ASIA TURBOMACHINERY & PUMP SYMPOSIUM
SINGAPORE | 22 – 25 FEBRUARY 2016
MARINA BAY SANDS

STRUCTURAL DYNAMIC BEHAVIOR OF FRAME 9E GTG MODULE FOR LNG PLANT

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ABSTRACT

The traditional onshore installation of heavy duty gas turbine generator trains, especially for power >80MW, is stick built on heavy and rigid concrete block foundation.

The challenge of modularization is that vibrations have to be transferred to concrete through structural steel, nevertheless implementing vibrations' acceptance criteria that were developed for direct concrete foundations.

This novel concept of modularization needs a deeper dynamic analysis at system level to ensure that flexible structure modes are not excited at any operating condition; with respect to this subject, the Appendix A to this lecture is a dynamic analysis tutorial for turbomachinery modules, having the aim to describe the process and the tools to be used for purpose, based on author's experiences and lessons learnt also during the experiment described herein.

The results of the dynamic analysis made on a complete system including module, foundation and sub foundation in "Full Speed No Load Test" (FSNL) configuration have been for the 1st time compared with field measurements.

INTRODUCTION

The dynamic behavior of the full GTG module has been analyzed during the design phase taking into consideration the mechanical excitations coming from GT + Generator at running speed, assuming as negligible the contribution given by other harmonics as well as other dynamic excitation forces such as those from rotating auxiliary equipment.

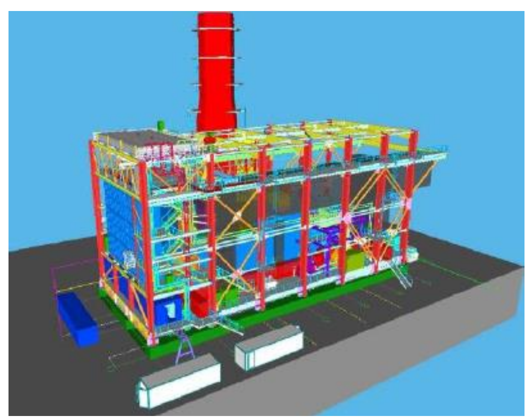

Figure 1: GTG Module 3D Model

The design targets were mainly to verify the acceptance of the vibration amplitude at GTG bearing points versus manufacturer criteria, as well as the accelerations amplitude at Local Control Cab bearing points, which should respect human health criteria as per AS2670.1 / ISO2631.1; further acceptance criteria were those for Air Coolers supporting structures, integrated into the module, as per API661.

Keeping into consideration the above mentioned approximation and those due to the relatively unsophisticated structural model (despite the structural dynamic FEA of the module was carried out building a model inclusive of module structures, concrete foundations and sub-foundation, the structural steel model was just a "beams" one), along the project development it has been decided to set a vibration monitoring campaign during FSNL test, in order to assess the actual behavior of the module and see if it fulfils requirements.

In fact also the structural vibrations' propagation to structural members supporting auxiliary rotating equipment (pumps and fans) was an item of interest.

The calculation model accuracy has been checked essentially by measuring the dynamic response of the system (0-peak amplitude) in 38 strategic locations of structural elements being part of the calculation model under the synchronous excitation harmonic (1xRev, 50Hz); for this purpose the 50Hz response coming from 100+ signals was extracted from the FFT of the overall dynamic response.

The overall system response during the FSNL test has been checked via both evaluating the broadband amplitude of vibrations and visualizing frequency response via FFT.

STRUCTURAL VIBRATION DATA ACQUISITION SYSTEM DESCRIPTION

A total number of 113 accelerometer channels have been installed on GTG Module 1 structure to monitor vibration levels and to assess its dynamic behavior.

Signals coming from the accelerometers were conditioned by specific hardware then digitalized and processed by Data Acquisition System software, which was instrumental to achieve the analysis goals .

Data Acquisition System (DAS) Setup

Acquisition Bandwidth:	5 ÷ 5000 Hz
Monitoring & Assessment Freq. range:	5 ÷ 200 Hz
Acquisition Sample Rate:	10240 Samples/s
Spectral Resolution:	0.25 Hz (800 sp. lines)
Expected Acquisition Noise Threshold:	0.4 mV (*) (**)

Note (*): measurement chains are intrinsically affected by electromagnetic noise; since the main object of structural vibration monitoring is to check vibration levels, a noise threshold has been defined, based on actual noise levels measured on accelerometer chains during pre-test campaign; values of vibration, transduced from this electrical level into EU level, below this threshold shall be considered as electrical noise and not structural vibration.

Note (**): The analysis of experimental data revealed a significantly lower noise level than expected. Since actual noise was concentrated at low frequencies, below 15 Hz, when a digital integration algorithm was applied to evaluate velocity and displacement amplitudes signal-to-noise ratio became lower in the low-frequency range of the spectrum.

DAS Description

DAS is mainly composed by two SW platforms, running on parallel and in synchronous mode:

- Static Platform, where all slowly variable (static) signals are acquired and processed;
- Dynamic Platform, where all fast variable (dynamic) signals are acquired and processed.

Several types of real-time calculations carried out on Dynamic platform signals (e.g. RMS, 0-peak, overall etc.) can be executed by the DAS calculation engine and relevant results transferred in synchronous mode to the Static Platform and then saved in time-stamped records (duly formatted binary files) together with static signals.

Post-processing in several fashions (e.g. averaging, data integration etc.) and exports in text formats (including csv or MS Excel, starting from the binary archive file) were in the system capabilities and were used after the test for reporting and data matching with theoretical models.

STRUCTURAL VIBRATION DATA MONITORING SYSTEM DESCRIPTION

Real time monitoring during test execution was carried out on DAS by means of several client SW applications, either running on Static or Dynamic Platform; while data monitoring on Static platform was made available to the customer during FSNL test, the Dynamic data were recorded for specific post-processing analysis.

Dynamic Platform monitoring applications

Digital Spectrum Analyzer

A digital spectrum analyzer displays FFT plots (amplitude vs frequency). For the test in subject, a Multi FFT plotting system, displaying up to 8 single-signal FFT plots in the same window, was used. Peak hold function was available.

For GTG Module 1 structural vibration monitoring, each of 113 accelerometer channels has been assigned with a set of "scope limits" values obtained by results coming from the theoretical forced harmonic response analysis.

On each FFT plot up to 3 different limit levels can be assigned, using a traffic-light displaying logic, to easily check whether assigned vibration amplitude limits are exceeded.

The amplitude of the accelerometer signal to be considered for scope limit levels comparison is that corresponding to the GTG speed, which is normally defined as the first engine order (or 1x REV) amplitude; since the order tracking is not executed by the SW digital algorithm on a single spectral line but it is carried out by means of a narrow pass-band filter, whose width is defined by 1xREV Frequency +/- 1%, the amplitude considered is the peak value inside this narrow band frequency range.

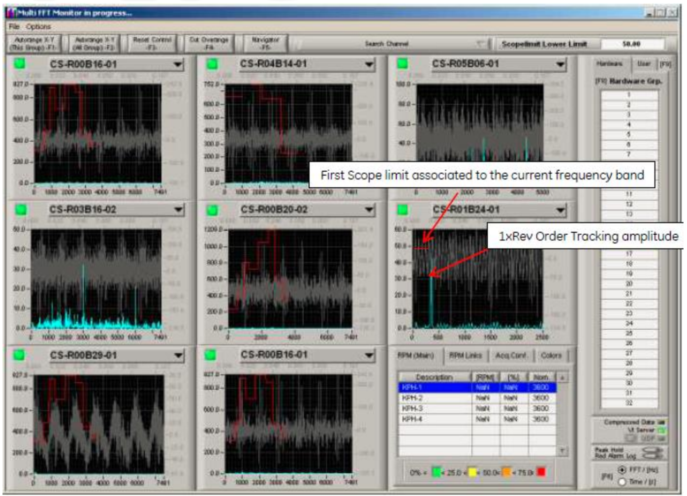

Figure 2: Multi FFT Screenshot

Static Platform monitoring applications

DAS View Plots

DAS View is a trend plot that visualizes trends of up to 10 different signal amplitudes versus time, with a plot refresh time of approx. one time per second. Two Y axes are available for different units of measurement (e.g. RPM and m/s^2).

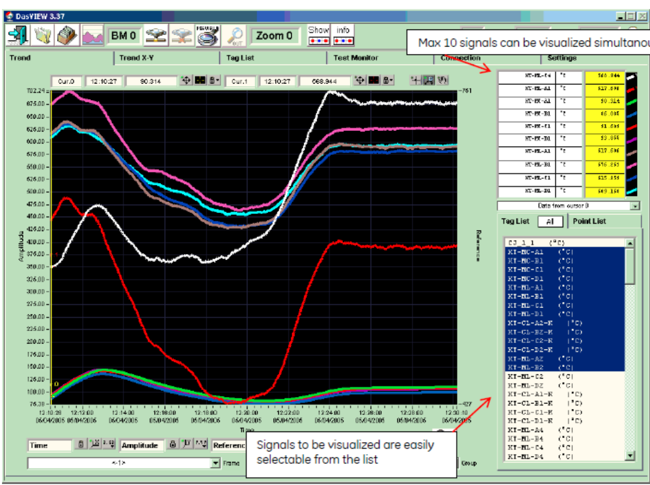

Figure 3: DAS View screenshot

Scorecards

Scorecards are pictorial visualization of real-time signals values on static platform; signals values refresh time is approximately 1 time per second; several background pictures are made available, representing the apparatus under test.

Both signals and expected values are reported as tables located on specific point of the picture to help understanding where the sensor associated to the signal is positioned on the apparatus and if the apparatus is performing in accordance with design values.

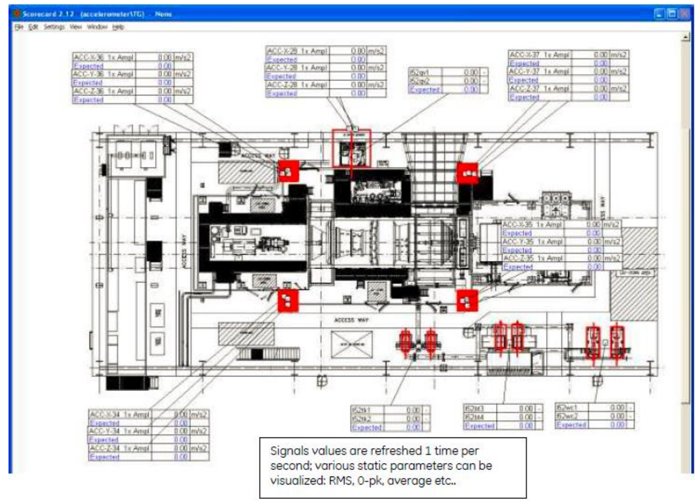

Figure 4: Scorecard screenshot

LOCATION OF PERMANENT ACCELEROMETERS

As said, the calculation model accuracy has been checked essentially by measuring the dynamic response of the system (0-peak amplitude) in 38 locations of the module's structure.

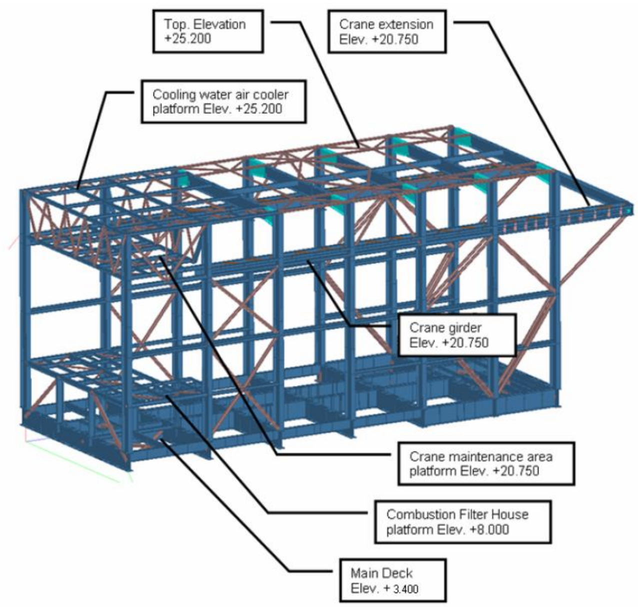

Figure 5: Primary Structures 3D General Arrangement

All the (10) GTG supports locations on the main deck were obviously included, with the purpose of direct monitoring the machinery-structure interaction.

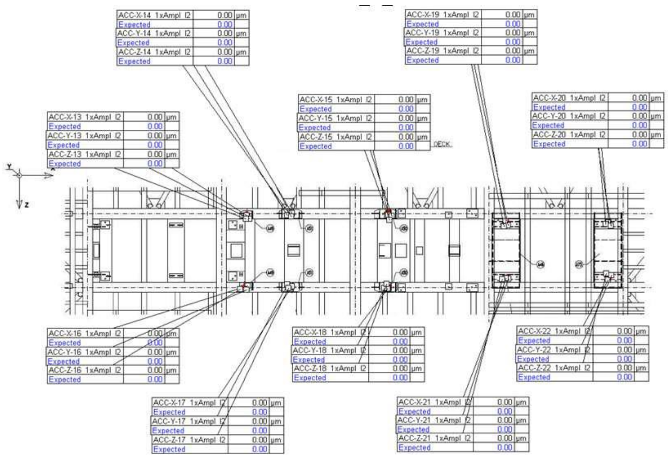

Figure 6: Accelerometer positioning at GTG bearing points

Figure 7: Accelerometers installation on GT supporting girder

Other locations (8) were on the structural steel elements supporting the Local Control Cab installed underneath the GT filter house on the same main deck of the GTG, so to measure the vibrations propagating along the structures and potentially impacting on the health of the personnel which could temporarily stand within the cabinet.

Figure 8: Accelerometers installation on LCC foot

Lastly, 10 additional locations have been selected within those steel members supporting auxiliary equipment such as pumps, fans and air coolers.

As part of the start-up activities for the FSNL test, these auxiliary equipment have been brought online sequentially prior to power turbine cranking and the response of the accelerometers mounted on the structure captured by the data acquisition system in the ways and with the tools described before.

STRUCTURAL VIBRATION AT STEADY STATE - DETAILED FFT GRAPHICS

The following diagrams show frequency domain elaboration (FFT) examples of the recorded dynamic response of the structure for each group of permanent accelerometer installed on the module.

Vibrations ‘amplitudes are shown on the ordinates in the same units of the acceptance criteria; the main sources of excitation are well visible, as peaks, for all the accelerometers.

For a right understanding of the graphs, also the following has to be remembered:

- X=GTG Longitudinal (shaft) direction
- Y=GTG Vertical direction
- Z=GTG Transversal direction
- Readings relevant to digitally integrated Engineering Units, such as μm or mm/s are more impacted by low frequency noise (up to 15Hz) than those relevant to the primary signals Engineering Unit [m/s^2]
- Digital integration is performed starting from the minimum acquisition bandwidth frequency, 5 Hz.

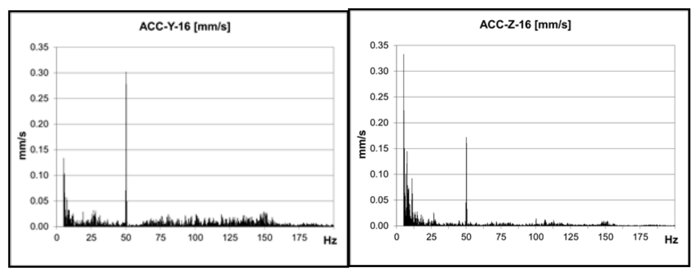

Figure 9: Typical GT Response Spectrum at baseplate bearing points

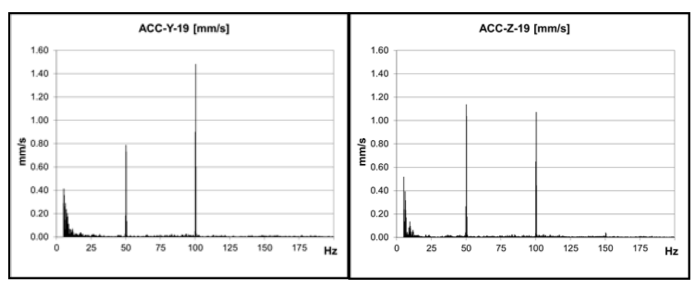

Figure 10: Typical Generator Response Spectrum at baseplate bearing points

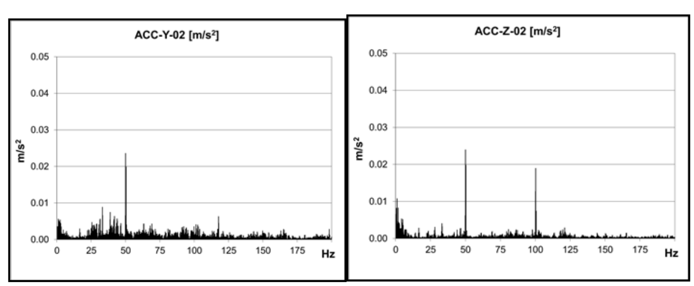

Figure 11: Typical LCC Supporting Structure Response Spectrum

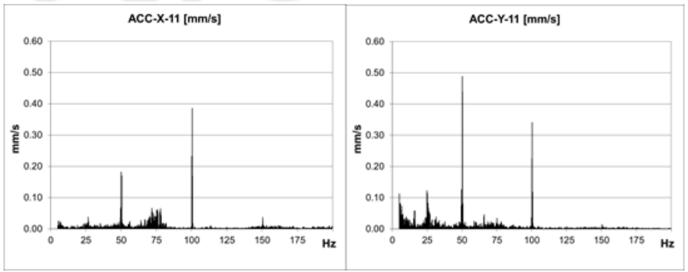

Figure 12: Typical Water Pumps Supporting Structure Response Spectrum

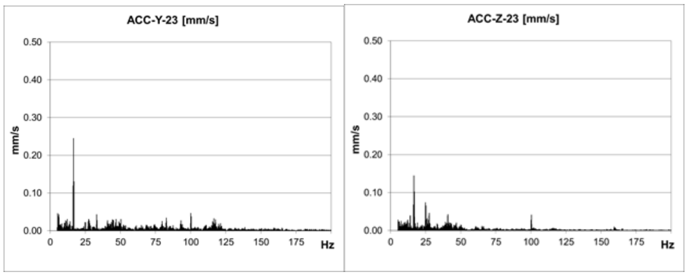

Figure 13: Typical (AVM) Fans Supporting Structure Response Spectrum

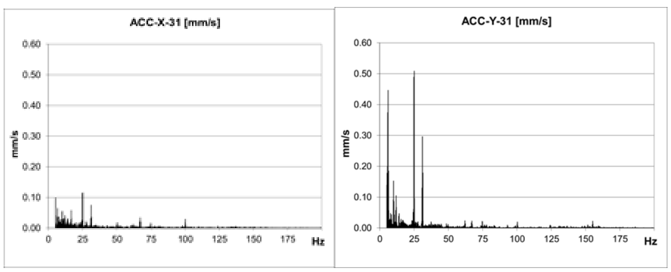

Figure 14: Typical Air Cooler Supporting Structure Response Spectrum

HIGH LEVEL CONSIDERATIONS ABOUT RESULTS

GT Response

The 1xREV contribution at 50Hz is clearly the main one; all other potential sources of vibrations are negligible.

Generator (No Load) Response

The 1xREV contribution at 50Hz is almost equivalent to the 2xREV (100Hz) one.

LCC Supporting Structure Response

The LCC was installed on the main deck, the same of the GTG and the 1xREV contribution (50Hz) is an important one, but also the 2xREV contribution (100Hz) is noticeable in the horizontal direction.

Water Pumps Supporting Structure Response

Despite the pumps were installed on the main deck, the same of the GTG (with the pumps shaft perpendicular to the GTG one) somewhere the 50Hz is not the main contributor of the supporting structures' vibrations.

Also the response at the electric motor speed (25Hz) is noted, but is not the main contributor

On the contrary, the response at the 2xΩ (100Hz), double of electric net frequency, was unexpectedly high.

AVM Fans Supporting Structure Response

The response at the 1xREV excitation 980 rpm (16.33Hz) is well visible as main source of vibrations, despite the background noise; the response at the electric motor speed (25Hz) is also visible.

The response at the 2xREV harmonic (32.66 Hz) of the mechanical excitation is lower than the response at the 2xΩ (100Hz), double of electric net frequency.

The 50Hz main contribution from the GTG didn't propagate up to the mezzanine floor at el.+8000 where the Main Fans were installed.

Air Coolers Supporting Structure Response

The showed graph is relevant to the CW cooler, running at 307rpm (5.12Hz) driven by 4 poles electric motor (25Hz); the fan had 6 blades (blade pass frequency 30.72 Hz).

The API 661 acceptance criterion is given in vibration amplitude (μm) so the response has been integrated twice.

Unfortunately the 1xREV excitation was not visible due to the background noise, however the response at the electric motor speed (25Hz) is visible, as well as the response at the blade pass frequency (30.72 Hz).

The 50Hz main contribution from the GTG didn't propagate up to the el. +25000 where the AC was installed.

TEST RESULTS DRILL DOWN

GTG results check

Test versus FEA results' comparison

As said the structural dynamic FEA of the module was carried out building a model inclusive of structural steel, concrete foundations and sub-foundations. The structural steel members including plate girders were modeled as beam elements (plates meshing were later used for generator supports only, for an analysis drill down) while a hinged "dummy structure" represented the main rotating equipment (GTG).

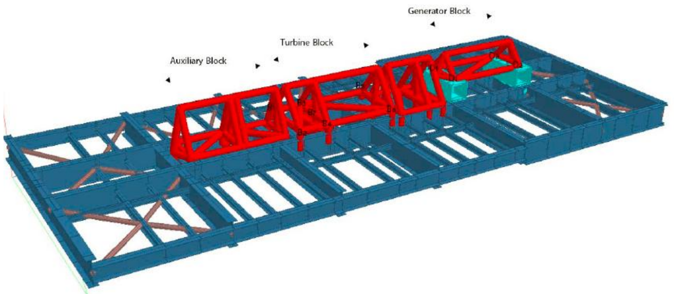

Figure 15: Finite Element Analysis Model Detail

Foundation at the testing yard was a slab (modeled with plate elements) based on a certain number pf piles; the plate elements were so supported by the same number of spring elements.

The dynamic loads used as an input for the analysis in the FSNL test conditions were (only) those reported on the GTG foundation loads drawing (mechanical unbalance loads given at operating speed), duly scaled to represent the newly commissioned machine conditions (see ref.[1], [2]).

The following table shows a comparison between the actual synchronous harmonic responses during FSNL Test (1xRev, 50Hz, as extracted from the FFT) and FEA results for the GTG bearing points.

Overall Amplitudes 0-p			Experimental Results during FSNL test [Peak-Hold, mm]	Contributors (1x ==> 50Hz; 2x ==> 100Hz)	FEA Results in Test cond. [mm]	Exp/FEA Δ (%)
Generator	19	**X**	0,006	1x 50% + 2x 50%	0,012	50%
node 1476 (D1)		**Y**	0,005	1x 50% + 2x 50%	0,018	**28%**
		Z	0,006	1x 66% + 2x 33%	0,012	50%
Generator	20	**X**	0,007	1x 33% + 2x 66%	0,010	70%
node 1478 (D2)		**Y**	0,006	1x 33% + 2x 66%	0,015	**40%**
		Z	0,004	1x 100%	0,010	40%
Generator	21	**X**	0,004	1x 50% + 2x 50%	0,012	33%
node 1474 (D1)		**Y**	0,004	1x 50% + 2x 50%	0,018	**22%**
		Z	0,004	1x 50% + 2x 50%	0,012	33%
Generator	22	**X**	0,003	1x 33% + 2x 66%	0,010	30%
node 1477 (D2)		**Y**	0,003	1x 33% + 2x 66%	0,006	**50%**
		Z	0,003	1x 100%	0,010	30%
Turbine	13	**X**	<0,001	1x 100%	0,008	<10%
node 1445 (B1)		**Y**	0,001	1x 100%	0,015	**7%**
		Z	<0,001	1x 100%	0,008	<10%
Turbine	14	**X**	<0,001	1x 100%	0,007	<10%
node 1447 (B3)		**Y**	0,002	1x 100%	0,032	**6%**
		Z	<0,001	1x 100%	0,007	<10%
Turbine	15	**X**	<0,001	1x 100%	0,007	<10%
node 1451 (B5)		**Y**	0,001	1x 100%	0,025	**4%**
		Z	<0,001	1x 100%	0,007	<10%
Turbine	16	**X**	<0,001	1x 100%	0,008	<10%
node 1444 (B2)		**Y**	0,001	1x 100%	0,012	**8%**
		Z	<0,001	1x 100%	0,008	<10%
Turbine	17	**X**	<0,001	1x 100%	0,007	<10%
node 1446 (B4)		**Y**	0,002	1x 100%	0,027	**7%**
		Z	<0,001	1x 100%	0,007	<10%
Turbine	18	**X**	<0,001	1x 100%	0,007	<10%
node 1450 (B6)		**Y**	0,001	1x 100%	0,022	**5%**
		Z	<0,001	1x 100%	0,007	<10%

Table 1: GTG Synchronous harmonic responses comparison

Reference acceptance criteria

The acceptance criteria for newly commissioned machines (see ref. [2]) are the following:

For the Industrial GT (API617/ISO10816-4):

Zone boundary	**r.m.s. vibration velocity at zone boundaries** mm/s
A/B	4,5

Corresponding to 20µm (0-p) amplitude @ 50Hz.

For the Generator (ISO10816-2):

Zone boundary	**Shaft rotational speed** r/min	
	1 500 or 1 800	3 000 or 3 600
	r.m.s. vibration velocity at zone boundaries mm/s	
A/B	2,8	3,8

Corresponding to 17µm (0-p) amplitude @ 50Hz.

Table 2: GTG acceptance criteria as per ISO 10816

To be noted that only the above ground part of the foundation used for the testing of the modules was designed for purpose during the revamping of the yard, so FEA results were somewhere borderline or slightly exceeding the reference acceptance criteria; since (one of) the purpose of the test is to validate the accuracy of the calculation model, this hasn't to be considered as an issue.

Human Exposure Check

Test versus FEA results' comparison

The following table shows the actual synchronous harmonic responses during FSNL Test (1xRev, 50Hz, as extracted from the FFT) and the acceptance criteria used at the LCC resting/anchoring points (see ref. [3]).

Overall Amplitudes 0-p			Experimental Results in FSNL test / Peak-Hold (	Contributors (1x ==> 50Hz; 2x ==> 100Hz)	Ref. Acceptance Criteria (Design)(*)
Underside Control Room	02	**X**	0.07m/sq.s	1x 33% + 2x 66%	1m/sq.s (RMS_W)
		Y	0.03m/sq.s	1x 100%	1m/sq.s (RMS_W)
		Z	0.05m/sq.s	1x 50% + 2x 50%	1m/sq.s (RMS_W)
Underside Control Room	04	**X**	0.05m/sq.s	1x 100%	1m/sq.s (RMS_W)
		Y	0.05m/sq.s	1x 33% + 2x 33% + others	1m/sq.s (RMS_W)
		Z	0.02m/sq.s	1x100%	1m/sq.s (RMS_W)
Underside Control Room	06	**X**	0.06m/sq.s	1x 50% + 2x 25% + others	1m/sq.s (RMS_W)
		Y	0.08m/sq.s	1x 25% + 2x 25% + others	1m/sq.s (RMS_W)
		Z	0.02m/sq.s	1x 66% + 2x 33%	1m/sq.s (RMS_W)
Underside Control Room	08	**X**	0.04m/sq.s	1x 66% + 2x 33%	1m/sq.s (RMS_W)
		Y	0.03m/sq.s	1x 50% + 2x 50%	1m/sq.s (RMS_W)
		Z	0.02m/sq.s	1x 50% + 2x 50%	1m/sq.s (RMS_W)
Underside Control Room	03	**X**	0.06m/sq.s	1x 100%	1m/sq.s (RMS_W)
		Y	0.03m/sq.s	1x 33% + 2x 66%	1m/sq.s (RMS_W)
		Z	0.03m/sq.s	1x 66% + 2x 33%	1m/sq.s (RMS_W)
Underside Control Room	05	**X**	0.04m/sq.s	1x 66% + 2x 33%	1m/sq.s (RMS_W)
		Y	0.06m/sq.s	1x 66% + 2x 33%	1m/sq.s (RMS_W)
		Z	0.06m/sq.s	1x 66% + 2x 33%	1m/sq.s (RMS_W)
Underside Control Room	07	**X**	0.05m/sq.s	1x 66% + 2x 33%	1m/sq.s (RMS_W)
		Y	0.01m/sq.s	1x 100%	1m/sq.s (RMS_W)
		Z	0.02m/sq.s	1x 50% + 2x 50%	1m/sq.s (RMS_W)

Table 3: LCC Feet peak-hold responses

Reference acceptance criteria

The AS 2670-1 (BS ISO 2631-1, see ref. [3]) provides health effect evaluation criteria according two different equations, which converge when the occupational duration of the area subject to vibrations is in the range from 4 to 8 hours (normal labor shift duration).

The accelerations have to be weighed both on frequency base and depending on human position and vibrations direction.

The limit for the RMS weighed acceleration was compared with the graph below, where the upper limit of the dashed area ($1m/s^2$) was considered not acceptable.

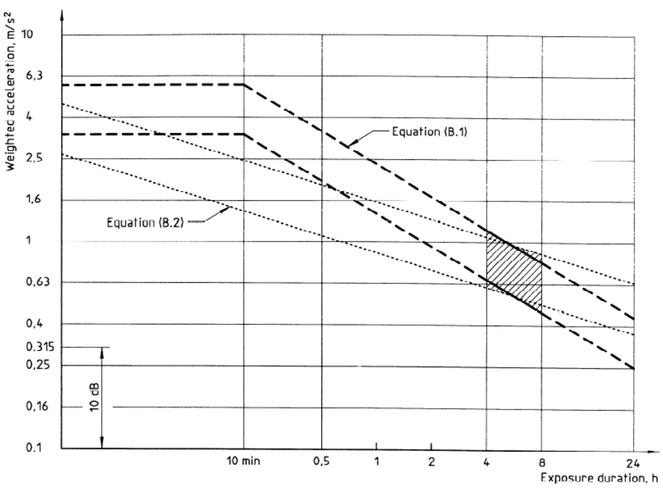

Figure 16: Health caution zone as per AS 2670-1(ISO 2631-1)

The following table is relevant to the elaboration of the measured acceleration, in one location and directions, required by the applicable code: the frequency domain response extracted as per 1/3 octave band, in the range 10÷200Hz, have been weighed according to AS 2670-1 (BS ISO 2631-1) table 3.

Wk - Y direction					
1/3 octave [Hz]	sum [m/s2] Y		sum rms [m/s2]	factor x1000	weighted (m/s2)
10	0,09		0,06	988,0	0,06
12,5	0,01		0,00	902,0	0,00
16	0,01		0,01	768,0	0,01
20	0,01		0,01	636,0	0,01
25	0,06		0,04	513,0	0,02
31,5	0,07		0,05	405,0	0,02
40	0,11		0,08	314,0	0,02
50	0,11		0,08	246,0	0,02
63	0,11		0,08	186,0	0,01
80	0,09		0,06	132,0	0,01
100	0,14		0,10	88,7	0,01
125	0,12		0,08	54,0	0,00
160	0,12		0,09	28,5	0,00
200	0,05		0,04	15,2	0,00
					0,20
					total m/s2 (green if <1)

Table 4: LCC feet typical weighed RMS responses

Auxiliary Equipment Check

The table below shows the results' summary of the measurement that had been taken on the auxiliary equipment both during commissioning and during the full unit's testing, with the purpose to get through the differences in readings, the influence of the vibrations coming from the GTG on the overall performance.

While a set of permanent accelerometers were positioned on an agreed sample of supporting structures, vibrations of the equipment casing had been taken with a portable device.

Overall Amplitudes 0-p			Experimental Results in FSNL test / Peak-Hold (**)	Contributors (1x ==> 50Hz; 2x ==> 100Hz)	Staad Results in test cond.	Ref. Acceptance Criteria (Design)(*
Water Pump	11	X	0.6mm/s	1x 50% + 2x 50%	2mm/s	
node 1233 (closer to line F)		Y	0.8mm/s	1x 50% + 2x 50%	1mm/s	3mm/s RMS (***)
(GTG at 3000rpm)		Z	<0.5mm/s	1x 50% + 2x 50%	5mm/s	
Water Pump	11	X	below of "GTG at 3000rpm"		2mm/s	
node 1233 (closer to line F)		Y	below of "GTG at 3000rpm"		1mm/s	3mm/s RMS (***)
(Auxiliaries pre-test)		Z	below of "GTG at 3000rpm"		5mm/s	
Water Pump	12	X	0.8mm/s	1x 50% + 2x 50%	1mm/s	
node 1258 (closer to line G)		Y	1.5mm/s	1x 50% + 2x 50%	0.1mm/s	3mm/s RMS (***)
(GTG at 3000rpm)		Z	<0.5mm/s	1x 50% + 2x 50%	2mm/s	
Water Pump	12	X	below of "GTG at 3000rpm"		1mm/s	
node 1258 (closer to line G)		Y	below of "GTG at 3000rpm"		0.1mm/s	3mm/s RMS (***)
(Auxiliaries pre-test)		Z	below of "GTG at 3000rpm"		2mm/s	
Air Coolers (°) (°°)	30	X	0.001mm	25Hz 66% + 31Hz 33%	<0.001mm (°°)	
Lube oil & CW		Y	0.007mm	25Hz 66% + 31Hz 33%	<0.001mm (°°)	0.150mm (p-p)
(GTG at 3000rpm)		Z	0.002mm	25Hz 100%	0.002mm (°°)	
Air Coolers (°) (°°)	30	X	0.0013mm	25Hz 80% + 31Hz 10% + 62Hz 10%	<0.001mm (°°)	
Lube oil & CW		Y	below of "GTG at 3000rpm"	25Hz 50% + 31Hz 50%	<0.001mm (°°)	0.150mm (p-p)
(Auxiliaries pre-test)		Z	0.0022mm	25Hz 70% + 31Hz 30%	0.002mm (°°)	
Air Coolers (°) (°°)	31	X	0.001mm	25Hz 66% + 31Hz 33%	<0.001mm (°°)	
Lube oil & CW		Y	0.005mm	25Hz 66% + 31Hz 33%	<0.001mm (°°)	0.150mm (p-p)
(GTG at 3000rpm)		Z	0.003mm	25Hz 100%	0.002mm (°°)	
Air Coolers (°) (°°)	31	X	0.0025mm	25Hz 33% + 31Hz 66%	<0.001mm (°°)	
Lube oil & CW		Y	0.0055mm	25Hz 66% + 31Hz 33%	<0.001mm (°°)	0.150mm (p-p)
(Auxiliaries pre-test)		Z	below of "GTG at 3000rpm"	25Hz 50% + 31Hz 40% + 62Hz 10%	0.002mm (°°)	
Air Coolers (°°)	32	X	<0.001mm	25Hz 66% + 31Hz 33%	<0.001mm (°°)	
CW side		Y	<0.001mm	25Hz 66% + 31Hz 33%	<0.001mm (°°)	0.150mm (p-p)
(GTG at 3000rpm)		Z	<0.001mm	25Hz 100%	0.002mm (°°)	
Air Coolers (°°)	32	X	below of "GTG at 3000rpm"	25Hz 50% + 31Hz 50%	<0.001mm (°°)	
CW side		Y	0.0015mm	31Hz 80% + 62Hz 20%	<0.001mm (°°)	0.150mm (p-p)
(Auxiliaries pre-test)		Z	below of "GTG at 3000rpm"	25Hz 100%	0.002mm (°°)	
Air Coolers (°°)	33	X	<0.001mm	25Hz 80% + 62Hz 20%	<0.001mm (°°)	
CW side		Y	<0.001mm	25Hz 66% + 31Hz 33%	<0.002mm (°°)	0.150mm (p-p)
(GTG at 3000rpm)		Z	<0.002mm	25Hz 100%	<0.002mm (°°)	
Air Coolers (°°)	33	X	0.0018mm	25Hz 60% + 18Hz 40%	<0.001mm (°°)	
CW side		Y	0.0035mm	25Hz 45% + 31Hz 45% + 62Hz 10%	<0.002mm (°°)	0.150mm (p-p)
(Auxiliaries pre-test)		Z	below of "GTG at 3000rpm"	25Hz 100%	<0.002mm (°°)	
Main Fan (°°°)	23	X	0.5mm/s	16.5Hz (30% mass mode X-18Hz)	1mm/s	
(GTG at 3000rpm)		Y	0.4mm/s	16.5Hz mainly	0.5mm/s	5mm/s RMS (***)
		Z	0.3mm/s	16.5Hz mainly	1mm/s	
Main Fan (°°°)	23	X	below of "GTG at 3000rpm"		1mm/s	
(Auxiliaries pre-test)		Y	below of "GTG at 3000rpm"		0.5mm/s	5mm/s RMS (***)
		Z	below of "GTG at 3000rpm"		1mm/s	
Main Fan (°°°)	24	X	0.4mm/s	16.5Hz (30% mass mode X-18Hz)	1mm/s	
(GTG at 3000rpm)		Y	1.2mm/s	16.5Hz mainly	0.5mm/s	5mm/s RMS (***)
		Z	0.3mm/s	16.5Hz mainly	0.2mm/s	
Main Fan (°°°)	24	X	below of "GTG at 3000rpm"		1mm/s	
(Auxiliaries pre-test)		Y	below of "GTG at 3000rpm"		0.5mm/s	5mm/s RMS (***)
		Z	below of "GTG at 3000rpm"		0.2mm/s	
Main Fan (°°°)	25	X	Malfunctioning	Distributed in range 14-42Hz	0.5mm/s	
(GTG at 3000rpm)		Y	0.8mm/s	Distributed in range 14-42Hz	0.5mm/s	5mm/s RMS (***)
		Z	0.3mm/s	16.5Hz mainly	0.5mm/s	
Main Fan (°°°)	25	X	Malfunctioning	Distributed in range 14-42Hz	0.5mm/s	
(Auxiliaries pre-test)		Y	below of "GTG at 3000rpm"		0.5mm/s	5mm/s RMS (***)
		Z	below of "GTG at 3000rpm"		0.5mm/s	
Main Fan (°°°)	26	X	0.4mm/s	16.5Hz (30% mass mode X-18Hz)	0.5mm/s	
(GTG at 3000rpm)		Y	1.0mm/s	Distributed in range 14-42Hz	0.5mm/s	5mm/s RMS (***)
		Z	0.4mm/s	16.5Hz (30% mass mode X-18Hz)	0.5mm/s	
Main Fan (°°°)	26	X	below of "GTG at 3000rpm"		0.5mm/s	
(Auxiliaries pre-test)		Y	below of "GTG at 3000rpm"		0.5mm/s	5mm/s RMS (***)
		Z	below of "GTG at 3000rpm"		0.5mm/s	

Table 5: Aux. equipment responses (GTG unit FSNL test vs Stand Alone test)

The only unexpected observed behavior was a 100Hz vibration on the electric motor pump casing that the supporting structure wasn't able to damp. The source of this kind of vibration was likely the UMP, which generates a quite important excitation source at 2xΩ.

Since the phenomena was not associated to a specific, skid based, electric motor pump but to a certain installation position, it means that there was a local modal shape on the structure close to 100Hz leading a dynamic stiffness reduction at the same frequency.

After some attempt to fix the issue by stiffening the motor support, it has been found much more effective to install the electric motor pump skid on duly selected, basic type, AVM.

STRUCTURAL VIBRATIONS DURING TRANSIENT

All the previous results were relevant to steady state operations at the nominal speed but also the transient conditions may be an area of concern, due to the several natural modes of the structure encountered during unit's startup and shutdown.

It is paramount ensuring that during the transient nothing having the power to trip the units may happen.

The startup sequence of the GTG foresees that as 1st step the unit reaches the CRANK speed (few Hz) where it will remain for several minutes; as 2nd step there is the ramp-up from CRANK speed to FULL speed (50Hz), that happens quite quickly (about 10'time for this unit); once reached the full speed it practically starts a warm-up period that (at No Load), as per measurement, took about 60'time for this unit.

The expectations in terms of structural vibrations were that the "steady state" conditions at crank wouldn't create any issue, since the natural modes of vibrations of the structure were properly segregated; on top of this also the excitation from the rotating mass, depending on the square power of the rotating speed is drastically reduced.

For different reasons, basically for the velocity of the process, also the ramp-up was not expected to create issues.

In theory the warm-up period could be the most troublesome because the machines connected along the shaft line are progressively reaching the optimal alignment

The following trends show typical vibrations' amplitudes (µm) and gas turbine speed (RPM) versus time that have been monitored respectively during start-up and warm-up.

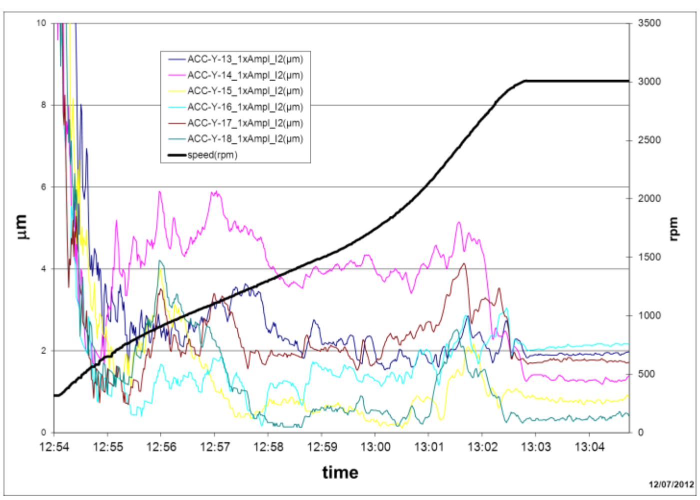

Figure 17: GT Vertical Vibrations Trends during Startup (5')

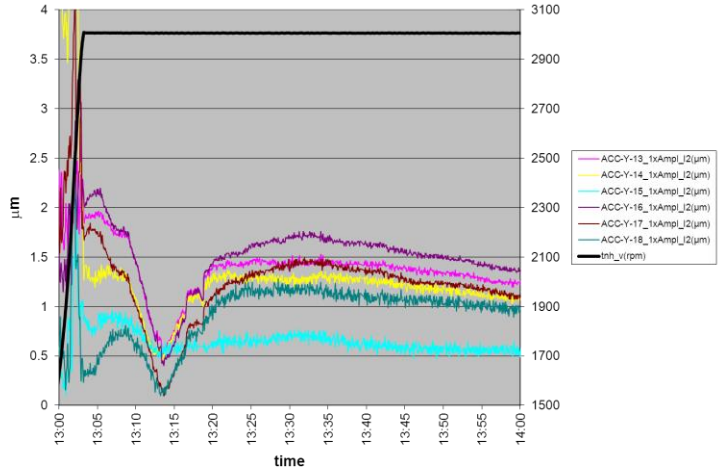

Figure 18: GT Vertical Vibrations Trends during Warmup (1h)

CONCLUSIONS

The structural vibrations' monitoring campaign during the GTG unit FSNL test proved both that the module structure, sitting on a properly designed foundation, was fitting for purpose and that the calculation model was able to predict with enough accuracy the structural behavior.

The steelwork vibrations' levels, recorded after GTG thermal stabilization (fully reached about after about 75') along three directions in the 38 locations agreed with the project team, were better than expectations (especially for the GT) and well below expectations for newly commissioned machines.

Despite not shown within this paper, the vibrations at GT bearing points were fully consistent with measurement recorded from GT seismic probes on the GT casing.

Some lessons have been also learnt and implemented within Design Practices for structural modelling (see also Appendix A); only the fact that the real damping of the system was probably underestimated will not be (conservatively) taken into account in the next future.

NOMENCLATURE

Variables

mm/s	= Vibration's Velocity	(L/T)
μm	= Vibration's Amplitude	(L)
N/m	= Stiffness	(F/L ≡ M/t^2)

Acronyms

AS = Australian Standard
API = American Petroleum Institute
AVM = Anti Vibration Mount
BS = British Standard
DAS = Data Acquisition System
EU = Engineering Unit
FEA = Finite Element Analysis
FFT = Fast Fourier Transformer
FMEA = Failure Mode Effect Analysis
FSFL = Full Speed Full Load
FSNL = Full Speed No Load
GT = Gas Turbine
GTG = Gas Turbine Generator
ISO = International Standard Organization
OEM = Original Equipment Manufacturer
LCC = Local Control Cab
RMS = Root Mean Square
RPM = Round per Minute
TM = Turbo Machinery
UMP = Unbalanced Magnetic Pull
USL = Ultimate Limit State
Ω = Electric Net Frequency
0-p = Zero to Peak
p-p = Peak to Peak
1x REV = One per Revolution, machinery rotating speed

APPENDIX A - TURBOMACHINERY MODULES' DYNAMIC ANALYSIS

When modularization involves rotating equipment, it is necessary to ensure a favorable dynamic behavior of the whole system to achieve long and successful machinery operation.

This tutorial has the purpose to describe the analysis and design process, clarifying which kind of checks have to be done on the industrial module's structure and where, in order to ensure consistent performances.

While defining all the tasks in charge of the structural engineers responsible for the modularization, also roles & responsibilities of all other parties involved will be clearly addressed.

The tutorial is formulated thinking to a modularized onshore installation (that's the most complicated) but the concepts are easily transferable to offshore installations.

Introduction

The pillars for every successful analysis are almost always the same: solid design inputs/data, consistent design acceptance criteria and a robust process in place to obtain reliable results.

At a deeper level, also building an appropriate analysis model, having the availability of performant tools and adequate technicalities to properly use them are paramount for the specific purpose, so they will be covered too by this tutorial.

Design Inputs/Data

The design inputs/data come from two main sources: the Original Equipment Manufacturer (OEM) and the responsible party for the civil works execution (in general the customer).

The input data from the TM OEM have to be properly interpreted and used, starting from dynamic excitation loads usually reported on the machinery foundations' drawings.

Such loads should be clearly divided in those referring to abnormal (short-term) conditions of the machines, relevant to emergency / catastrophic events (i.e. the loss of a blade for a GT or the short circuit of an electric machine) and those referring to normal (long-term) operating conditions.

Since the emergency dynamic loads are due to events that are such to cause an immediate trip of the machine, they substantially act as an impulse and are commonly treated as static loads, eventually applying a dynamic magnification factor into the load combinations for the strength design.

On the contrary, the dynamic loads in "normal" operating conditions come from a situation in which the machinery can operate for "long term" without getting in trip.

The main source of the operating dynamic loads is a mechanical one, due to the rotors' unbalance. These loads are always present because, although the new turbo-machinery units have to match balance quality requirements for rotors such those defined from standard such as ISO 1940 or API RP 684, the mass centroid never coincide with the center of rotation.

The eccentric rotating masses produce centrifugal forces that are transmitted to the foundation through its bearings. These forces acting to the bearings are function of many factors: level and axial distribution of unbalance, geometry of the rotor, bearings' type and position, rotation's speed and rotor-dynamic; nevertheless the unbalance forces vary also with the time because, starting from the newly commissioned machines conditions, the rotor unbalance is expected to constantly grow within the machinery major maintenance interval, generating vibrations that can be up to the boundary with alarm level.

The input data from customer, assuming that the industrial module's conceptual layout has been frozen, are instead relevant to the foundations and sub-foundations engineering.

In facts, despite the dynamic analysis of a module foresees preparing a comprehensive FEM structural model, including the module, its foundation and the sub-foundation, the party in charge of designing and delivering a TM module is usually responsible for designing only the above ground part of the foundation, while the underground part is in charge of others.

The foundations' design starts with the selection of the foundation type (shallow or deep), which is generally driven by the native soil bearing capacity, followed by the preliminary sizing of the foundation based on the static loads acting on it.

Since the analysis model has to include also the sub-foundations duly characterized in terms of stiffness & damping, the responsible party for the foundations' design should also provide a fully interpreted geotechnical report, giving clear indications about the most appropriate distribution of stiffness & dampers to be used for completing the analysis model.

Uncertainties are surely foreseen, but they have to be managed in a practical way, for example setting Lower Bound & Upper Bound values for the parameters that are directly measured during the geotechnical investigations (i.e. Vs), which are later used to generate workable inputs (i.e. dynamic shear modulus G and, finally, the springs distribution).

Design Process

The high level design process is described here below.

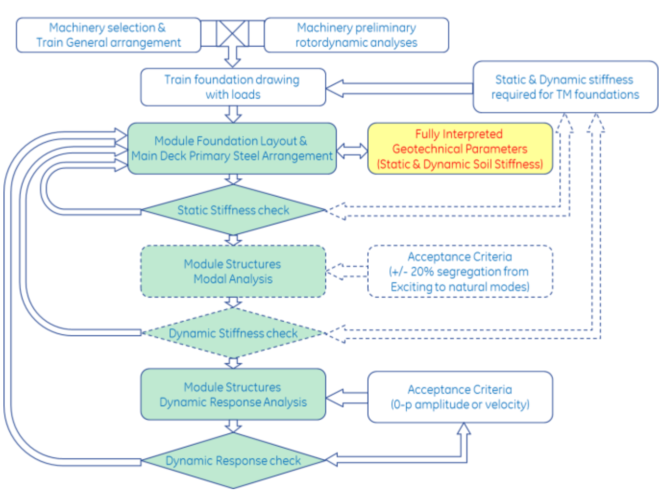

Figure 19: TM Module Dynamic Analysis Process

Dynamic analysis of machine foundation has the purpose to evaluate vibrational behavior of the system under dynamic loading and check if vibratory motions at TM bearing points are within allowable limits.

In details, the assessment is usually done through the following steps:

- Modal Analysis
- Dynamic stiffness analysis
- Dynamic response analysis

The above diagram indicates also the static stiffness analysis as a preliminary check (usually it is checked the maximum differential settlement of the bearing points under dead weight only, so higher is the value and easier is to get the differential settlement requirement) to be executed prior to start the dynamic analysis.

This not only because an adequate static stiffness of the machinery bearing points is needed to be successful in the TM string alignment operations prior of the unit start up but also because the static stiffness value is in some way a "baseline" for the dynamic stiffness (being the dynamic stiffness limit value when "f" goes to zero), so it can be an useful reference to evaluate the dynamic stiffness trend in the operating frequencies range.

Design Acceptance Criteria

The installation of turbomachinery is historically stick-built onshore, on heavy and rigid concrete block foundation.

As a matter of fact several standards still define very stringent acceptance criteria making reference to the technical literature developed for the aforesaid cases some decades ago, when also calculations' tools had strong limitations.

In that scenario, when unwanted performances came out from the analysis, the most common strategy to implement corrective action was increasing block mass & stiffness; in short, a quite cheap solution was available to apply the conservatisms in the design that were needed to match the stringent acceptance criteria, as advisable in the past to mitigate the design approximations.

In modularized solutions the dynamic excitation from machinery is transferred through a "flexible" structural steel layer to another "flexible" foundation (considering the modularized unit's dimensions, even onshore the foundation shouldn't be a block type one), so both the geometry and the expected behaviour of the system are such that it is very hard to apply conservatisms in the design.

At the end the dynamic performance of the system has to satisfy the minimum requirements in terms of acceptance criteria that are set from TM OEM based on the actual needs of the machines, including rotordynamic and depending on the hypothesis under with dynamic loads have been generated.

On top of the above there are other criteria aimed to mitigate the effect of the vibrations that may propagate from the machinery through the steel structure, elsewhere on the module. For example, for manned installations, there are regulations regarding the evaluation of whole-body vibration with respect to human's health that have to be satisfied.

Analysis Model

The peculiarities of the dynamic analysis of a TM module in operating conditions with respect to the more common structural dynamic analysis under earthquake effect are that the excitation source is not external to the system but within the system and that the excitation frequency is much higher.

This means that some of the concepts that usually apply to the structural dynamic analysis for earthquake don't apply to our case. This topic will be further developed later on but it has been introduced now as background for the following considerations.

As said, the analysis model has to include the module steel structures, foundations and sub-foundations duly characterized in terms of stiffness & damping.

The steel structure model has to include all the masses that will be actually present onto the industrial module in operation, in the right position and elevation; since the structural steel self-weight and stiffness play a decisive role in the results it's strongly recommended to perform the final run of the dynamic analysis once the strength design is completed and the primary steel geometry and size is not supposed to change further.

Despite the module's structure is basically a beam's grid, it's anyway advisable to model using shell/plate elements at least the structural members that are supposed to be more sensitive to the effect of the input loads.

For an onshore industrial module, due to the extension of the plot area, the foundation will be usually made by a relatively thin slab supporting an adequate number of plinths; such plinths will be linked to the structural steel starting in way to represent the real restraints, keeping in mind that the amplitude of the vibrations that have to be kept under control should not exceed few tenths of microns.

The sub-foundation will be modelled differently depending on the foundation type, shallow or deep; in both cases a fully interpreted geotechnical report, giving clear indications about the most appropriate distribution of stiffness & dampers to be used for completing the analysis model, is needed.

For shallow foundations it has been found that the well-known Winkler's model is not enough accurate to describe the sub-foundation proprieties and so more advanced models should be used in order to take into account boundary conditions at the edges of the mat.

For deep foundations there are fewer doubts about the springs and damper distribution, but caution has to be given to the mutual influence of piles (in other words, for keeping into account the "piles' group effect", if applicable).

Modularized TM configurations

TM train purpose

At first the engineers has immediately to understand which kind of TM's application has to be analysed between mechanical drive and generator drive.

The main difference between the two is that the generator drive operates at constant speed, while the mechanical drive operates on a continuous basis over a variable speed range.

For both cases the eventual presence of a gear in between the driver and the driven equipment makes wider the range of exciting frequencies to be analysed.

Single Deck Modules or Pre-Assembled Units (PAU)

In these configurations the TM modularization concept as "plug & play" solution applies for its minimum extent; the machinery and their auxiliaries are mounted on a common steel deck together with a local electrical and control room (see fig.20) anyway allowing the whole unit testing prior to shipment.

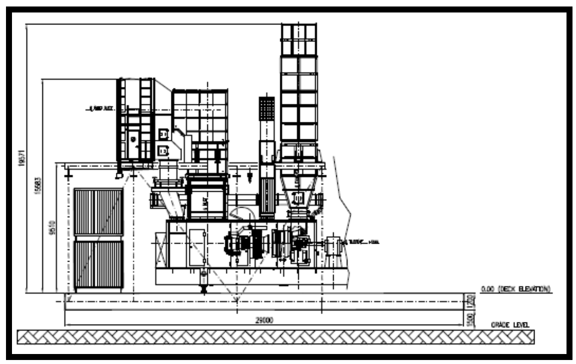

Figure 20: Single Deck Module – Side view

Full Modules

In this configuration the TM modularization concept as "plug & play" solution applies for its maximum extent.

The machinery and relevant auxiliaries are mounted together with a local electrical and control room within a steel grid (see figure below), not only allowing connecting and testing the whole unit prior to shipment, but also providing all the devices, tools and spaces for maintenance operations on site.

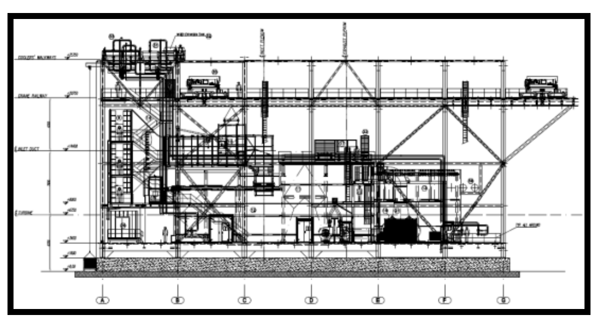

Figure 21: Full GTG Module – Side View

This configuration can give better flexibility and improve the cycle time of the maintenance operations; it is obviously heavier but also because, once decided to move in this direction, the bigger structure offers opportunity to maximize the items that can be installed on board of the module.

For several reasons, including dynamic performances, the typical arrangement foresees the machinery installed on the lower level deck (see fig.22), which is directly sitting on foundations.

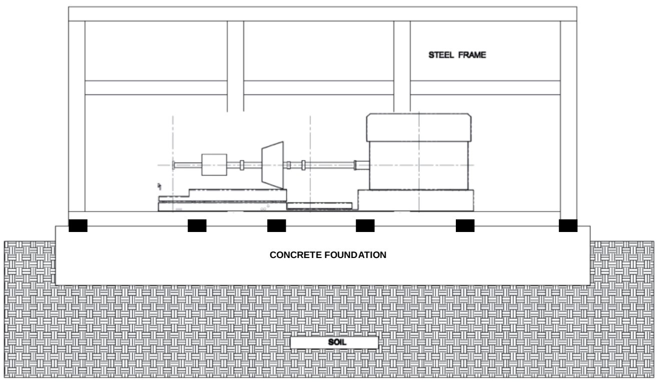

Figure 22: Onshore TM Module on Shallow Foundations – Conceptual Side View

For the dynamic behaviour standpoint, there aren't major differences between the two configurations but full industrial modules' solutions, being more complex, require longer analysis time.

All the previous can be modularized both as single unit and as multiple units installed on the same module; limitations for multiple units solutions usually come from plant layout and logistics, but also the dynamic behaviour needs to be deeply investigated.

Foundations arrangement

As a matter of fact, the TM foundation' system includes at least two layers: the module's deck (see fig.23) with all the structural elements necessary to create a suitable foundation profiling and the foundations of the whole module.

Figure 23: GTG Module – Machinery Deck

Either the installation on a "3 gimbals" type common baseplate or the installation on AVM are not mandatory and should be used only when there is a not mitigable risk of lack of static and dynamic stiffness of the lower foundation "layer".

Machinery skids bearing plates' planarity have to be ensured keeping into account the status of the art achievable for such big steelworks and adequate erection and installation procedures.

In general the module will be installed on a multipoint foundation made by at least of four (4) resting points; a bigger number of points can be advisable if the module is not a "single lift" one, so to better distribute the static loads and create opportunity for saving structural steel on the machinery deck.

The basic requirements for the aboveground foundation's design and execution are:

- Adequate planarity tolerances;
- Full compatibility with module handling and laydown equipment and procedures (see fig.24);
- Provisions for thermal expansion control (see fig.25).

Figure 24: Onshore Module Aboveground Foundation Detail

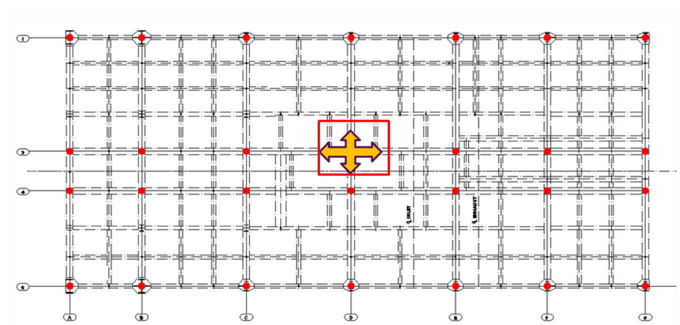

Figure 25: Module thermal expansion driven by shear keys lines

The design requirements for the underground foundations are instead:

- Adequate bearing capacity, driving selection of shallow or deep type;
- Adequate tools to predict dynamic stiffness;
- Flexibility to adapt the frameworks strength design to eventual needs of improving stiffness coming from module dynamic analysis.

Dynamic analysis steps

Modal Analysis

The purpose of the modal analysis is to perform a qualitative check of the dynamic behaviour of a system based on the presence or the absence of significant natural vibration's modes in the closeness (commonly +/-20%, lowered at +/-10 % for 50Hz and above) of the excitation forces, giving in this way a fast "high level" feedback about the feasibility of the modularization.

Since natural frequencies are function of stiffness and masses' distribution, the analysis have to be done when at least the primary structures sizing has been finalized and building a model representing the whole system including TM train.

As said, the main dynamic excitation source coming from a rotor is surely the mechanical synchronous (1xREV) one, but it is also worth to say that there are other sources of excitation that shouldn't be neglected; for example, API 584, Brushless Synchronous Machines 500 kVA and Larger (see ref. [2]), explicitly recognizes "twice the (electric) line frequency" as one of the excitation sources and tells that the natural frequency of the foundation should not occur within 80 % to 120 % of running speed frequency (1xREV +/-20%) or 180 % to 220 % of both the running speed frequency (2xREV +/-10%) and electric line frequency.

In facts the electromechanical interaction in rotating electric machines induces additional forces between the rotor and the stator, called Unbalanced Magnetic Pull (UMP), which are strongly dependent from 2xΩ, being Ω the frequency of the electricity generated/supplied; so, despite the amplitude of such forces is usually not given, some evaluations at modal analysis (or dynamic stiffness) level are advisable.

When earthquakes induce vibrations to the structures from their foundations (ground excitation), modes with a noticeable mass participation (i.e. 5% and above) are usually significant contributors to the overall system's response.

On the other side, for TM modules dynamic analysis, the Mass Participation Factor associated with each mode couldn't be a good indicator of the risk (for the machines) associated to the activation of that specific mode.

Since the target is to analyse vibrations that are induced by harmonic loads acting within the system and directly applied on few locations of the foundation/supporting structures, the engineer has to evaluate which modes can be dangerous, independently from the Mass Participation Factor but in relation with the actual possibility that excitation coming from rotating equipment will activate some natural modes locally.

Coming back to the acceptance criterion, it's worth to say that there are cases (i.e. TM mechanical train having machines running at 3 different and variable speeds) for which almost the whole frequency range between the lowest and the highest is covered without (or apart from small) gaps and so obtaining a "successful" modal analysis could be practically impossible also applying reduced segregation margins.

In these last cases the modal analysis cannot be effective at all to drive preliminary decisions about design and it has to be skipped, passing immediately to the next steps.

On top of the above, other design targets are to verify that vibrations propagating from the machinery bearing point through the module's deck are not such to create troubles both to humans (in case of permanently or temporarily manned areas when machines are in operation) and to auxiliary equipment installed onto the industrial module; for this purpose the criteria of avoiding that natural modes with mass participation > 5% are present in the machine's operating range +/- 20% could be anyway helpful for understanding if there are important risks of excessive vibration propagation from the girders supporting the main equipment to all around.

Dynamic Stiffness Analysis

The Dynamic Stiffness of the "foundation" at TM bearing points has to be checked under test loads versus the requirements that are set from TM OEM with the purpose to validate the hypothesis used for running the (preliminary) rotordynamic lateral analysis and for generating the dynamic loads on foundation.

API RP 684 at this regard says that:
- *when bearing support stiffness, including effects of frequency dependent variation, are less than 3.5 times the bearing (oil film) stiffness values, the support stiffness values derived from modal testing or calculated frequency dependent support stiffness (and damping) values shall be used for the lateral rotordynamic analysis.*

Despite the minimum stiffness threshold should be calculated case by case (i.e. machine by machine) as per manufacturer practices, it also says that:

- *The bearing support stiffness should in most cases be no more than 8.75 E+05 N/mm [8.75 E+08 N/m].*

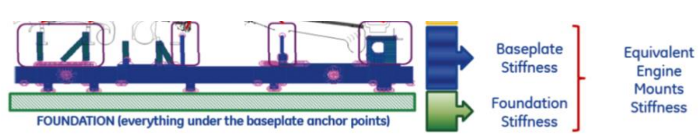

Figure 26: Modularized TM Unit Foundation Stiffness

In case the Dynamic Stiffness shouldn't match the target value given by machinery dept., which could be particularly hard for some kind of modularized solutions (i.e. offshore), this doesn't necessarily mean that the "foundation" design has to be changed, but new process iteration has to be activated.

The actual values have to be communicated to the rotordynamic engineers for running again the lateral analysis and eventually confirm the acceptability; once get this confirmation, the TM train integration engineer have to eventually regenerate the foundation loads.

In general, when the foundation dynamic stiffness goes below the minimum static stiffness required for TM train alignment somewhere within operational range this is not acceptable and the foundation system redesign is required.

Dynamic Response Analysis

The Dynamic Response Analysis is the only the last part of the process, having the main purpose is to guarantee good performance in terms of maximum vibrations' amplitude ("peak-to-peak" or "zero-to-peak") or velocity of the TM foundation (in our case module's machinery deck) under the effect of the given dynamic operating loads at the frequencies within the machine's operating range.

When rotors operating at different speeds are involved, the dynamic response can be obtained by:

- Performing separate steady state analyses, each one involving excitations coming from rotors running at the same speed only, which responses can be later (algebraically) superimposed to obtain the maximum displacements' amplitude at each bearing point;

- Performing, in alternative, a Time History Analysis applying simultaneously the excitations at different frequencies; in this case the results (maximum displacements' amplitude at each bearing point) will be later translated, through Fast Fourier Transformer, in the frequency domain to better understand the weight of the different contributions.

To be noted that TM OEM usually give dynamic loads at each support point of baseplate, decomposing the centrifugal force due to the rotor unbalance in the sum of two orthogonal harmonic loads (vertical and transversal), using their know-how for applying the most appropriate transfer function from the shaft line level to the machinery skid foundation level.

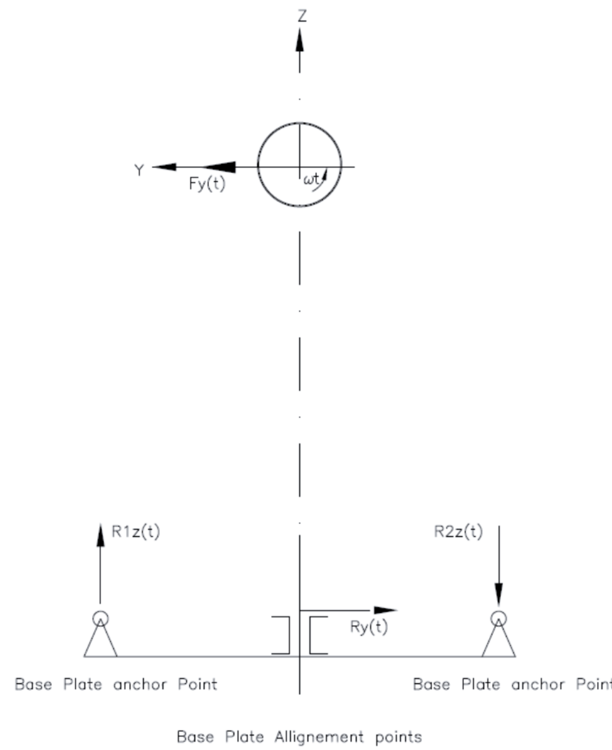

Figure 27: Dynamic Loads on Foundation from Horizontal Harmonic Load at Shaft Line Level

Dynamic analysis modelling tips

The FEA model has to be adequate for representing the real system and capturing its behavior, being enough accurate where it's needed but also "lean" where possible, so to reach an optimal compromise between accuracy and elapse time.

With the above premise, despite in principle the same structural modeling used for the static analysis of a module should be used also for the dynamic analysis, it can be also acceptable that the two models are not the same for a lot of practical reasons.

Two of the main improvements that can be done to simplify the calculation model and reducing elapse time are:
- eliminate the secondary to secondary steelworks;
- consider the friction effect for redefining module to foundation horizontal restraints under operational loads.

There are other kinds of tricks that don't go in the direction to simplify the model but are aimed to reach a better accuracy.

An example is relevant to the machinery train modelling that should be done in way to position the masses at the right elevations and having the dynamic loads at machinery-module interface as given on machinery foundation drawings.

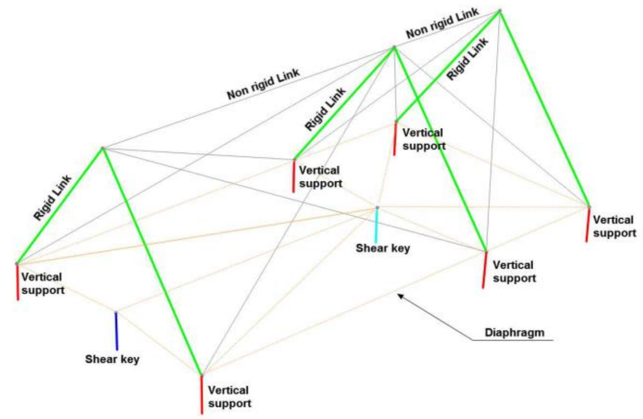

Figure 28: Machine modeled using link elements with different properties and suitable geometrical arrangement

Another example the dynamic loads applied at the bearing points are usually and conservatively imputed as concentrated loads; this although each bearing "point" has a not negligible surface.

Analysis experience demonstrated that "complicating" a bit the model to create a loads' distribution closer to the reality will help in matching the acceptance criteria for the different analysis steps.

A last example, since the model have to capture the response from a dynamic excitation coming from the system itself, at least main girders/elements directly supporting rotating equipment shall be modeled (or simply transformed by launching an appropriate auto-mesh routine on the calculation software) as shell/plate elements to better capture local modes.

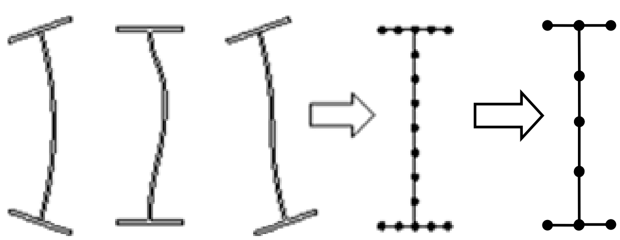

Figure 29: Steel Module Main Beams Meshing Optimization

Dynamic loads & acceptance criteria

The dynamic loads are variable excitation force given as function of time for a fixed frequency, but the dynamic loads reported in the TM foundation drawings are relevant to nominal (100%) shaft speed; for different speed values within the operating range the dynamic loads will change keeping into account that their value is directly proportional to the square power of operating speed "n" (RPM).

The acceptance criteria for this analysis is that in the TM train operating range these vibrations shall be less than the allowable value for each machine.

So the dynamic response analysis results have always to meet the acceptance criteria indicated by manufacturers (usually on the machinery Foundation Drawing's notes) according to their proprietary Design Criteria.

This concept is paramount because only in this way it's ensured that dynamic loads for the design and acceptance criteria are generated under consistent hypothesis.

To be noted that since 5th edition (2011) the standard API 616 for Gas Turbines started in some way to put in relation the rotor unbalance used to generate the foundation loads and the corresponding acceptance criteria by putting in parallel vibration limits from the standards ISO 7919 and ISO 10816, respectively relevant to rotating and non-rotating parts, assessed against four evaluation zones established as follows:

Zone A: Newly commissioned machines.
Zone B: Unrestricted long-term operation.
Zone C: Unsatisfactory for long-term continuous operation
Zone D: Damage to the machines

Table 6: Vibration Limits According to ISO 10816-4 and ISO 7919-4 (from API 616, 5th)

Zone	Bearing Housing Pedestal Criteria Vibration Limits per ISO 10816-4 mm/s (RMS)	Shaft Relative Vibration Limits per ISO 7919-4 $A_{(\rho-\rho)}$ µm
A	≤4.5	≤4800/ $\sqrt{N}$
B	4.5 to 9.3	4800/ $\sqrt{N}$ to 9000/ $\sqrt{N}$
C	9.3 to14.7	9000/ $\sqrt{N}$ to 13,200/ $\sqrt{N}$
D	≥14.7	≥13,200/ $\sqrt{N}$

The same concept apply when evaluating the dynamic response analysis results versus criteria different from machinery ones, for example when evaluating the severity of human exposure to whole-body vibration.

In this case the acceptance criteria wouldn't be negotiable; the machinery engineer has to generate dynamic loads (typically considering the higher rotor unbalances acceptable for unrestricted long term operations, corresponding to machinery alarm level) that are consistent with the contractual / regulatory acceptance criteria.

In case of lack of mandatory regulations it's strongly advisable to apply criteria based on studies that drilled down the strong dependency of the vibrations' severity from the frequency, such the ISO 2631.1 one that has been recognized from several national standards organizations such BS and AS.

Experimental validation of results

The testing of the modularized units in a duly equipped yard and with a proper testing strategy prior of the module's shipment offers a great opportunity to validate the results of the Dynamic Response Analysis and to strongly mitigate the risk of performing troubleshooting at final installation site.

At the moment of the test the unbalance loads are those of newly commissioned machines, so the results obtained under the effect of normal operational design loads will be duly scaled (i.e. referencing to the table 1 above) to set the baseline for comparing analysis results with test results.

Replicating as close as possible the same above ground foundation layout at testing yard and final installation site is paramount to obtain the same kind of behavior of the module structural steel under internal dynamic excitation at the two locations; differences in the underground part of the foundations should instead influence mainly the behavior of the whole system and the modes having a big participation mass.

Based on the above considerations once the performances at testing yard are validated, it will be possible to predict the performances at final installation site as well.

To do this, a proven soil and/or pile characterization at both sites would be also paramount (see also ref. [7]).

Strategies for preventing/fixing site issues

Foundations' Execution and Module Set-Down

The module behavior is strongly related to the actual restraint's conditions, which could be different from those used for the analysis, especially for a multipoint installation.

It's paramount to set both a foundation's execution strategy and a module set-down procedure aimed to ensure that the all the rests are effectively engaged.

Localized Structural Steel Members Excessive Vibrations

Vibration's amplitude in transversal direction could be significantly reduced by appropriate stiffening, while the same occurrence in vertical direction could be mitigated by adding permanent ballast if feasible.

Excessive Dynamic Response at Unpredicted Frequencies

For all equipment and/or frequencies where dynamic loads are not given it can be preventively advisable investigating the dynamic stiffness of the supporting structure keeping the "as is" static stiffness value as benchmark.

If the issue suddenly happens during test in correspondence to auxiliary equipment the best and cheap strategy could be to select and install duly selected AVM.

Conclusions

The TM modularization is often really attractive from the business standpoint but requires a strong engineering effort for several reasons and the system dynamic behavior is probably the main one, representing a road block for the feasibility.

Since the TM are installed on a "flexible" foundation, some of the common practices relevant to TM installations on block foundations are not fully applicable, the analysis process is longer and fixing issues at design level requires both an open mind approach and a lot of attention to details.

The modal analysis may give important feedback in short time about the modularization feasibility but, when passed, needs always further drill down.

The dynamic stiffness analysis is paramount for validating the hypothesis for the rotordynamic analysis; in case it fails the alternatives are a new run of the rotordynamic analysis and/or perform the industrial module/foundations redesign. It's strongly advisable to avoid having this kind of risk in project execution phase, launching feasibility studies for each and every modularization novelty.

Moving to the Dynamic Response Analysis, both dynamic loads and acceptance criteria for the machinery have to come from the OEM, having the right knowledge to define both the transfer function for transforming the centrifugal force at the shaft line due to rotor unbalance in harmonic loads at machinery bearing points and consistently set the acceptance criteria for safe long term operations.

Acceptance criteria for assessing the severity of the vibrations that may propagate from TM through the steel deck up to other equipment or manned posts (i.e. Human Machine Interface within a Local Control Cab) couldn't be negotiable and so in this case the TM OEM has the duty to apply dynamic loads consistent with the acceptance criteria.

The experimental validation of the analysis results is a great residual risks mitigation opportunity that can be kept, having the right processes, expertize and tools, during the modularized unit's testing prior to the shipment.

REFERENCES

[1] API 616, 5th 2011: "Gas Turbines for the Petroleum, Chemical, and Gas Industry Services"

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[4] API 661, 6th 2006 (ISO 13706-1) "Air-Cooled Heat Exchangers for General Refinery Service"

[5] API 584, 3rd 2008: "Brushless Synchronous Machines 500 kVA and Larger"

[6] API661, 6th 2006: (ISO 13706-1) "Air-Cooled Heat Exchangers for General Refinery Service"

[7] "Frequency dependence of piles 'dynamic stiffness", Technical Brief presented at ATPS 2016.