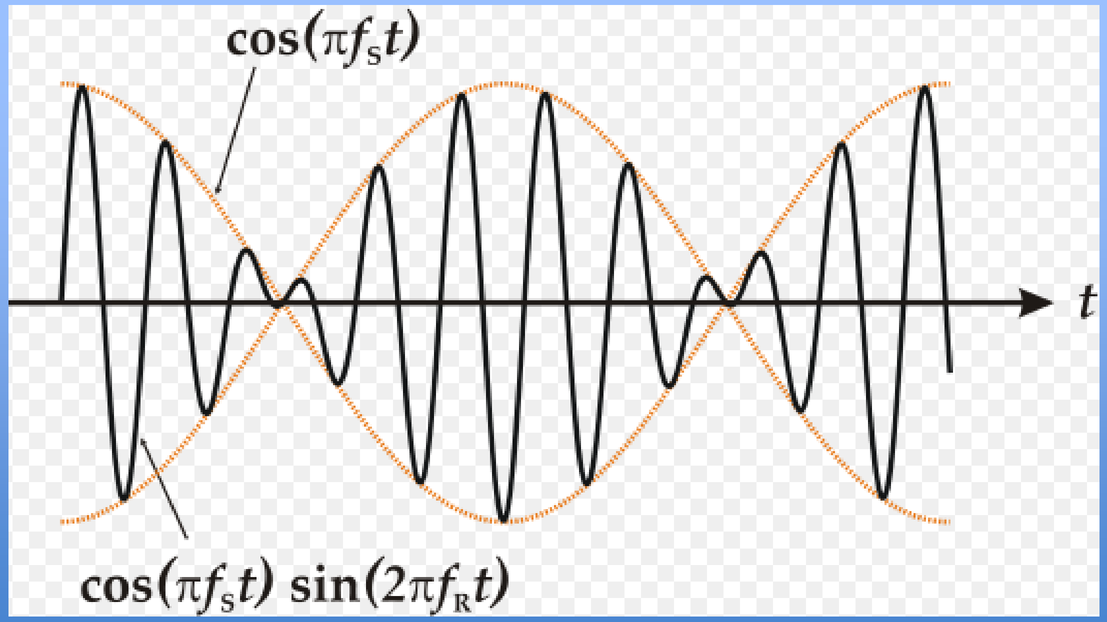
Let's assume to have two waves vibrating close in frequency (2X Mechanical Component and 2X Electrical Component) that acting on the same body:

$$V_m = A \cdot \sin(\omega_m \cdot t)$$
$$V_e = B \cdot \sin(\omega_e \cdot t)$$

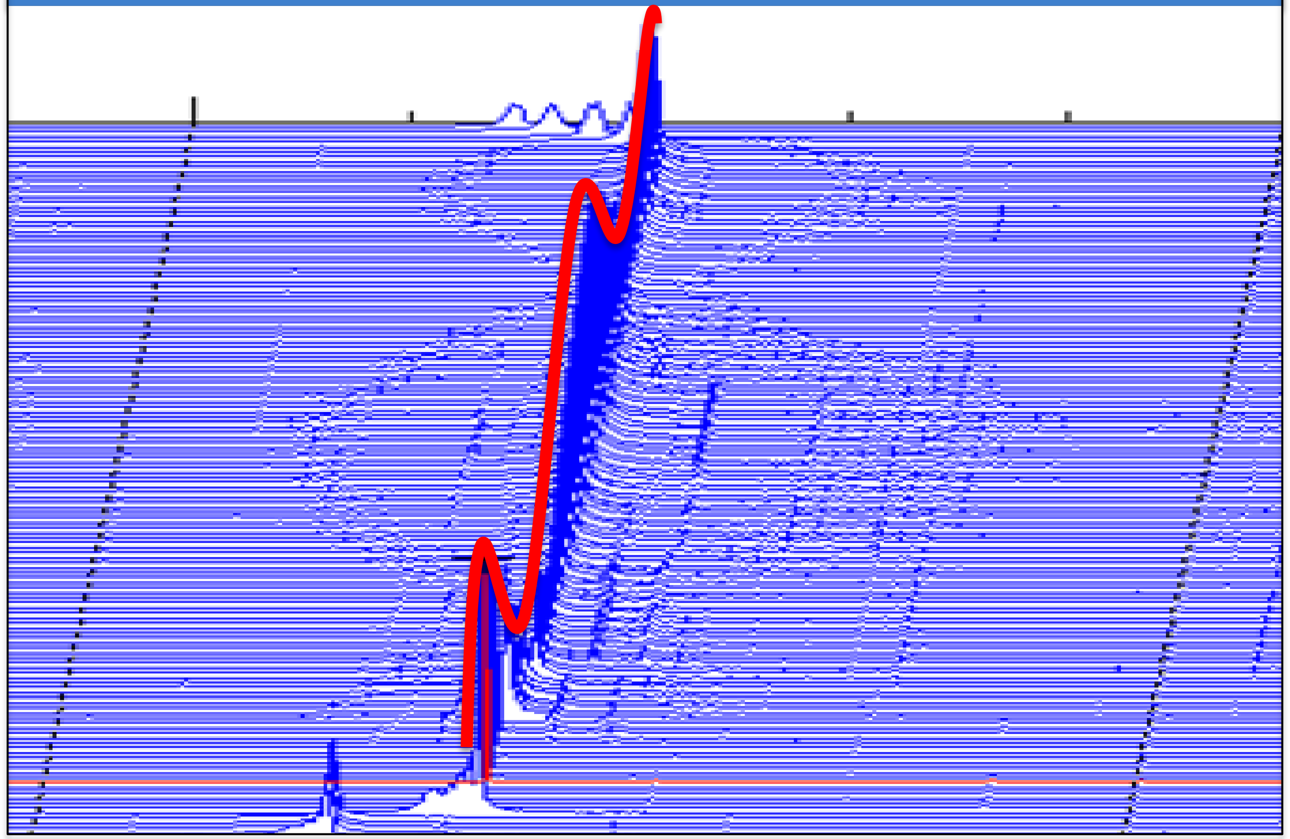
Following Prostaferesi formulation and Beat Theory it is possible to affirm that equivalent vibration is:

$$V_T = C \cdot co\,s\left(\frac{\omega_e - \omega_m}{2} \cdot t\right) \cdot \sin\left(\frac{\omega_e + \omega_m}{2} \cdot t\right)$$

Electrical Component V_e and Mechanical Component V_m are summed when they are in phase and subtracted when out of phase with a period of 2 time the slip defined as $slip = \frac{\omega_e - \omega_m}{\omega_e}$.



From Wikipedia.org



Global Local Indicator

This indicator is calculated for each mode *m,* summing the overall DOF's translation amplitudes and dividing by the number of the DOF's N_j multiplied by the maximum translation amplitude for each mode.

$$Global/Local\ Indicator_{\,m} = \frac{\sum_j |\phi_{jm}|}{N_j \cdot \max_j\{|\phi_{jm}|\}}$$

EMA - OMA - ODS

Experimental Modal Analysis (EMA): allows to identify natural frequencies, mode shapes and damping ratio of the structure by the analysis of vibration data collected exciting the system by an artificial and well known excitation like impact hammer or vibration shakers;

Operational Modal Analysis (OMA): allows to identify natural frequencies, mode shapes and damping ratio by the analysis of all vibration data (removing the 1X component) collected during the normal operation of the machine. It means that the structure is only solicited by machines in operation and not through an artificial and well known excitation like impact hammer or vibration shakers;

Operating Deflection shapes (ODS): allows to reproduce the vibration patterns of the structure observed for some relevant speeds/frequencies by the analysis of vibration data collected during the normal operation of the machine. It allows to identify which are the modes (and relative mode shapes) that are dominating the dynamic response of the structure.