Guidelines to engine dynamics and vibration

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In recent years shipyards and shipowners have increasingly tuned in to the negative consequences of engine vibrations. The pressure on engine manufacturers to improve vibration behaviour has increased, which has also driven the development and use of advanced vibration analysis and simulation methods. The challenge is to keep vibrations on an acceptable level while output power increases and structures become lighter. A piston engine is, by its nature, a vibrating machine. To keep vibrations under control it is essential to understand the basics and the main sources of engine vibrations.

Structural vibration is generally a result of the exciting force and dynamic properties of the structure (Figure 1). The same analogy can be used for any vibrating system.

In modern engine development work all these different components - excitation, structure and vibration response - can be analyzed and taken into account at the design phase itself, as has been done in the case of the Wärtsilä 46F engine, using advanced numerical calculation and simulation techniques. Excitation forces with all possible firing orders can be compared and analyzed, structural properties like the stiffness of the engine can be optimized, and finally a dynamic simulation of the vibration response can be performed.

Mathematically the schematic diagram in Figure 1 is often described with the general equation of motion:

$$M\ddot{x}(t) + C\dot{x}(t) + Kx(t) = f(t) \qquad (1)$$

where M, C and K are matrices of mass, damping and stiffness, $x(t)$ is the vector of displacement (response), and $f(t)$ is the vector of applied force (excitation).

Vibration response

As this article will make clear, there are three important factors contributing to the vibration levels and behaviour of an engine and its components.

1. A rigid body motion of the engine on its flexible mounting
   The engine is moving as a solid part, sometimes causing problems with connections.
2. Deformation of the engine block (global vibrations)
   When in resonance, this causes high stresses that might lead to failures.
3. Local vibrations of a component
   Vibrations of a component with respect to its mounting.

It is not always easy to distinguish the global and local vibrations based on simple overall measurements. The contribution of the flexible mounting is easy to detect, but rigid body modes are of minor importance for most constructions because they cause only small deformations and stresses. Rigid body modes, however, might cause problems with flexible mounts or pipe connections.

When traditional vibration measurements are performed, only the vibration response is measured. This is the quickest way to get an impression of the vibration behaviour of an engine or generating set. But vibration response alone, without knowing anything about the engine's excitation and structural properties, doesn't actually tell us very much about the actual reason for the vibration.

The vibration amplitude is the first thing to give an indication of possible excessive vibration of an engine or component. Normally vibration is measured as the Root Mean Square (RMS) velocity value, which is proportional to the energy content of the vibration. Sometimes displacement or acceleration are measured as well.

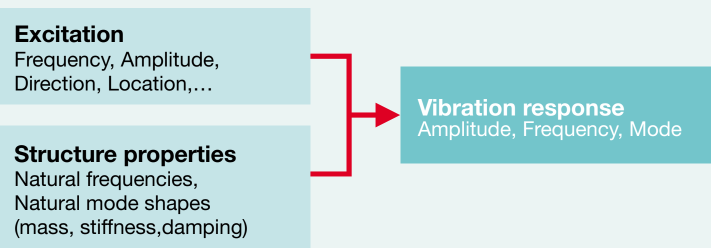


Fig. 1 Schematic diagram of the relationship between excitation, structure and vibration response.

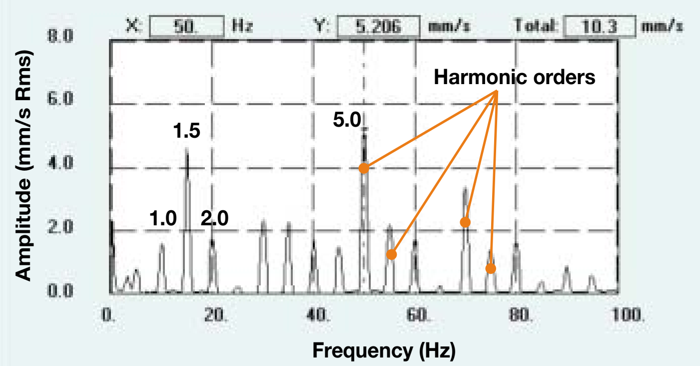


Fig. 2 A typical vibration spectrum measured from a running 4-stroke diesel engine.

To see the frequency content of the vibration curvature a Fast Fourier Transformation (FFT) must be applied to the vibration time signal. The result is called frequency spectrum, and it shows the vibration amplitude as a function of frequency.

FFT is often performed automatically by modern vibration measurement tools. A typical vibration spectrum recorded on a running diesel engine is shown in Figure 2. In this spectrum several peaks, called 'harmonic orders', can be seen. Harmonic order number 1 represents the vibration amplitude of the engine running frequency. In this case (Wärtsilä 6L46F) the running speed is 600 rpm; thus the first order is 10 Hz. The other harmonics are multiples of this frequency, i.e. orders 0.5, 1.5, 2.0, 2.5,... In a 4-stroke engine also half orders are present unlike in 2-stroke engines.

Analyzing vibrations this way in frequency domain, it is much easier for an analyst to determine the origins of the vibration.

When it is necessary to go even further in the analysis, vibration spectra can be measured from several locations in the structure and they can be combined with special software to build up a vibration mode shape. The vibration mode shapes that appear during engine operation are called Operational Deflection Shapes (ODS). Animated mode shapes are a powerful tool normally used in problem solving to localize weak points or identify failure mechanisms.

Dynamic properties

The dynamic properties of a structure essentially mean its natural frequencies and mode shapes, which depend on the mass and stiffness properties. Mathematically this means the solutions of the eigenvalue problem of an undampened system:

$$[M^{-1}K]\psi = \lambda\psi \tag{2}$$

where λ is a diagonal matrix of eigenvalues and ψ is a matrix of eigenvectors. Together with damping they define how the structure behaves under a certain dynamic loading.

A natural frequency is the frequency at which the structure 'likes' to vibrate. There it can be excited to vibrate very easily - even with a very small force. And when it is excited, it starts to vibrate in its natural mode shape, such as a bending or twisting mode. The structure is said to be 'in resonance' when a dynamic excitation force happens to influence at the same frequency as the structure's natural frequency. Most vibration problems are due to resonance phenomena and most of today's vibration control is about avoiding resonances.

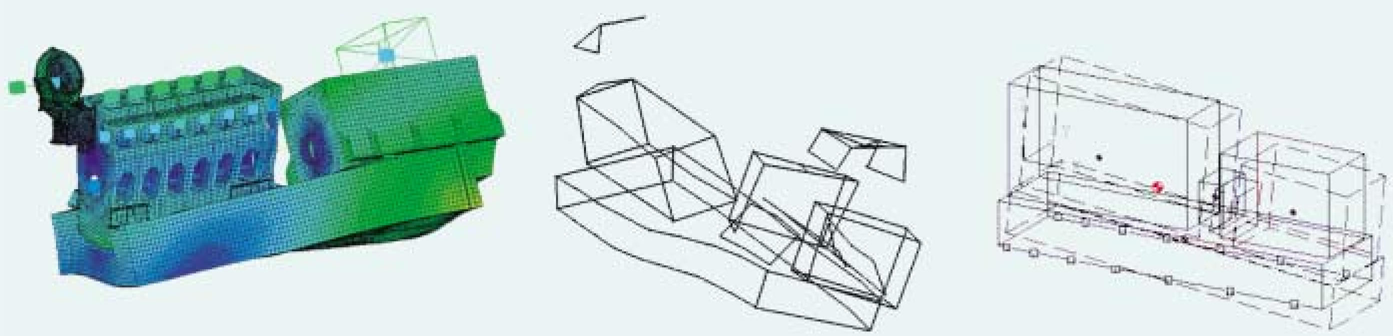
The dynamic properties can be determined by using the Finite Element Method (FEM) when the above mentioned eigenvalue problem is solved numerically, or by means of experimental modal analysis where a known excitation force - e.g. a hammer blow - is given to the structure and the response is measured. This way it is possible to see how a big force is needed to get the structure to vibrate at a certain level. The measurement gives the natural frequencies, often called resonance frequencies, as well as the damping properties of the measured structure, and if the measurement is performed at several locations, the natural mode shapes can be composed.

An engine or generating set with its flexible mounting is a good example of this kind of vibration system. In the dynamic analysis of such a system the engine itself is normally treated as a rigid body, i.e. a mass point with mass moments of inertia. The flexible elements are treated as springs where the stiffness and damping are located. Altogether six eigenmodes of rigid body motion can then be calculated, three translational and three rotational modes. Since the mathematical models are very small in these cases, the FE method is not needed but special dedicated software is used. The FE method is normally used for larger problems like calculating natural frequencies and elastic mode shapes of whole engines and generating sets. In these calculation models the mass and stiffness are uniformly distributed over the whole structure, but the same equations apply.

In addition to these so-called global natural frequencies, where the whole engine or generating set is taking part in the vibration mode, sometimes the reason for a component vibration can be found in a local resonance. In such cases a local part of the construction, e.g. a plate or a pipe, may vibrate heavily at a frequency where the engine itself is not vibrating too much. In some cases when the vibrating part, like a turbocharger, is relatively heavy compared to the engine, its resonance can have a considerable influence on the vibration behaviour of the whole system. Normally these problems can be handled by stiffening the local part with an additional support or by adding extra mass to it. The aim of both of these methods is to prevent resonance by moving the natural frequency of the component away from the harmful excitation frequency. Adding stiffness moves the natural frequency higher, and adding mass moves it lower.

Excitation forces

In a combustion engine the dynamic forces are created by various things like the crank mechanism, pressure pulses from gas, fuel or air flow, valves, gearwheels, an unbalanced turbocharger, etc. Practically all these forces are also 'periodic', i.e. they contain harmonic components, so the frequency domain analysis is normally well applicable. However, since gear hammering is a non-linear phenomenon, it is necessary to simulate the camshaft and gear vibrations using direct integration in a time domain.



Torsion mode calculated using the Finite Element Method.

Torsion mode measured using Experimental Modal Analysis.

Rigid body pitching mode calculated with special software.

Fig. 3 Some global natural mode shapes of a Wärtsilä 20 generating set.

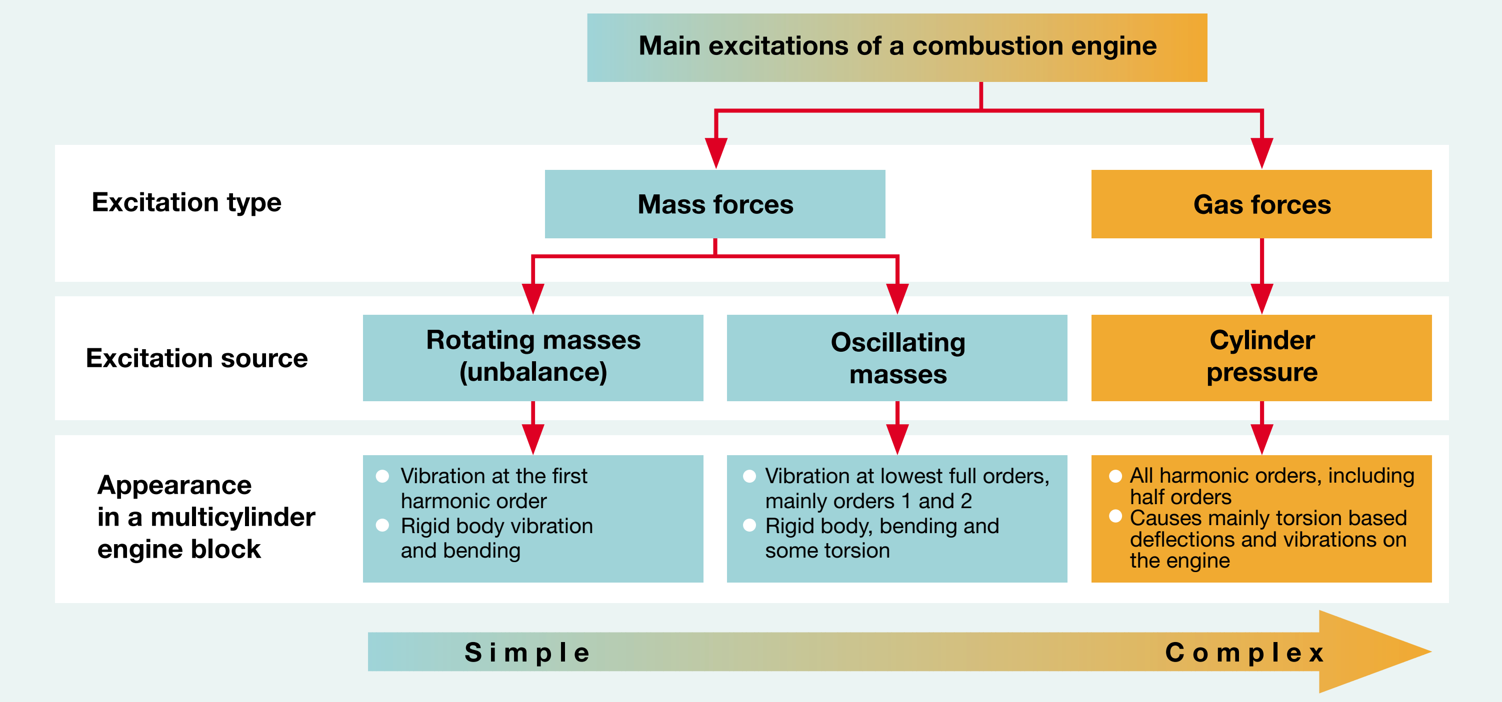


Fig. 4 Categories of main dynamic excitations.

The moving components in the crank mechanism, together with the cylinder pressure, are normally referred to as the main dynamic excitation sources in a combustion engine. These excitations and their appearance as vibration can be categorized in the way shown in Figure 4.

Roughly speaking the following relationships between the lowest harmonic orders, excitations and vibration mode shapes of elastically mounted engines can be defined:

Harmonic order 0.5	- Gas force unbalance	- Rigid body motion
Harmonic order 1.0	- Mass force	- Rigid body motion and bending
Harmonic order 2.0	- Mass force	- Bending
Half orders 1.5, 2.5, ...	- Gas force	- Torsion

However, if a local resonance frequency, e.g. a natural frequency of the turbocharger, happens to be close to one of these harmonic orders, the mode shape at that order is dominated by the vibration of this component.

The usual way to reduce the negative effects of mass forces is balancing. Counterweights are used primarily to balance single cranks internally to reduce local bending of the crankshaft and to minimize bearing loads. However, they also have an influence on the global bending deflections of the entire engine block.

The degree of balancing (DOB) tells us how big a part of the rotating masses are balanced by counterweights. Normally DOB has been around 75...95% in Wärtsilä engines but nowadays the tendency is to use higher balancing. In the Wärtsilä 46F engine, since the higher running speed causes increased mass forces, all the rotating masses have been balanced with 100% DOB. A special case is the Wärtsilä 12V46 where the DOB has been increased above 100% to achieve a minimized and optimum bending deflection of the engine block in both vertical and horizontal planes.

In engines having an even number of cranks, like 6L, 8L, 12V or 16V, the cranks are arranged symmetrically so that the moving masses balance each other out. These engines are said to be externally balanced; they have no so-called free forces or ‘couples’, i.e. excitations causing rigid body movements. However, in real life the moving components have always some weight variations due to manufacturing tolerances which cause free loadings. On resiliently mounted engines these forces and couples can be strong enough to excite the engine to vibrate when the running speed passes a resonance frequency of the elastic mounting system.

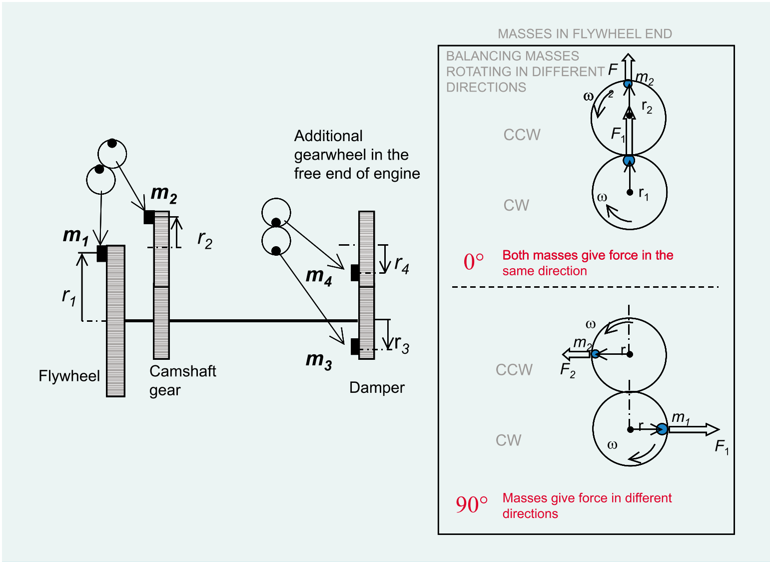


Fig. 5 Working principle of the balancing device used in the Wärtsilä 9L46.

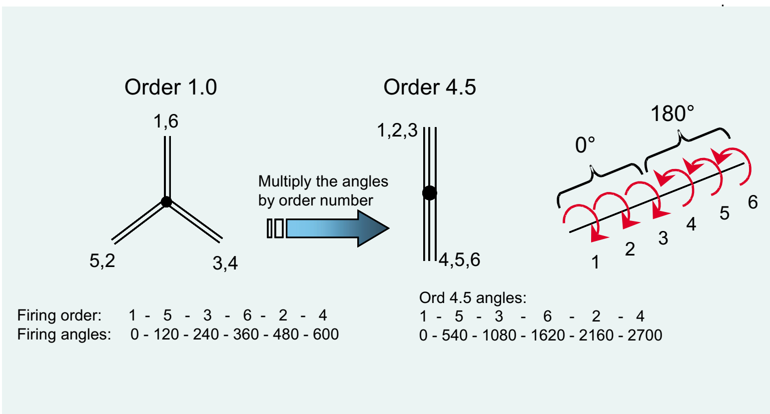


Fig. 6 Example of composing the torsional excitation mode shape of order 4.5.

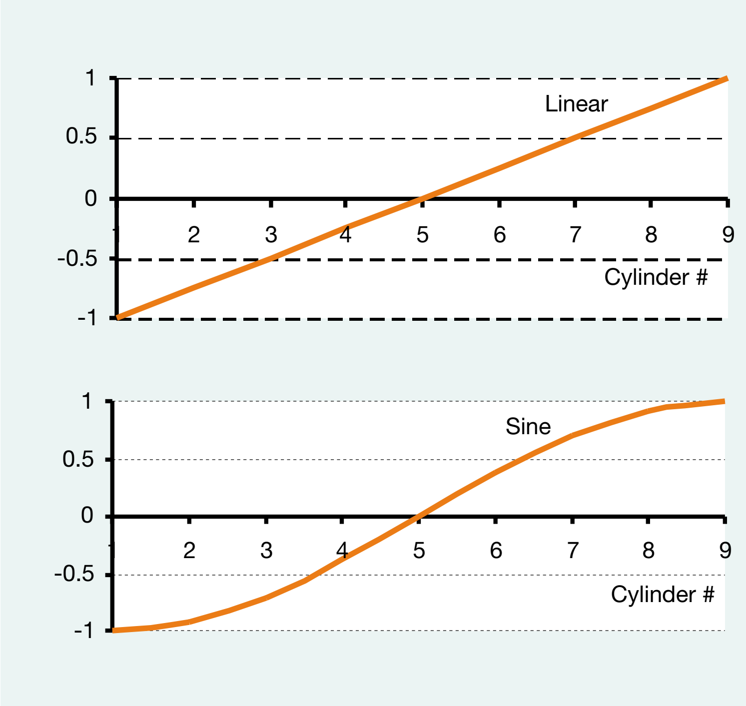


Fig. 7 Examples of different weighting functions used in the calculation of vector sums.

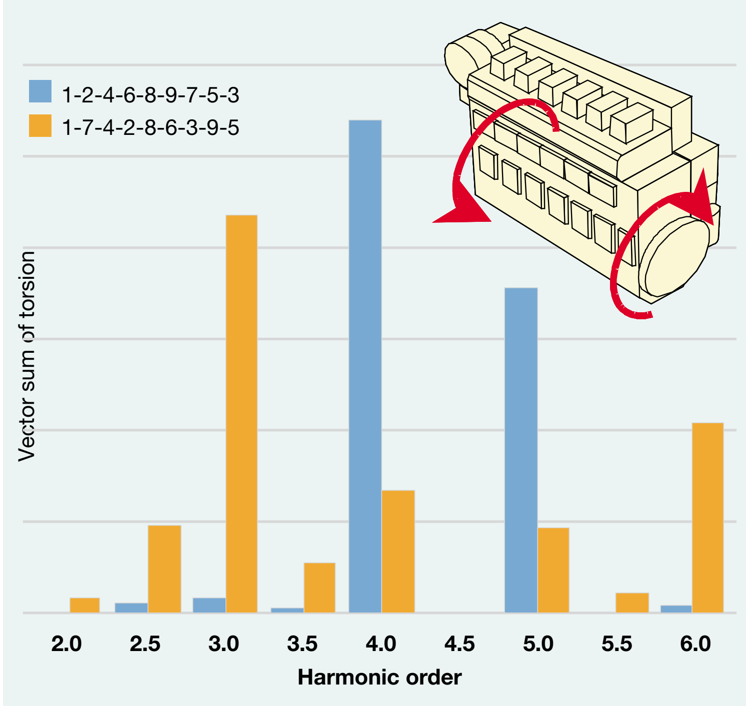


Fig. 8 Torsional reaction forces with two different firing orders on a 9-cylinder engine.

Resonance situations are often unavoidable, as in the case of variable speed main engines, and the vibration system must be dimensioned accordingly to cope with the resonances, e.g. by reducing the excitation forces sufficiently. The forces due to component weight variations are minimized by optimizing the locations of the components rotating with the crankshaft. When the optimization is done properly these forces and couples can normally be reduced by some 80-90% from the original level.

In engines with any odd number of cranks, like 7-, 9- and 18-cylinder engines, it is not possible to arrange the cranks symmetrically, which results in free couples. Special balancing holes in the flywheel and additional weights in the free end of the crankshaft make it possible to tune the ratio between the couples in vertical and horizontal planes.

However, an additional balancing device recently introduced in the Wärtsilä 9L46 engine, and with future implementation in 9-cylinder Wärtsilä 38 and 46F engines as well, makes it possible to eliminate all the first-order free couples altogether. This is done by placing an additional balancing mass on the counterclockwise rotating intermediate gearwheel of the camshaft gear, as well as an additional counterclockwise rotating wheel with an unbalanced mass at the free end of the engine. These, together with the clockwise rotating balancing masses, make it possible to reach zero couples. The working principle of the balancing device is presented in Figure 5.

Another way to balance free couples is to use non-equidistant angles between different cranks. This kind of balancing has been tested in a 9-cylinder Wärtsilä engine with promising results. From the vibration point of view, the small disadvantage of this crankshaft type is the fact that firing irregularities cause small rolling couples at low frequencies.

Sometimes it is possible to live with a resonance if the excitation direction does not match the natural mode shape, e.g. a bending excitation at a natural frequency of torsion. The worst possible resonance situation is congruity of the natural mode shape and the excitation mode shape at the same frequency. By comparing the excitation and natural mode shapes it is possible to form an impression of whether the excitation at a particular harmonic order is dangerous for the structure or not. The torsional excitation modes are easily composed, as shown in Figure 6.

The modes of torsional and bending excitations are often analyzed by means of vector summation, an easy method of comparing the excitation mode shapes and natural mode shapes. The vector sum is the sum of the force or torque vectors at the specified harmonic orders, and is calculated using the following formula:

$$V = \sqrt{\left(\sum_{i=1}^{c} f(i) A \sin \varphi_i\right)^2 + \left(\sum_{i=1}^{c} f(i) A \cos \varphi_i\right)^2} \qquad (3)$$

where c is the number of cylinders and φ_i is the firing angle of cylinder i multiplied by the harmonic order in question: φ_i = order$\cdot\alpha_i$. A is the force or torque amplitude and $f(i)$ is a function defining a weighting factor depending on the estimated mode shape. When analyzing torsional excitations, a linear weighting function or a sinus wave can be used, both of which are shown in Figure 7. It is also possible to adjust the location of the nodal point of the mode if the torsional mode shape is well known.

The excitation modes at different harmonic orders depend strongly on the firing order in use. Changing the firing order may significantly influence the vibration behaviour of an engine or genset. As an example, Figure 8 shows the difference in torsional reaction forces for the first torsion mode of two different firing orders in a 9-cylinder engine.

At the firing frequency, order 4.5, the excitation mode is rolling and thus gives a zero value for the first torsion mode if the nodal point is located in the middle of the engine. In modern engine development work, as in the case of the Wärtsilä 46F, the firing order is used at an early stage as one parameter in engine dynamics optimization. All possible firing orders can be quickly analyzed using special software and the optimum one chosen.

Vibration simulation

In order to perform a numerical vibration simulation of an engine, it is necessary to know the main excitation forces and the dynamic properties of the engine. The market offers a variety of software for solving this task. The analysis can be performed in a time domain or in a frequency domain. The method of direct integration in a time domain is a really heavy and

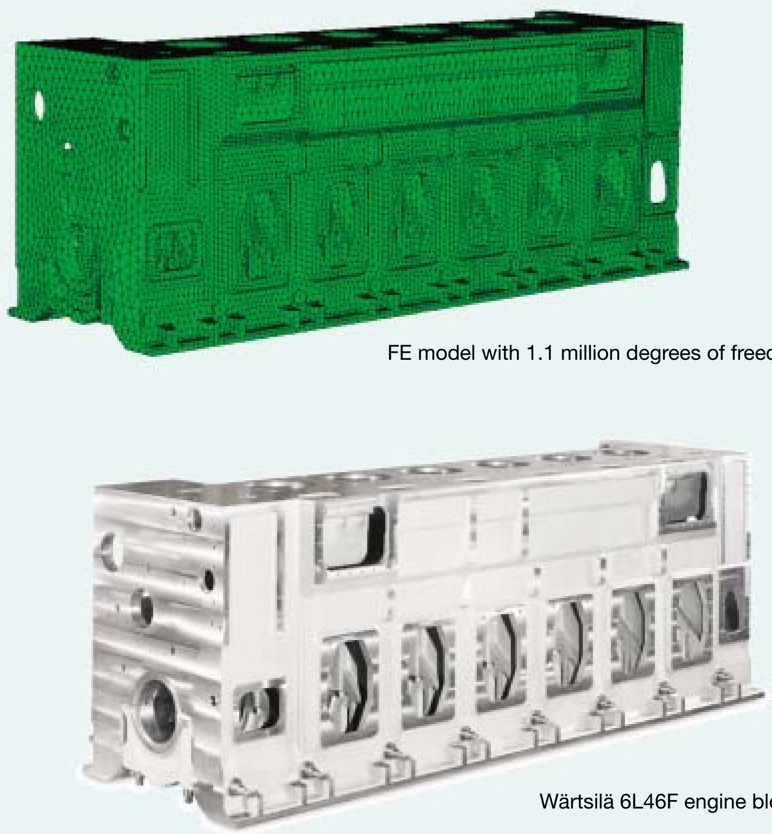


Fig. 9 The Finite Element model of the Wärtsilä 6L46F engine block.

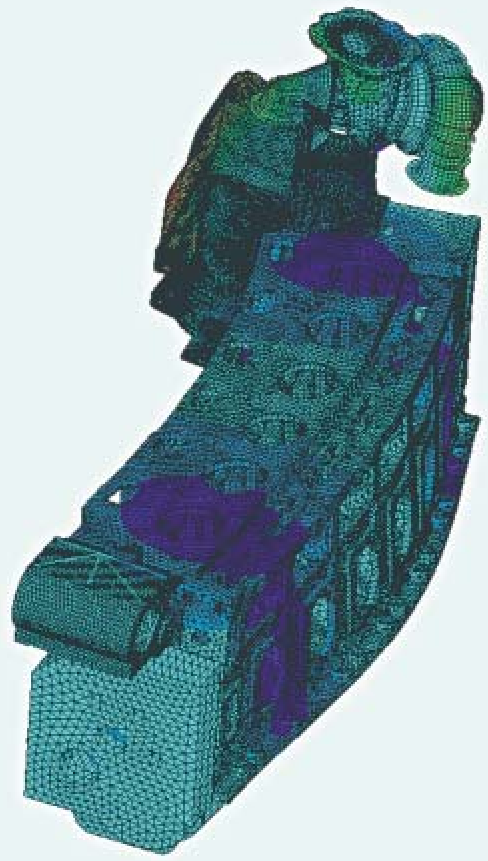


Fig. 10 Natural mode shape of horizontal bending of the Wärtsilä 6L46F engine. The FE model consists of about 3.5 MDOF.

time-consuming calculation, but it offers the possibility to include non-linearities in the model. It is much faster to perform the calculation in a frequency domain using methods based on superpositioning of eigenmodes, and bigger models can be used. This method also offers the possibility to quickly test different variations of the structure and excitations as well as to simulate an engine running at variable speed.

Building up an FE model intended to be used in dynamics simulation is a demanding job. To get reliable results from the response analysis the model must be highly accurate. The FE model of the Wärtsilä 6L46F engine was built up in several steps according to the order of engine assembly. First only the engine block consisting of about 1.1 MDOF (million degrees of freedom) was modelled (Figure 9). The remaining components (main bearing caps, oil sump, pump cover, crankshaft and camshaft, cylinder heads, turbocharger unit, etc.) were then added one by one or in groups. At each step an experimental modal analysis was performed and the FE model was verified with measurement results. The final FE model consisted of about 3.5 MDOF, and the model was verified using an extensive modal measurement where the structure was excited by a hydraulic shaker. Figure 10 presents the horizontal bending mode of the whole engine.

In the dynamic response simulation method adopted by Wärtsilä the dynamic forces at each harmonic order are calculated separately with special software. The forces are saved in a file which is then imported into the FE model and applied to the correct nodes located in the main bearings and cylinder liners. The calculation procedure is presented in the diagram in Figure 11.

The whole procedure is highly automated to avoid manual work since the number of generated complex-valued force vectors is large. This method allows us to simulate the dynamics of engines running at variable speed according to the propeller law or with constant torque. A typical calculated vibration response curve is shown in Figure 12.

Acceptable vibration levels

The vibration levels of engines are normally judged according to the following standards:

Reciprocating engines:	*ISO 10816-1 / ISO 10816-6*
Generating sets:	*ISO 8528-9 and ISO 10816-3*

In addition to these standards, different classification societies give their own guidelines on how to assess vibration levels in different applications. However, engine manufacturers are often asked nowadays to give their own opinions on the vibration levels of their engines and built-on components. For this reason Wärtsilä has been systematically collecting statistical information on normal and abnormal vibration levels in order to build up its own rules. Also manufacturers of components, e.g. turbochargers and alternators, sometimes give their own vibration limits, which should be respected.

For years Wärtsilä's general rule regarding vibrations has been that the overall level measured on the engine block in units of RMS velocity should not exceed 28 mm/s. For common baseframes, built-on components and pipelines the limits are given separately.

As the overall vibration level at any single measurement point is a sum of different kinds of structural behaviour like rigid body movement, elastic deflections and local behaviour, it is not always easy to give exact limit values. Engines with different numbers of cylinders behave differently, and it is only natural that some engines have higher vibration levels on average.

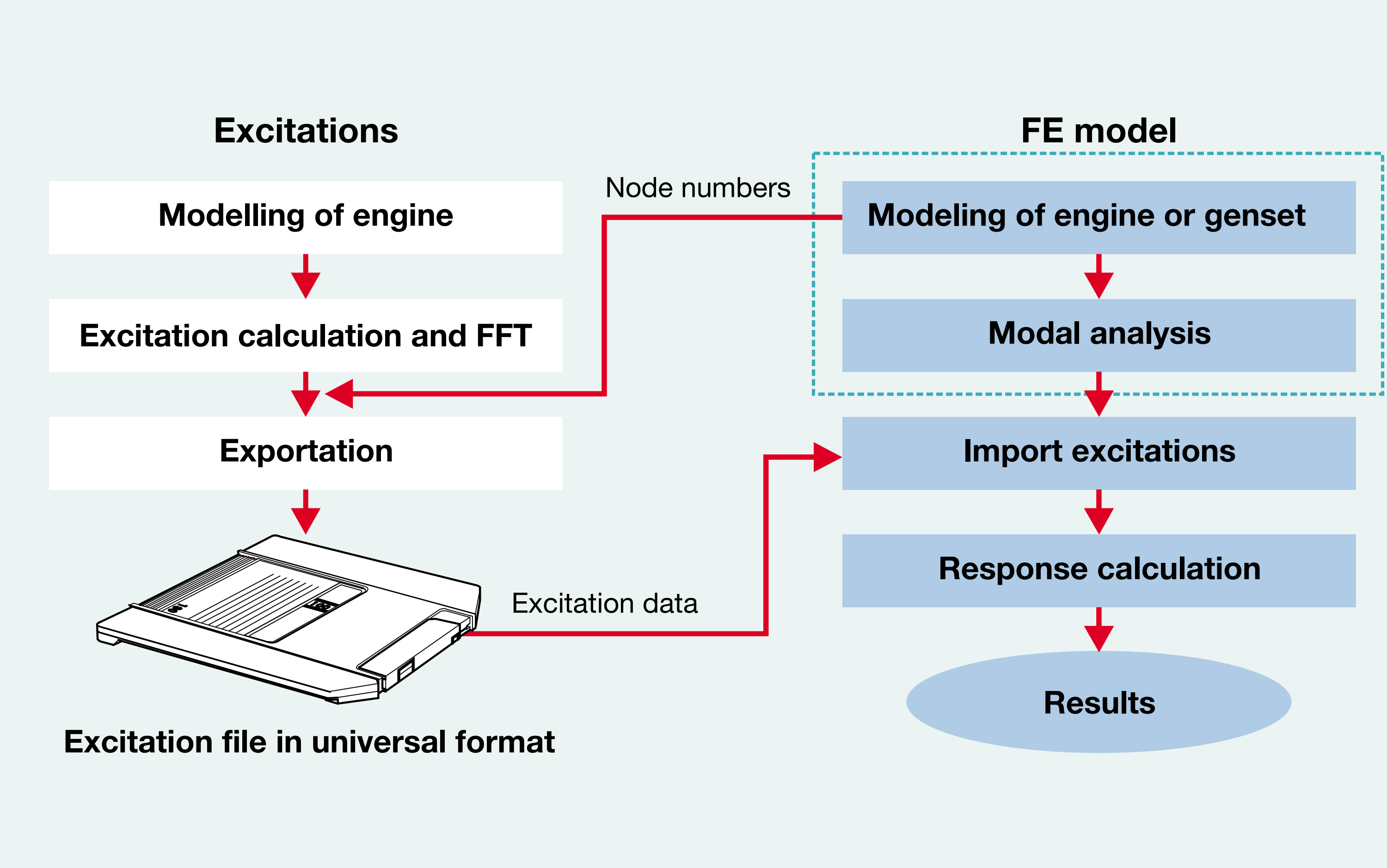


Fig. 11 Diagram of the dynamic response analysis procedure.

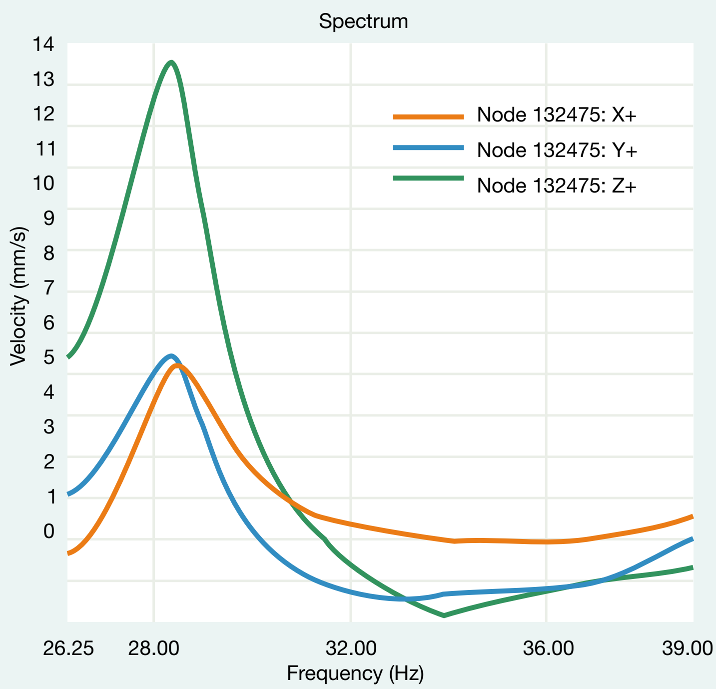


Fig. 12 A typical vibration response curve calculated using excitations of propeller loading within an engine running speed range of 350...530 rpm.

It may happen that a vibration level is not acceptable although the recorded levels are lower than the given limits. This may occur when the overall level is fully dominated by a single frequency component, which may be an indication of a resonance situation. To recognize this it is necessary to perform a spectral analysis as described earlier. This then allows the maximum value of the highest single component to be compared with the overall level or the given limit. A good rule of thumb is that the contribution of any single frequency component should not be more than 80% of the acceptable overall level.

Summary

The structural vibration response level of an engine is a combination of the structural properties, i.e. the natural frequencies, mode shapes and damping, as well as the dynamic excitation forces such as those coming from gas pressure in the cylinders and moving masses in the crank mechanism. The use of advanced analysis tools makes it possible to determine the vibration system, either numerically or experimentally, and to make a relatively accurate prediction of the engine's vibration behaviour.

Vibration optimization normally consists of a structural analysis and an excitation analysis as well as relevant modifications by which the resonance situations, i.e. congruity of the natural and excitation frequencies and mode shapes, are avoided as far as possible. Sometimes resonances are not avoidable and the excitations must be minimized or modified accordingly. Wärtsilä has adopted powerful numerical methods in order to better support engine development and to cope with increasing demands regarding vibrations. ■