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DYNAMIC PROPERTIES ASSESSMENT AND UPDATING OF LARGE DIESEL SHIP ENGINES

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Abstract

Large diesel engines are a challenging product for engineer: they are large, heavy, dangerous. Modelling is lengthy and cumbersome, testing is expensive and specific.

In this paper, the authors will highlight some of these problems and their influences in the correlation between test and numerical data set, and at the same time show that a good match between numerical simulations and test can be achieved with the appropriate procedure.

The paper reports about the analysis and updating of a “genset”, an aggregate formed by tree main parts: the diesel engine, the electric generator and the common base frame. These three elements combined weight roughly 200 tons, occupying a space of about 260 cubic meters.

INTRODUCTION

For the production of large diesel engines, qualities like performance and reliability are of top most importance. These have to be achieved with the maximum net margins possible and Wärtsila, as world leader in energy systems, where such engines are adopted, devotes a lot of efforts to constantly improve the quality of its methodologies and procedures as the following paper will show.

The paper reports about the analysis of a “genset”, an aggregate formed by tree main parts: the diesel engine, the electric generator and the common base frame. These three elements combined weight roughly 200 tons, occupying a space of about 260 cubic meters.

The finite elements model and the measured FRFs acquired during a modal test in "cold" conditions are available. The distinction between "cold" and "warm" conditions is rather important due to the large differences that appears for the behaviour of the visco-elastic mounts, the lubricant distribution and the characteristics and the contact forces between the moving elements.

The first step is to align the numerical and the experimental models, forcing the linearity of the first one around the "cold" operational conditions. Due to the limited modelling of the joints between the different parts, those elements and their characteristics are considered as variable parameters for the updating procedure.

This selection is also supported by the fact that there is a frequency overlap between the rigid body modes of the subcomponents and the flexible modes of the complete genset, demonstrating the influence of the resilient mounting among the different part and elastic coupling joint between engine and alternator.

In order to match the experimental mode shapes with the calculated ones and to perform the sensitivity and updating analysis (rubbers and coupling interfaces…) the commercial software FEMTools is used both in fully automated and manually controlled procedure.

GENSET FINITE ELEMENT MODELLING

The genset is a complex system formed by different elements:

- A 16 cylinders diesel engine with a wet total mass of 107600 kg .
- A turbo charger system.
- An electrical generator, weighting 54000 kg.
- A common baseframe with oil sump function, with a wet total mass of 34500 kg.
- A flexible coupling between the engine and the generator.
- Steel and resin block and chokes to assure the connection among the different elements of the genset and between the common base and the ground.

All these elements together weight about 200752 kg and occupy a space of about 3.9 by 5 by 13 meters. All the different parts are modeled with shell and brick elements within the commercial FE modeler IDEAS except for the blocks and the chokes that are represented with spring-damper elements whose initial stiffness characteristics were derived from the product data sheets [1] [2].

The full FE model (Fig.1) has 739,266 dof. It has been exported to FEMTools where both rigid and flexible modes are computed in a frequency range between 0 and 150Hz.

Specifically, the rigid body modes of the genset as a whole, on which the attention of the updating and correlation activities will be focused on, are listed in Table 1. It's worth noticing the closeness between the different natural frequencies and the proximity of some of these modes to the first harmonics and sub-harmonics of the engine operational running speed of 600 rpm.

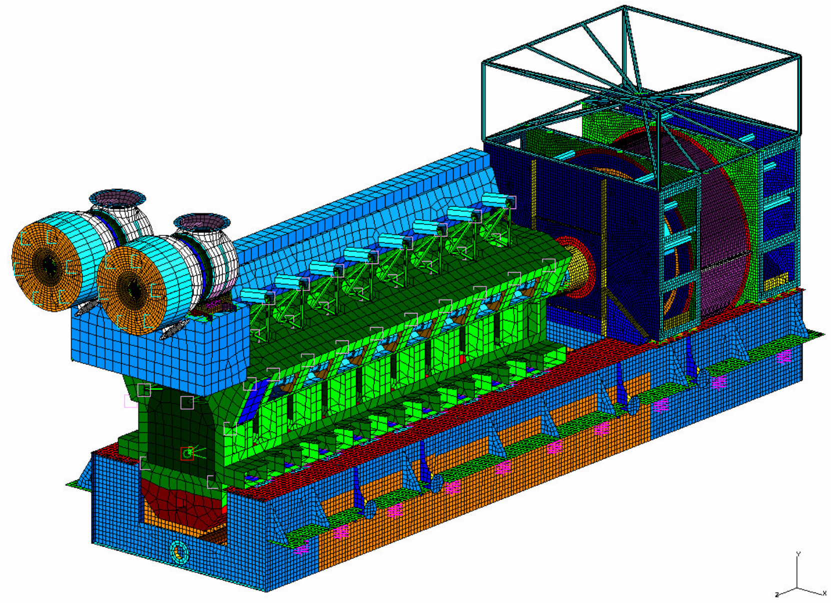


Figure 1 – The genset final configuration

Table 1 – The geneset rigid body modes

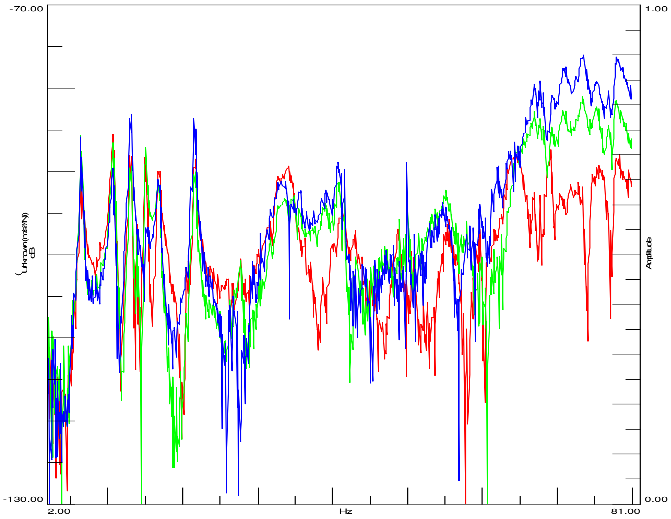
Mode	Frequency (Hz)	Description
1	4,2	Transverse Trasl. X
2	4,66	Axial Trals. Z
3	6,18	Vertical Trasl Y
4	6,25	Yawing
5	7,86	Pitching
6	12	Rolling
7	15.9	Main Bending
8	19,7	Main Torsion

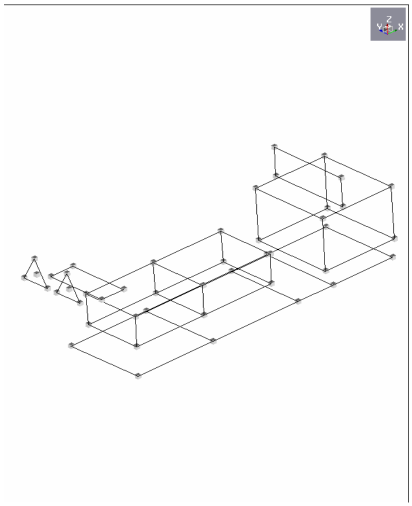
EXPERIMENTAL DATA

The whole structure is tested at the Wärtsila plant in Trieste, Italy, with impact hammer excitation and rowing accelerometers with a sampling frequency of 160 Hz and a frequency resolution of 0.1 Hz.

Totally 48 measurement points in three orthogonal directions are considered for a total of 144 responses; of which 30 on the common base frame 10 points, 36 on the engine block, 36 on the turbo chargers, and 36 on the alternator.

Fig.2 shows some of the typical acquired FRFs: at the low frequencies the rigid body modes are rather evident and well separated while at higher frequencies the modal overlap and the low S/N ratio degrade the data quality and increase the difficulties of the identification procedure . Fig. 3 shows the measurement mesh where the different





components can be easily identified.

Figure 2 – Typical Measured FRFs

Figure3- Measuring Mesh

The structure was excited vertically in a corner of the engine block after having performed a pre-test analysis of the aggregate, considering its' accessibility and the rigidity of the component in the specific point.

CORRELATION AND UPDATING

Even if the numerical and experimental modal analysis delivered realistic mode shapes, their correlation, at first, looked rather poor, as table 2 shows:

Table 2 – FEA and EMA paired modes

Mode Pair		FEA Mode	Frequency (Hz)	EMA Mode	Frequency (Hz)	Diff.	MAC
1	Transversal	1	4,2	1	6.1	-31,15	56
2	Longitudinal	2	4,66	2	6.41	-27,30	55
3	Vertical	3	6,18	5	11,52	-46,3	71
4	Yawing	4	6,25	3	6,89	-9,29	68
5	Pitching	5	7,86	4	10,78	-27,22	58
6	Rolling	6	12	7	16,8	-28	72
7	Bending	7	15.9	6	13,9	14,4	51
8	Torsion	8	19,7	8	20,9	-5,7	92

and fig.4 exhibits.

The reasons behind such a poor correlation can be manifold, but mainly due to some simplifications in the FE model (the fluids modelling, the elastomeric supports characteristics (varying with time and temperature), the positions of the pistons within the model and so on) and due to the difficulties to excite with an uniform energy level such big, heavy and damped structure.

To increase the correlation between the two mode sets, at least in the lower frequency region, the updating of the connection elements parameters is carried out.

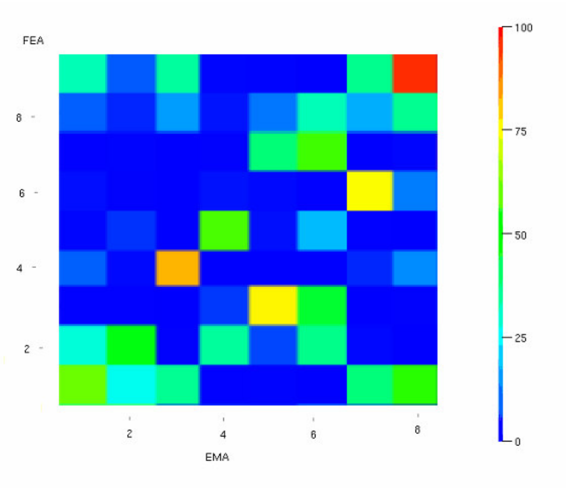


Fig.4 CROSSMAC matrix Numerical Vs Experimental-rigid body modes

Initially, a fully automated procedure is pursued, letting the software match the modes, select the appropriate parameters and so on, unfortunately with rather poor results mainly due to the modes frequency closeness and similarity.

A second procedure is than considered: this consists of a manual mode matching and definition of the critical updating parameters based on a extensive engineering knowledge of the structure. In this case the attention is focused on the relative relevance of the different elastic elements, their different loading conditions as function of their positioning within the genset, the stiffness coupling between the different directions and the importance of the movements of the rotor of the electric generator.

At first a sensitivity analysis, within FEMtools, on a total of 40 parameters of the elastic connections between the different parts of the assembly with respect to the six rigid body modes is conducted. Fig.5 and Fig.6 show the analysis results for the whole parameter sets and the four most influencing ones demonstrating that the dynamic behaviour of the genset in the low frequency band is dominate by a very limited amount of parameters.

The displayed sensitivity results are used to tune model. The updating strategy is divided in two steps:

- Updating1 using the GLOBAL parameters sensitivity information, 5 iteration, where all the elastic properties referencing the same FE property were assigned the same value.

- Updating2 using LOCAL parameters. The model tuned in the first run is then used as starting point for a new model updating run, 20 iteration, where each elastic property may be different.

To reduce the CPU time for each model updating iteration, each part of the assembly is condensed as SuperElement using the Craig-Bampton reduction technique, passing from a system of 739,266 dofs down to only 1,766 dofs.

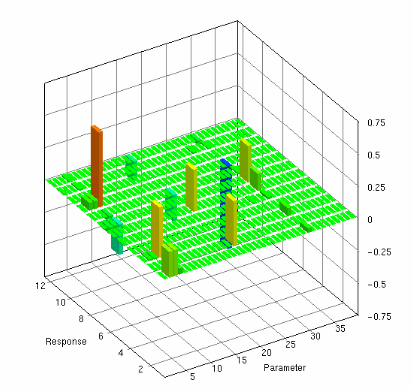


Fig 5 GLOBAL Sensitivity matrix (40 parameters)

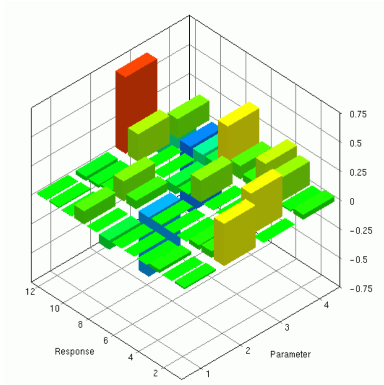


Fig 6 GLOBAL Sensitivity matrix – Most sensitive parameters (4)

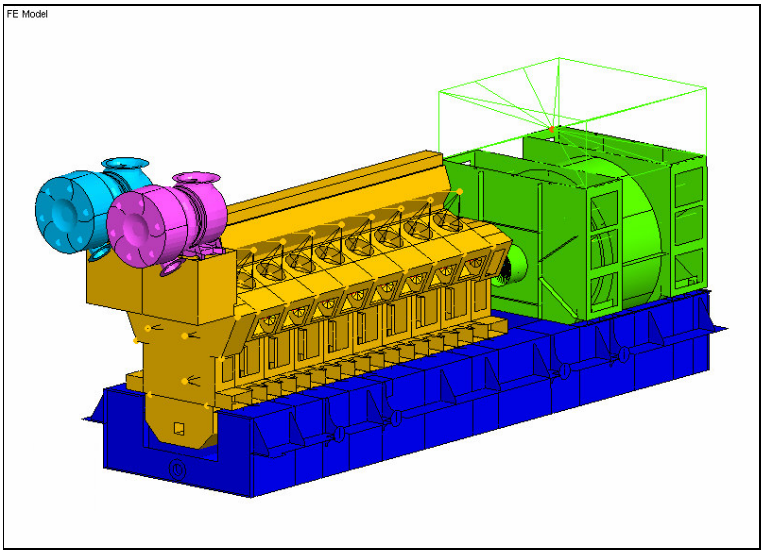


Fig.7 The Superelement Assembly

Each superelement is statically condensed on the translation dofs of the interface nodes (connection with elastic elements) and a given number of fixed interface component modes for each part is added to preserve the accuracy of the results in the frequency range. The superelement assembly (shown in Fig. 7) is validated with the reference un-condensed model, to ensure it's accuracy. For each updating iteration the analysis time decreased from 1,600 s down to less than 30 s, without loss of accuracy.

Table 3 shows the paired modes results. After 5 iterations the average frequency shift an MAC are improved to 7.7 Hz and 75.9. Please notice that now all the 6 experimental modes are paired to an analytical ones.

Table 3 – FEA and EMA paired modes after GLOBAL updating.

Mode Pair		FEA Mode	Frequency (Hz)	EMA Mode	Frequency (Hz)	Diff.	MAC
1	Transversal	1	6,1	1	6,1	0	61
2	Longitudinal	2	5.72	2	6.41	-10.82	70.3
3	Yawing	3	7.15	3	6.89	3.66	83.7
4	Pitching	4	9.87	4	10.78	-8.44	71
5	Vertical	5	11.07	5	11.52	-3.92	86.1
6	Bending	7	16.16	6	13,9	16.32	58.2
7	Rolling	8	16.29	7	16.84	-3.29	86.1
8	Torsion	9	19,8	8	20,9	-5,26	94

The results of the sequent LOCAL updating are shown fig.8 (cross MAC between updated and experimental mode shapes) and in Table 4. The average frequency shift is now reduced to 1.76 Hz, and at the same time the mean MAC value for the target modes is now increased to 84.2. Fig. 9 and 10 show the 6th EMA mode paired with correspondent analytical results, before and after the tuning of the stiffness properties of the different coupling elements.

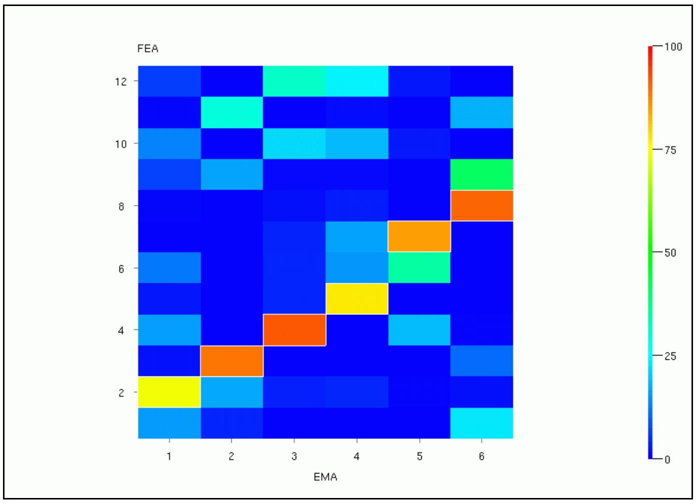


Fig.8 CROSSMAC matrix Numerical Vs Experimental after LOCAL updating

Table 4 – FEA and EMA paired modes after LOCAL updating.

Mode Pair		FEA Mode	Frequency (Hz)	EMA Mode	Frequency (Hz)	Diff.	MAC
1	Transversal	1	6,36	1	6,1	5,01	57

2	Longitudinal	2	6.47	2	6.41	0.83	73.7
3	Yawing	3	6.84	3	6.89	-0.73	88.5
4	Pitching	4	10.34	4	10.78	-4.05	91.4
5	Vertical	5	11.63	5	11.52	0.95	76.9
6	Bending	7	14.24	6	13.9	2.5	84.5
7	Rolling	8	17.09	7	16.84	1.5	90.1
8	Torsion	9	18,4	8	20,9	-11,96	87

CONCLUSIONS

This paper represents a first try of applying the updating approach to large naval diesel engine. This system presents different difficulties both for what concerns the modeling and the testing but with appropriate technological solutions, like FEMTools, and with a lot of critical engineering judgments some very useful and very physical results have be achieved.

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[2] Data Sheets Vibracons SM48 -Vulkan 140- Rato-S

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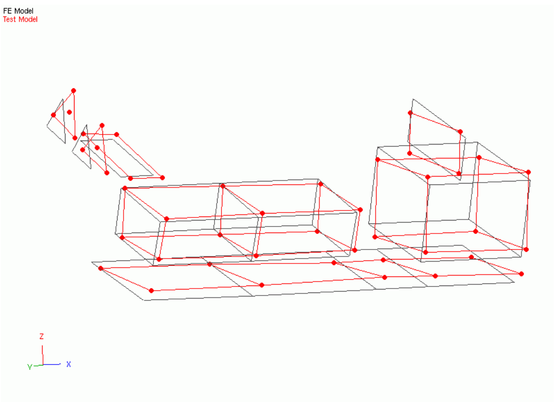

Fig.9 FEA #7 vs EMA #6 – 16.8 Hz
before updating
Frequency Error = -35.4%
MAC = 73.0

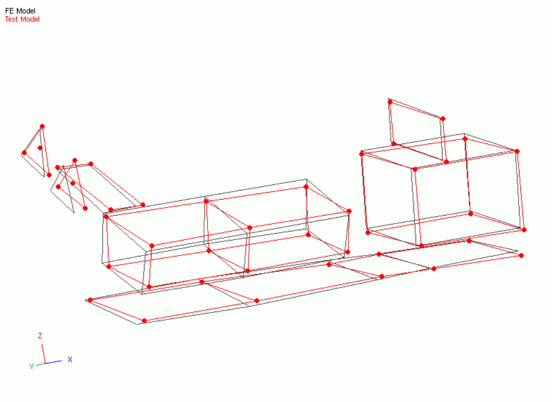

Fig.10 FEA #6 vs EMA #6 – 16.8 Hz
after updating
Frequency Error = 1.5 %
MAC = 90.1