

NUMERICAL VIBRATION ASSESSMENT ON LARGE 2-STROKE MARINE DIESEL ENGINES

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Keywords: Vibration Simulation, Reciprocating Engine, Modal Analysis, Harmonic Response Analysis, Modal Assurance Criterion.

Abstract. *Vibration measurement on large 2-stroke marine diesel engines is a very cost intensive and time consuming process. Each single measurement and optimization loop has large expenses as consequence. Therefore simulation plays an important role in the design process of such an engine.*

For the vibration assessment of the structure of Wärtsilä 2-stroke engines a simulation process was developed. In a first step a modal analysis is performed. Then the engine structure is loaded by the gas pressure resulting from the combustion process and by the inertial forces arising from the moving parts. A harmonic response simulation is carried out over the complete operation speed range of the engine. The resulting Campbell and order diagrams supply information about excited mode shapes, critical engine speed ranges, vibration amplitudes, etc. and allow optimizing the engine structure regarding minimization of vibrations.

The complete simulation procedure is demonstrated on the example of a 2-stroke diesel engine which is destined for the propulsion of a cargo vessel. The obtained numerical results are then proven with measurement data.

1 INTRODUCTION

Wärtsilä 2-stroke marine Engines have a power output between 4'550 kW and 80'080 kW. The cylinder bore diameter of the 5 to 14 cylinder engines varies between 350 mm and 960 mm. Due to the large machine size (Figure 1) prototyping of 2-stroke marine engines is difficult and extremely cost-intensive. Each single measurement and optimization loop has large expenses as consequence. Therefore simulation plays an important role for the design of such an engine. In most cases the engines are only verified by “virtual” or “computational prototyping” before the first engine that is ordered by a customer is audited on the test bed.

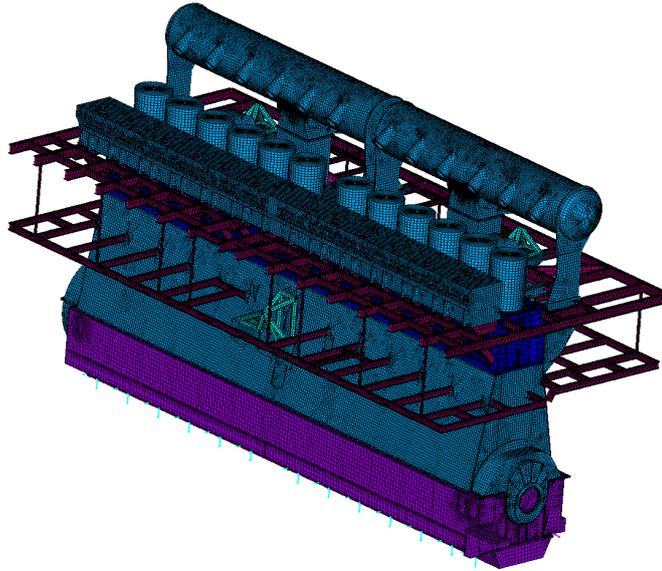


Figure 1: Large Wärtsilä 2-stroke engine destined for the propulsion of large container vessels..

For an efficient prediction of the vibration behaviour of the structure of a 2-stroke marine engine an optimized simulation process was developed. The method has to be accurate enough for identifying vibration critical regions on the engine components and to deliver results in an acceptable period of time.

2 THE SIMULATION PROCESS

The proposed methodology for vibration assessment consists of three parts. First a simplified finite element model of the engine is built. In the next step a numerical modal analysis demonstrates the mode shapes the engine can adopt when vibrating with its eigenfrequencies. Finally the finite element mesh is loaded with the gas and inertial forces occurring during operation and a harmonic response simulation by modal superposition is carried out.

2.1 The finite element model

For achieving an efficient use of resources during simulation the finite element model of such a complex structure, as it is a 2-stroke marine engine, has to be optimized considerably (Figure 2). For simplification of the model bolted components are attached to each other by means of multipoint constraints (MPC). Unnecessary bores and surface protuberances are eliminated. Most components of WÄRTSILÄ 2-stroke engines are built with welded plates. In order to save degrees of freedom, these plates are modelled with shell elements instead of using solid elements. Components of lower interest or parts from external suppliers whose geometry is not properly known like turbochargers and auxiliary blowers are discretized by

using lumped masses. After all simplifications the finite element mesh of a 2-stroke engine can be reduced to a mesh size of 1.5 to 2.5 million of FE-nodes, depending on the engine size.

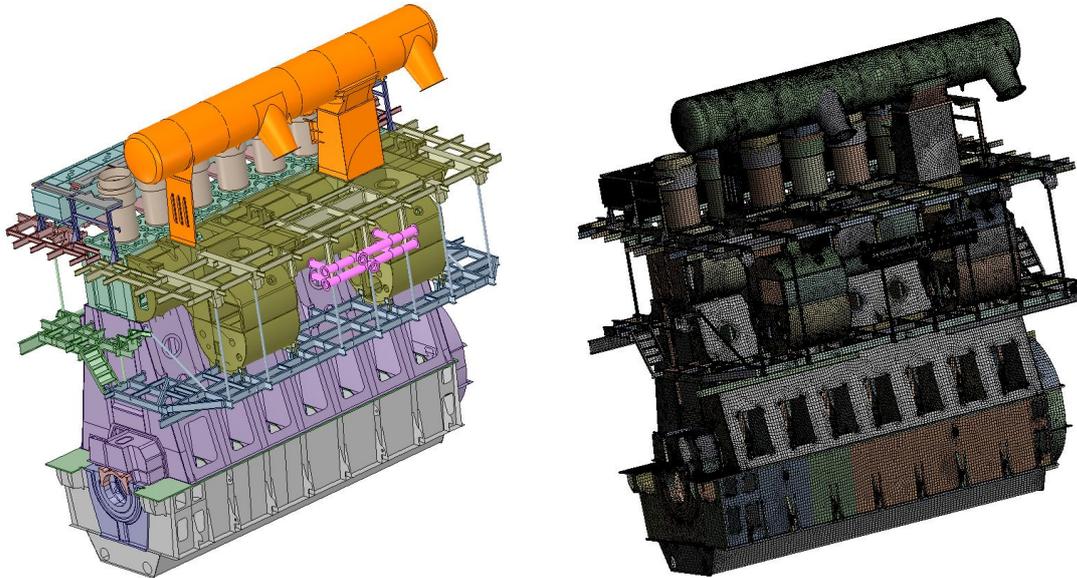


Figure 2: Optimized finite element model of a 2-stroke marine engine without (left) and with FE-mesh (right).

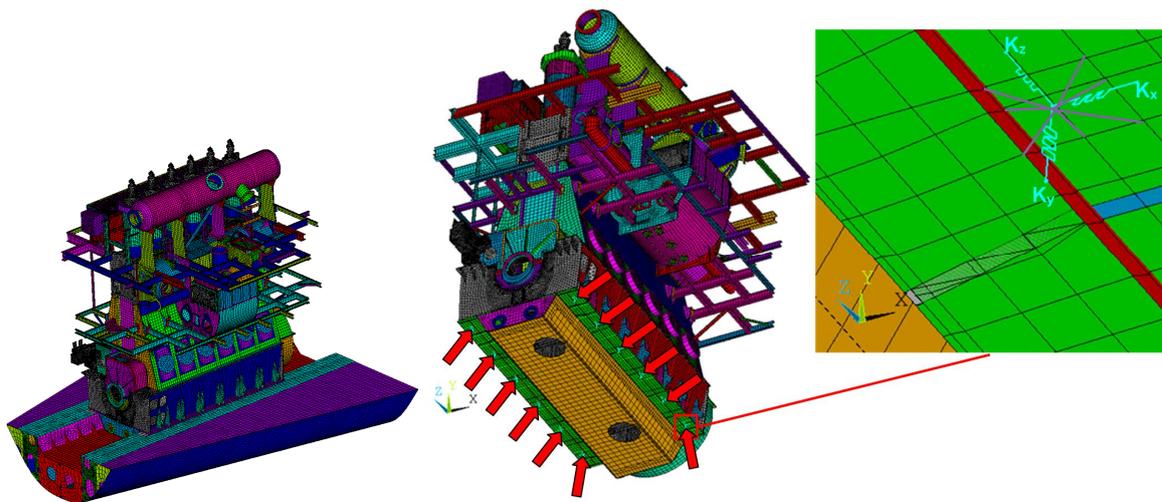


Figure 3: 2-stroke engine seated on a cut-out of the ship hull (left) and engine on a substitutory spring foundation (right). At the tip of each arrow a spring package is applied. Each package contains one spring acting in transversal, one in vertical and an other in longitudinal direction.

Only in very few cases ship builders supply information about the configuration of the supporting plates in the ship hull (Figure 3 left). Usually the crucial information about the constitution of the engine seat in the vessel is unknown. As the engine bed stiffness determines considerably the engine's H-mode (transversal vibration mode of the complete engine) and L-mode (longitudinal mode of the entire engine), under these circumstances the engine seat is simulated by means of a substitutory spring foundation consisting of a set linear springs (Figure 3 right). The stiffness K_x in transversal, K_y in vertical and K_z in longitudinal direction are adjusted by means of an in-house tool, so that the eigenfrequency of the H-mode and of the L-mode meet the modal frequencies that are known from a vibration measurement on vessels with a similar engines. Anyway these two mode shapes are of minor importance in

the engine development process, because their vibration amplitudes are mostly determined by the constitution of the interface between the engine and the ship. These vibration modes can be attenuated by e.g. introducing supporting stays between the engine and the ship hull or by the application of a compensator consisting of rotating masses which transfer a force on the engine that acts in counterphase to the mode shape. The need of one of these measures is usually determined during the first engine test on the vessel.

2.2 The modal analysis

The second part of the simulation process is the modal analysis. This very important component of the calculation process helps to understand the structure's vibration mechanism (Figure 4) [1]. It allows locating the points on the structure on which to apply a design modification for avoiding that a determined mode shape occurs.

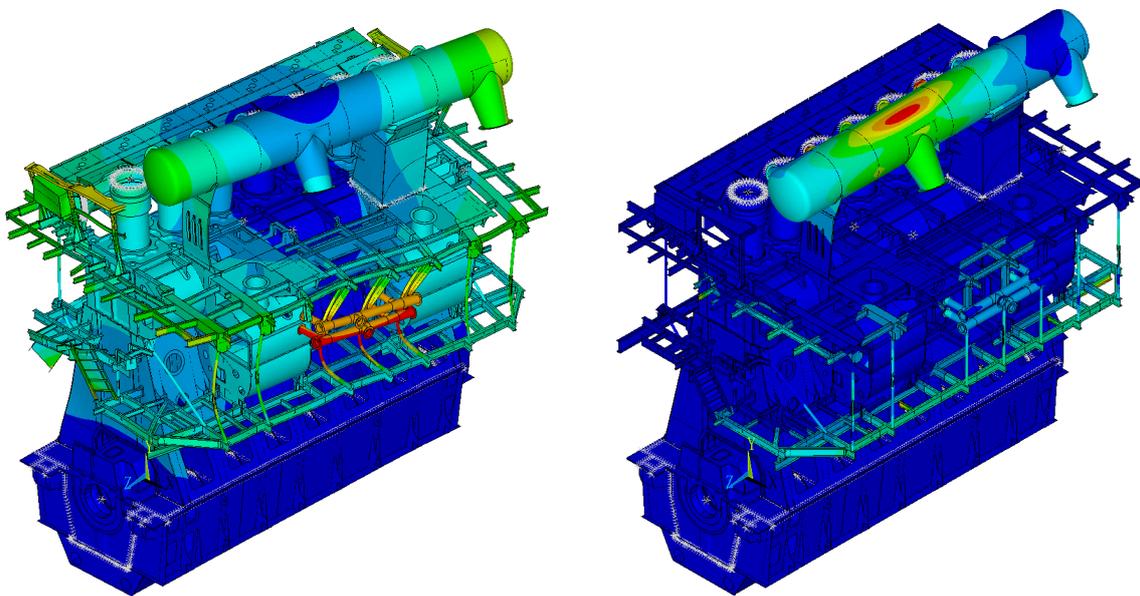


Figure 4: Engine Torsion mode around the vertical axis (X-mode, left) and mode shape of the exhaust manifold (right) on a marine 2-stroke engine.

The modal analysis also delivers the eigenfrequency at which a determined mode shape can be excited. Knowing this resonance frequency helps to determine critical operation speeds of the engine.

2.3 The harmonic response simulation

The third part of the simulation chain consists of the harmonic response simulation. This component of the calculation process reveals the dynamic reaction of the engine structure to the gas forces occurring during combustion in the cylinder and to the inertial forces arising from the moving parts (crankshaft, connecting rod, crosshead, piston, etc.) [2], [3], [4], [5]. The experience has shown that considering the engine structure's first 100 mode shapes for the harmonic response simulation by modal superposition keeps a good balance between accuracy and simulation time.

Moving parts are considered in the simulation model only in form of concentrated mass points that are attached to the FE-mesh at the interface points (A, C and D in Figure 5). The gas and mass forces acting at these points (B and A, C and D respectively) are determined with the in-house tool EnDyn (described extensively in [6]). This software computes the dy-

dynamic load at the interface points in the frequency domain for the complete engine speed range by considering nonlinearities at the bearings, the deformation of parts of the driving train (e.g. each single crank) and the firing order of the different cylinders. For allowing to code phase delays and excitation amplitudes ENDYN supplies a file with complex load data at the interface points. This file then serves as input data for performing a harmonic response simulation of the engine structure by using the FE-solver.

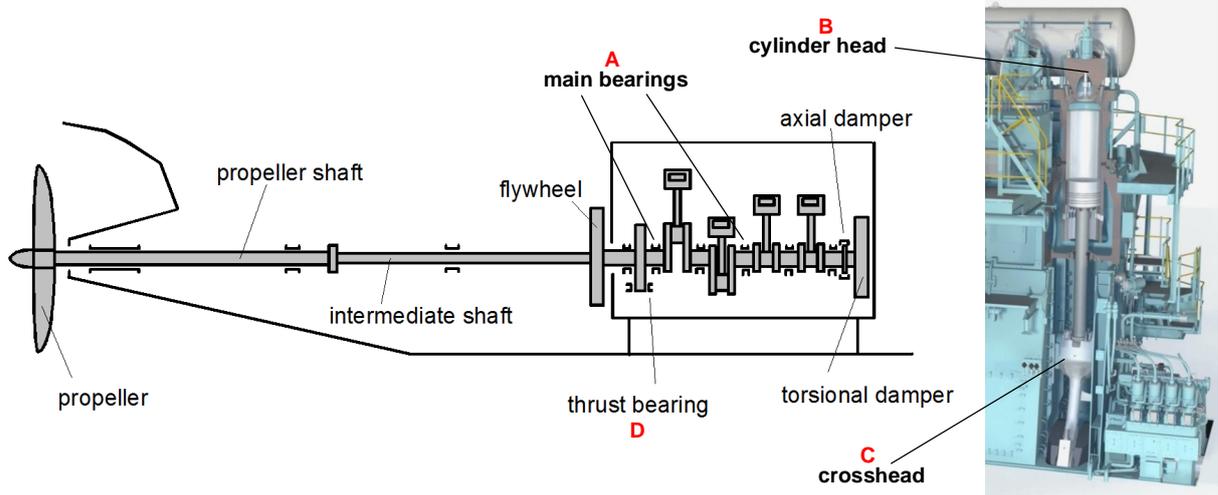


Figure 5: Installation of a 2-stroke marine engine on a vessel. The interface points on the structure on which gas and inertial forces act are marked with the letters A to D.

Unfortunately the damping properties of large 2-stroke engines are quite unexplored. On such a machine damping occurs mainly due to friction between bolted parts. As the location of the regions where deformations take place during vibration changes for each mode shape also the modal damping properties vary from mode to mode. For the simulation a universal damping ratio ξ of 4%, from which it is known by experience that it is applicable to many mode shapes of the analyzed marine engines, was assumed for all superposed eigenforms. As the target of the calculations is comparing designs with each other and analyzing the impact of modifications on the structure this approach is completely licit.

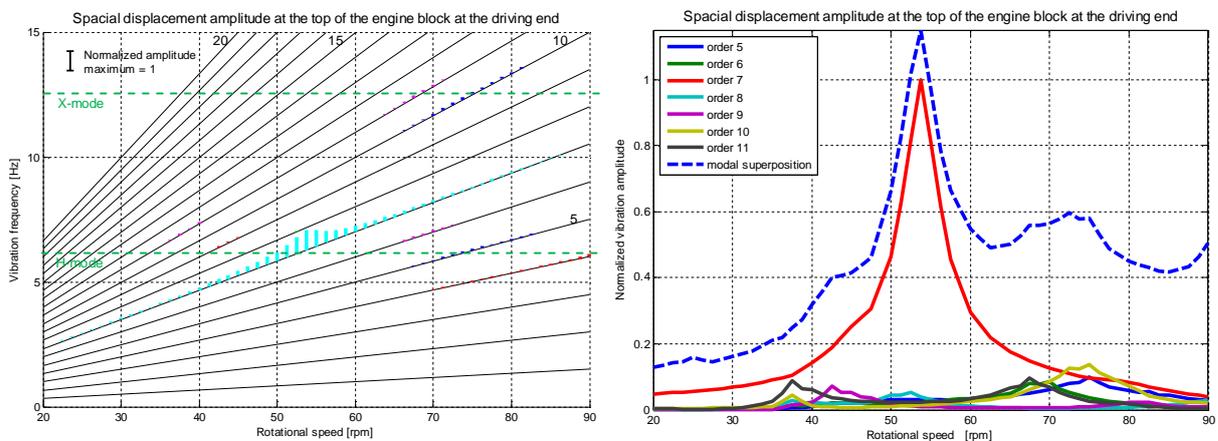


Figure 6: Numerically determined Campbell (left) and order diagram with modal superposition curve (right) for the top of the engine block at the driving end of a 2-stroke marine engine. The diagrams are normalized so that a value of 1 corresponds to the largest modal amplitude (H-mode excited with 7th order load in this case).

The results of such a harmonic response simulation by modal superposition can be visualized in form of a Campbell or an order diagram (Figure 6 shows both diagrams for a point on the cylinder block of a 2-stroke marine engine at the driving end). The Campbell diagram contains more information than the order diagram. In the diagram the vibration amplitude is represented in form of magnitude lines in dependency of the engine's rotation speed and of the vibration frequency. Order lines help to identify the engine order at which a resonance occurs. The diagram allows an easy recognition of engine mode shapes like the H- and X-mode in this case. The advantage of the order diagram is that amplitude relations can be determined more easily. Both diagrams in Figure 6 show clearly that in case of the analyzed engine the 7th order is predominant in the response amplitudes.

3 APPLICATION AND VALIDATION OF THE SIMULATION PROCESS

On large 2-stroke marine engines with many cylinders the exhaust manifold is subdivided into two parts (Figure 7). Both exhaust pipes are linked by a compensation bellow. On a test-motor a vibration measurement revealed that during engine operation both exhaust manifolds vibrate with an antiphase mode shape. For understanding this dynamic phenomenon and for investigating the effects of such a motion on the connecting link between both pipes a finite element model of the engine was built and a numerical modal analysis was performed.

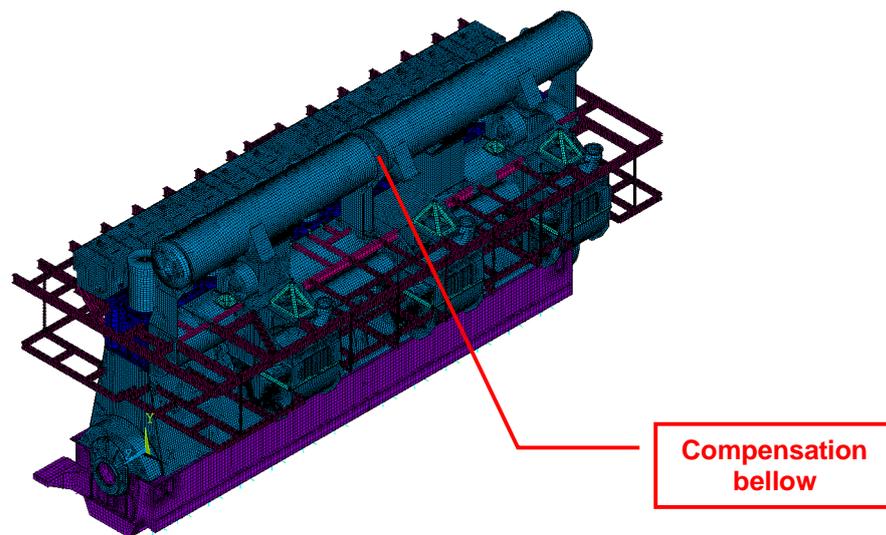


Figure 7: Finite element model of a large 2-stroke marine engine with an exhaust manifold that is subdivided into 2 parts. A compensation bellow serves as link between both exhaust pipes.

For understanding which mode shape was measured on the test engine the modal assurance criterion (MAC) was used [7], [8]. A MAC-table (Figure 8 left) was built by computing its elements with (eq. 1), where $\{\varphi_C\}$ and $\{\varphi_M\}$ are the calculated and the measured modal vector respectively. The indices c and m denote the c^{th} calculated and the m^{th} measured modal vector and the operator T signifies transposition.

$$MAC(c, m) = \frac{|\{\varphi_C\}_c^T \cdot \{\varphi_M\}_m|^2}{(\{\varphi_C\}_c^T \cdot \{\varphi_C\}_c) \cdot (\{\varphi_M\}_m^T \cdot \{\varphi_M\}_m)} \quad (1)$$

In addition, a matrix representing the difference in eigenfrequency between the calculated mode shape c and the measured mode m was generated. Both tables allowed to pair measured

and calculated mode shapes and to identify the antiphase vibration mode that was measured on the engine (Figure 8 right).

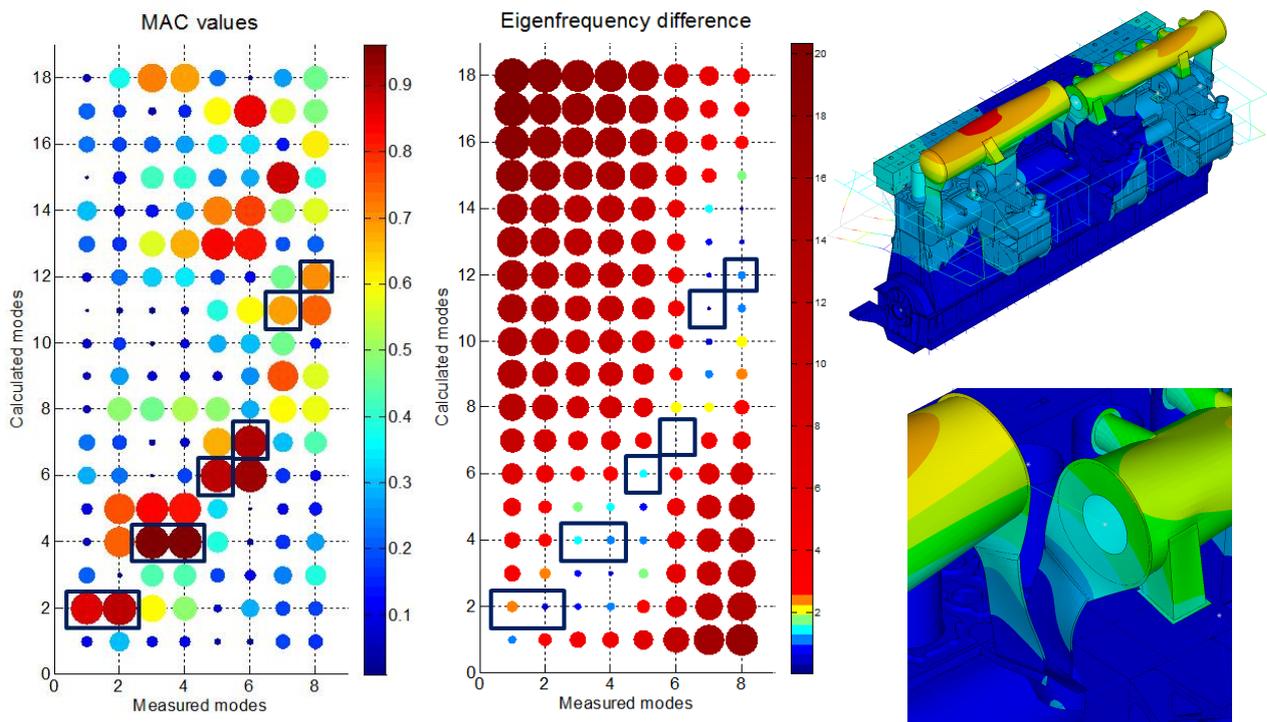


Figure 8: MAC-table and matrix with eigenfrequency differences for attributing measured eigenforms to simulated mode shapes (left) and acting antiphase mode shape reproduced by simulation (right).

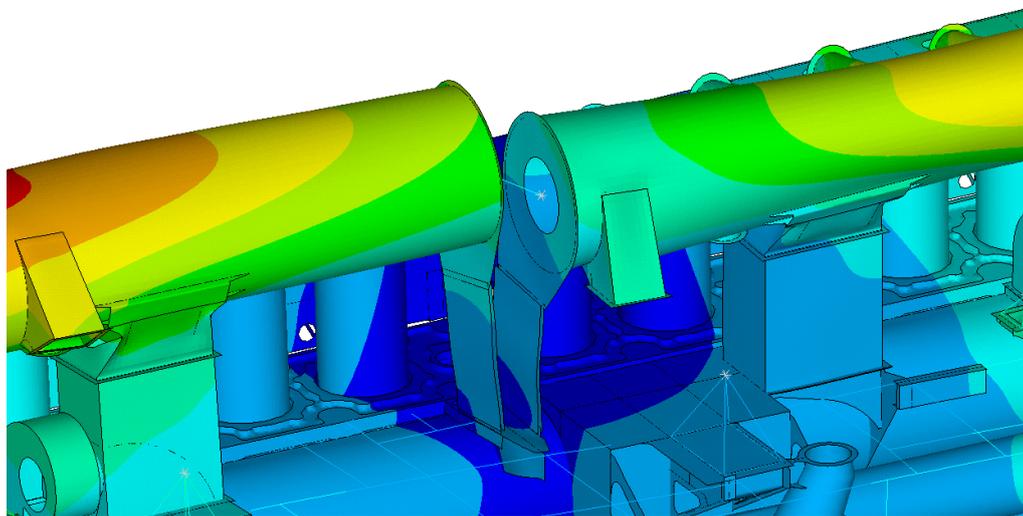


Figure 9: Results of a numerical modal analysis showing the antiphase mode shape of interest after applying stiffeners on the exhaust manifold feet.

In Figure 8 (right) it can be observed that for the mode shape of interest the plates of the exhaust manifold feet buckle. It was assumed that the relative vibration amplitude between both manifolds could be reduced by inhibiting this buckling effect. Therefore special stiffening devices were designed and applied on the manifold feet. A numerical modal analysis demonstrated that the supposition was correct and that stiffeners reduce the feet's buckling effect (Figure 9).

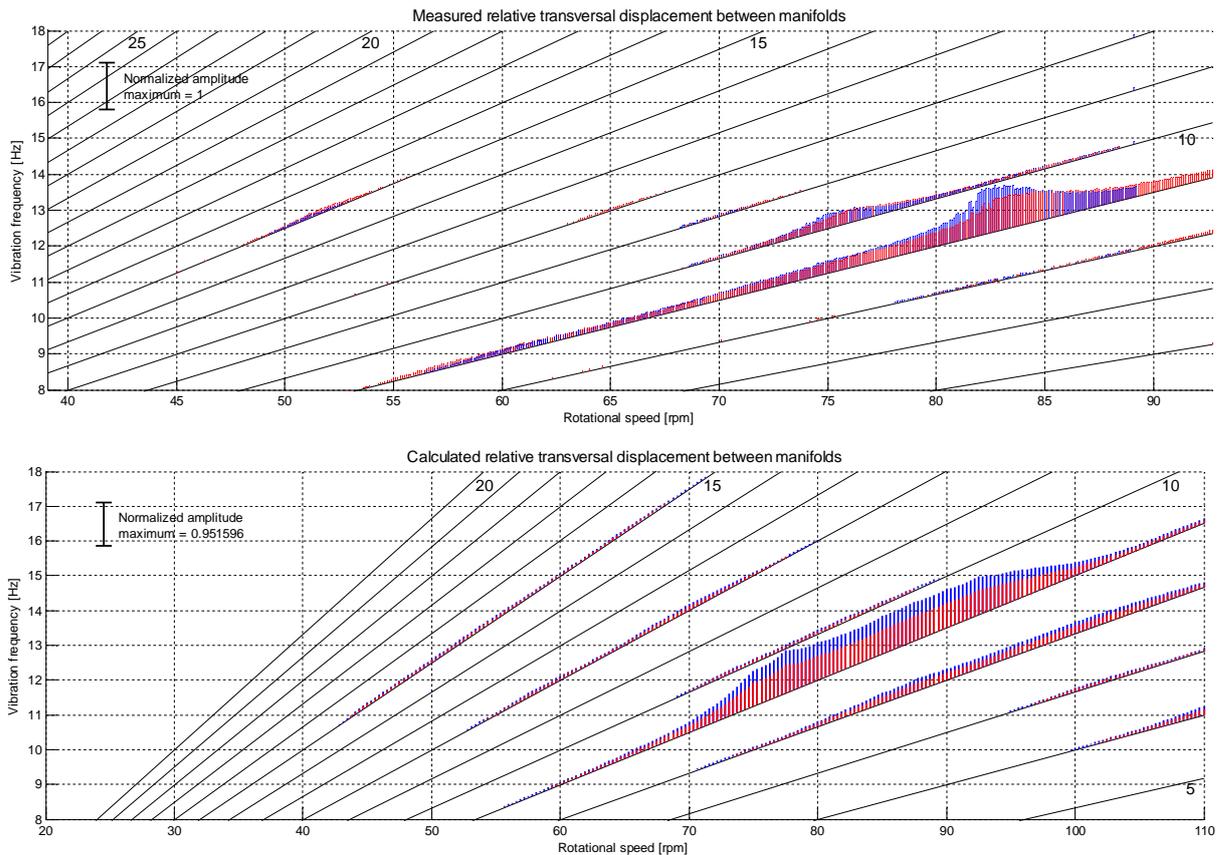


Figure 10: Campbell diagrams showing the measured (top) and the simulated (bottom) relative transversal displacement between both exhaust manifolds. Blue lines represent the original design and red ones the design with stiffening devices. All displacement amplitudes were normalized so that the value 1 corresponds to the largest measured amplitude.

From the Campbell diagrams that were obtained by measurement and by simulation respectively (Figure 10) it can be deduced that the presented calculation method represents the reality very truthfully. The simulation reproduces the measured predominance of the 9th order in the vibration. Also the measured amplitudes are well met. For the design process it is important to know the potential of improvement of the stiffening devices. The decay of the amplitude that was predicted by simulation for the design with stiffening devices is equal to 36%, which is very close to the measured 33%.

These results demonstrate that the applied simulation method is an efficient and useful tool for performing virtual prototyping of large 2-stroke engines and for avoiding extraordinary large costs for testing engines.

4 THE SIMULATION METHOD AS PART OF THE DESIGN PROCESS

The presented simulation method is regularly used in Wärtsilä's development process. After concluding the first design of a new or modified engine a vibration assessment is performed for identifying regions on the structure that are susceptible to vibration.

Figure 11 shows such an engine on which a platform mode was identified by simulation after finishing the first design of the structure. The Campbell diagram demonstrates that this mode could resonate with minor amplitude during engine operation. For achieving the complete customer satisfaction it was decided to perform a slight design modification.

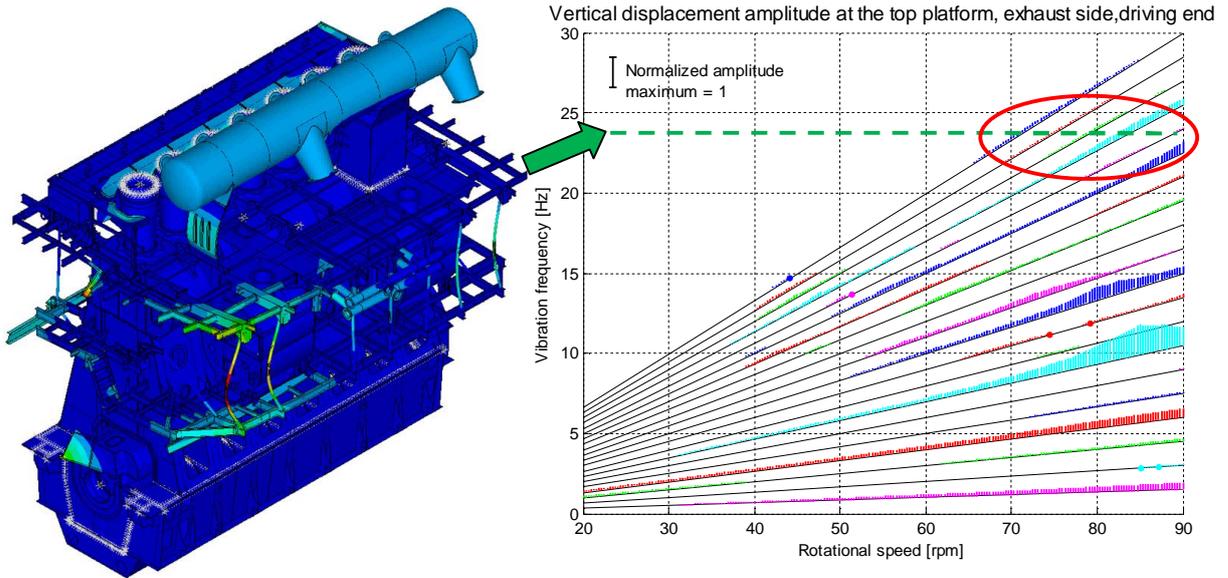


Figure 11: Platform mode shape that was identified on a marine engine after the first design loop. All displacement amplitudes in the Campbell diagram were normalized so that the value 1 corresponds to the largest amplitude.

Figure 12 shows the Campbell diagram before and after the design modification. It can clearly be identified that the mode of interest is not excited anymore after introducing the changes.

This shows how important vibration simulation is for the development of large marine engines. If such calculations did not take place design deficiencies would not be discovered before the engine is delivered to the customer. This would result in large expenses and the loss of the trust of the client.

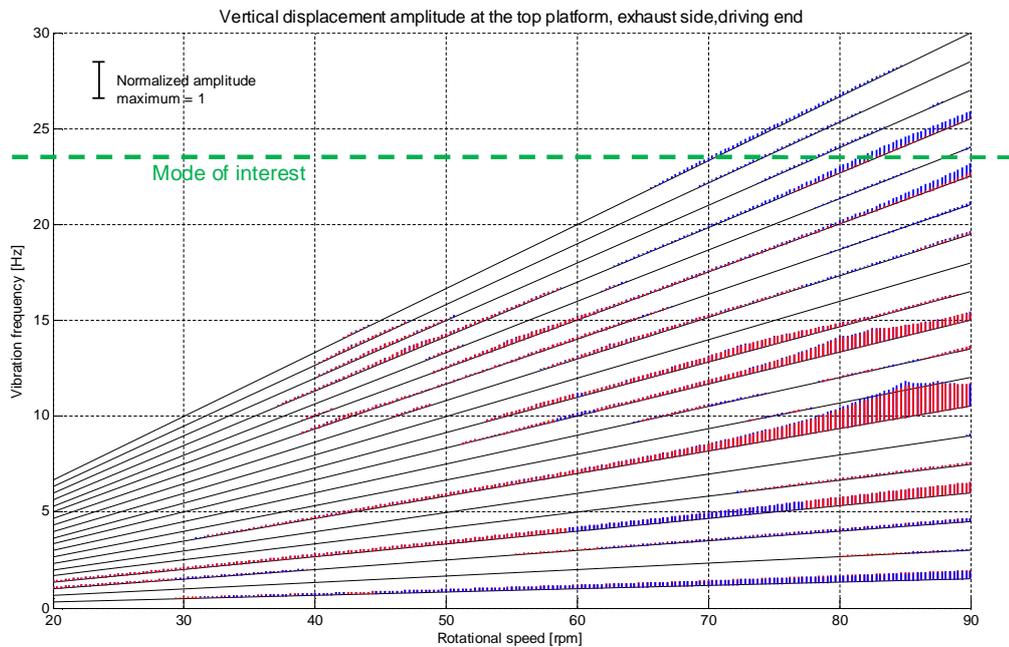


Figure 12: Campbell diagram of the analyzed marine engine before (blue lines) and after introducing a design modification. All displacement amplitudes in the Campbell diagram were normalized so that the value 1 corresponds to the largest amplitude.

5 CONCLUSIONS

- An optimized simulation method for performing an efficient vibration assessment on large 2-stroke marine engines was presented.
- Base of the simulation is a finite element mesh with a minimized number of degrees of freedom for saving computation time.
- A numeric modal analysis and a harmonic response analysis by modal superposition allow to understand the vibration mechanism and to determine which mode shapes are excited during engine operation.
- The modal Assurance Criterion allows an association of measured and numerically computed mode shapes.
- The simulation method was successfully validated with experimental results.
- The virtual prototyping method is a useful and cost efficient tool for supporting the design process.

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