



Structural response of the ship hull elements subject to excitation generated by the main engine

Andrey Smolko

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Supervisor: Dr. Maciej Taczala, West Pomeranian University of
Technology, Szczecin

Reviewer: Dr. Lionel Gentaz, Ecole Centrale de Nantes

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ABSTRACT

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By Andrey Smolko

The central subject of the investigation in this paper is real existent bulk carrier m/s Miedwie (DWT 30000). During exploitation in certain operation condition excessive vibration levels were observed in the engine room and reported by the crew. Moreover, structural failures occurred in the form of extended cracks along foundation of hull supporting elements connected to the engine. Due to all these facts vibration of the main engine and the hull supporting structure in the engine room has been studied in detail.

The bulk carrier m/s Miedwie is equipped with the main engine of 6RTA48T-B type produced by Wartsila Company. It is a low-speed direct-reversible two-stroke engine with 6 cylinders. In the configuration there are also two lateral side stays of friction type installed on exhaust side of the engine. Their role is to reduce the engine vibration and the vibration transmission to the ship's bottom and side structure.

Numerical analysis has been performed to find out the reasons of the excessive vibration problem. For this purpose a very accurate finite element model of the aft part of the ship including engine room was created. The procedure of model generating was divided into two parts. First step is geometrical and initial FEM modeling of the ship structure in Poseidon GL software. The second stage is mesh modification, adding of the propulsion system and other debugging of the numerical model in ANSYS software. The main engine has been modeled with maximum concern about its stiffness and mass distribution. Interaction of the intermediate shaft is expected to be important, thus the whole simplified shaft line is represented in the model. The superstructure has been incorporated in the model in approximate way to represent only mass inertia interaction.

Forced harmonic vibration analyses have been performed in ANSYS APDL software. Studying of the forces induced in this type of engine was done. Measurements carried out on board the ship showed vibration of the engine with the 6-th order frequency dominating component. Therefore it has been concluded that the mode of the occurred lateral engine vibration (rocking) was of so-called "H-type". This type of excitation is caused by lateral guide forces and the value of that forces are known from Marine Installation Manual for engine.

Firstly numerical analysis without influence of lateral stays was performed. The natural frequency of “H-type” vibration and corresponding amplitude were found. The influence of the double bottom structure stiffness on the engine natural frequency was determined. Comparison between service engine speed and engine speed when resonance effect occurred was done.

Second analysis was dedicated to modeling of forced vibration with presence of the active lateral friction type stays. Due to high friction side stays were not installed properly, making joint between the engine block and the hull structure almost stiff. Resonance frequency and amplitude were also found and the comparison with service speed was performed. The formation of local stress concentration areas, which is able to cause the fatigue crack in short time, was observed.

After all numerical simulations conclusions about influence of the hull supporting elements stiffness were made. Importance of the correct friction type stays installation was shown. Several recommendations about avoidance of the dangerous resonance effects during exploitation period were given.

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1. INTRODUCTION

Because low-cost building and operation aspects of a ship increasingly influence the design, vibration problems occur more frequently. The following design trends contributed to this:

- Light weight construction and, therefore, low values of stiffness and mass (low impedance).
- Arrangement of living and working quarters in the vicinity of the propeller and main engine to optimize stowage space or to achieve the largest possible deck openings of container ships and bulk carrier.
- High propulsion power to achieve high service speed.
- Small tip clearance of the propeller to increase efficiency by having a large propeller diameter.
- Use of fuel-efficient *slow-running engine*.

Studying of the structural response of the ship hull elements which is excited by a main engine is the aim of this thesis research.

Inevitable a diesel engine produce several different loads on a ship structure with different frequency orders. Therefore, the full classification of all loads and moments created by a diesel engine is revealed in Part 3.

There are huge amount of possible ship constructions and possible scenarios of engine – hull vibration interaction. Part 2 shows examples of global hull girder vibrations, local vibrations and vibration of some substructures which can be caused by engine forces under certain conditions. For this reason the real existent ship has been chosen that, using this example, show the ways of the problem analysis and solution.

Genfer Design Company has provided information and all necessary materials about vibration problem which occurred on the bulk carrier m/s “Miedwie”.

This ship was built in 2010 in China and nowadays is operated by POLSTEAM Company (Poland) (see Fig. 1.1).



Figure 1.1 m/s Miedwie

Main dimensions are:

Length overall – 190 meters;

Length between perpendiculars – 182.6 meters;

Breadth moulded – 23.60 meters;

Depth moulded – 14.60 meters;

Freeboard draught – 10.10 meters.

Class:

D.N.V. class: +1A1 ICE-1C Bulk Carrier CSR ESP BC-A Holds 2, 4, 6 or 4 may be empty, GRAB(20), ES(D) EO NAUT-OC BWM-E(s, f) TMON BIS.

General arrangements are presented in Appendix 1.

During exploitation the high engine vibration levels reported by crew and the structural cumulative cracks observed (see Fig. 1.2) occurred when the engine was running with reduced speed (80-95 rpm). Engine was equipped with friction type side stays what is standard for Wartsila engines. Preliminary investigation has stated that engine had a lateral so-called “H-type” vibration.



Figure 1.2 AFT side stay – crack at the bottom weld of the main support element.

In this thesis research an attempt to investigate reasons of the failures and high level vibration inside the engine room of m/s Miedwie has been performed. For that a modal and forced harmonical FE simulation in ANSYS APDL has been done.

First step in all numerical simulation is a preparation of the accurate model. Detailed information how FE model was created, what assumptions were made and what elements were used is presented in Part 5.

Results of all modal analysis which were accomplished in that research are given in Part 6. Natural frequencies of H-type vibrations were found (engine without stays). Influence of the hull presence on the natural frequencies of engine H-type vibrations was studied. Also importance of engine stiffness characteristics was analysed. The role of the superstructure presents in the vibration process was observed.

Part 7 is devoted to the description of the results of forced vibration simulation. Two variants of the structure were investigated. First one is the engine system without stays and

the second one is the same structure but included side stays. Friction type side stays are considered as solid structure (too tight regulation). Amplitude-frequency curves for both cases were obtained. Stress concentration areas were found in the stays support structure. Velocity amplitudes of the engine top were extracted and compared with acceptable level. Derived results are in good coincidence with results obtained in the company and experiments.

Part 8 is a conclusion where some recommendations how to decrease level of vibration were suggested.

2. OVERVIEW OF NATURAL VIBRATION CALCULATIONS

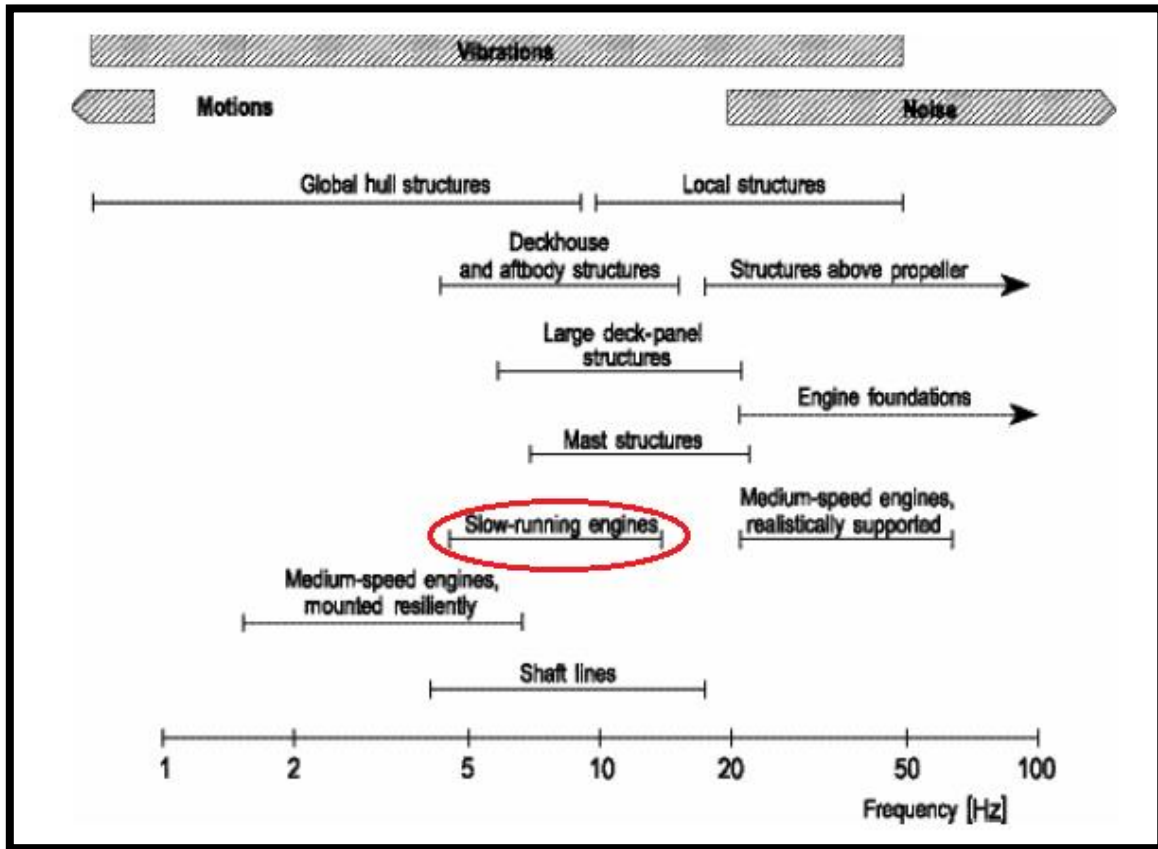


Figure 2.3 Natural frequency ranges in shipbuilding applications

In Fig.2.1 the vibration phenomena relevant in shipbuilding applications are plotted versus frequency. The frequency limits indicated are valid for standard designs and for normal ship types.

The transitions between ship motions, ship vibrations and ship acoustics are smooth. In the field of vibration, it is possible to distinguish between three different phenomena: global hull vibration, vibrations of substructures and local vibrations.

In general, the higher the frequency, the greater the modal density, i.e., “the number of natural frequencies per Hertz”. As a result, the system response in the higher frequency range is defined by the interaction of more natural modes than at low frequencies. In the transition to structure-boom noise, the mode density finally becomes so large that a frequency-selective analysis of the structures dynamic behavior requires an unacceptably large effort. One then has to make do with characteristic energy values averaged over frequency intervals (Statistical Energy Analysis, Noise-FEM, etc.). Today of course, FEM is used to some extent in this

frequency range too. However, with the current available power of computers, frequency-selective computation is limited to partial areas of particular interest, such as engine foundations. For example, an FE model intended for reliable computation of natural frequency of about 200 Hz has about the same number of degrees of freedom as a complete hull model used to compute the natural vibration up to 20 Hz.

Fig.2.2 (beam vibration example) explains the reason of using bigger number of nodes for modeling of high frequency modes.

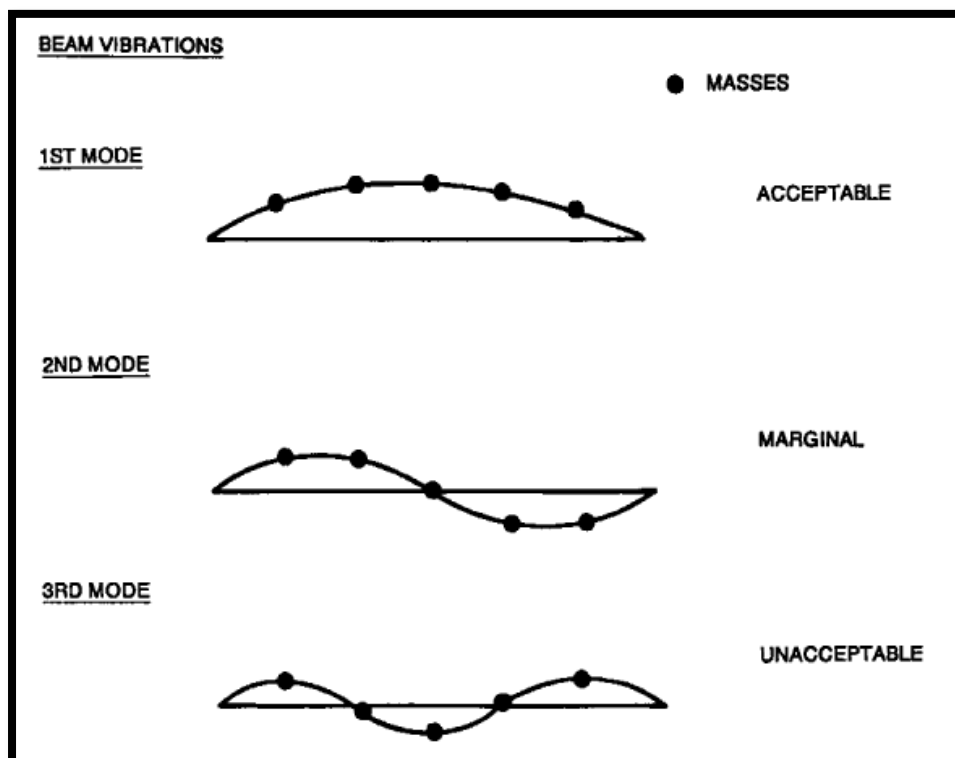


Figure 4.2 Nodes required for accurate determination of natural frequencies

Therefore for higher modes a more detailed representation of nodes is required because the mode shape is more complex.

As the object of research in this thesis is main diesel engine, it is possible to see on Fig.1 that frequency range of interest lies between 3-18 Hz. Such values can be classified as low-frequencies. That aspect allows us avoid heavy FE model of the structure with rather moderate number of nodes and elements.

Below all three main phenomena: global vibration, local vibration and vibration of substructures will be described. Some important aspects of their calculation will be presented

briefly. Understanding the vibration processes on all levels is crucial point as they are strongly connected and very often occur simultaneously.

2.1 Global structures

Global vibrations in this context are vibrations of the ship's entire hull in the frequency range from about 0.5 to 10 Hz. Typical large substructures, such as the aft part of the ship, the deckhouse and the double bottom, are coupled in a way that cannot be considered isolated. Thanks to advances in computer technology, computation methods for determining global vibrations progressed rapidly during the past two decades. From today's point of view, classical approximation formulas or simple beam models for determining natural bending frequencies of a ship's hull are in many cases no longer adequate. For container ships with a high deck-opening ration e.g., for which coupled horizontal and torsional vibration modes play an important part, they do not offer the necessary degree of accuracy. In the past, one had to make do with beam models of a more complex type to cover shear and torsional stiffnesses of the ship's hull. However, in the meantime FE analyses using 3D models of the hull became the standard computational tool.

2.1.1 Modeling

The representation of a ship's structure in an FE model is generally the most laborious step of the analysis. For global vibration, it turns out to be sufficient to represent primary structural components with the aid of plane stress elements. Bending stiffnesses of the deck and wall girders are not covered by this type of modeling, since they are generally simulated by truss (beam-type element) elements. Large web frames are taken into account by plane stress elements as well. For sake of simplicity minor structural components lying outside the planes of the modeled sections are considered as additional element thickness or are ignored altogether. The division of the model is oriented relative to deck planes and to main longitudinal and transverse structures. The numbers of degrees of freedom is 20 to 40 thousand, yielding 50 to 150 natural vibration modes in the range up to 20 Hz. Three typical models are shown in Fig.2.3, namely, a 700 TEU container ship, a smaller double-hull tanker. Where TEU is twenty-foot equivalent unit (based on the volume of a 20-foot-long container).

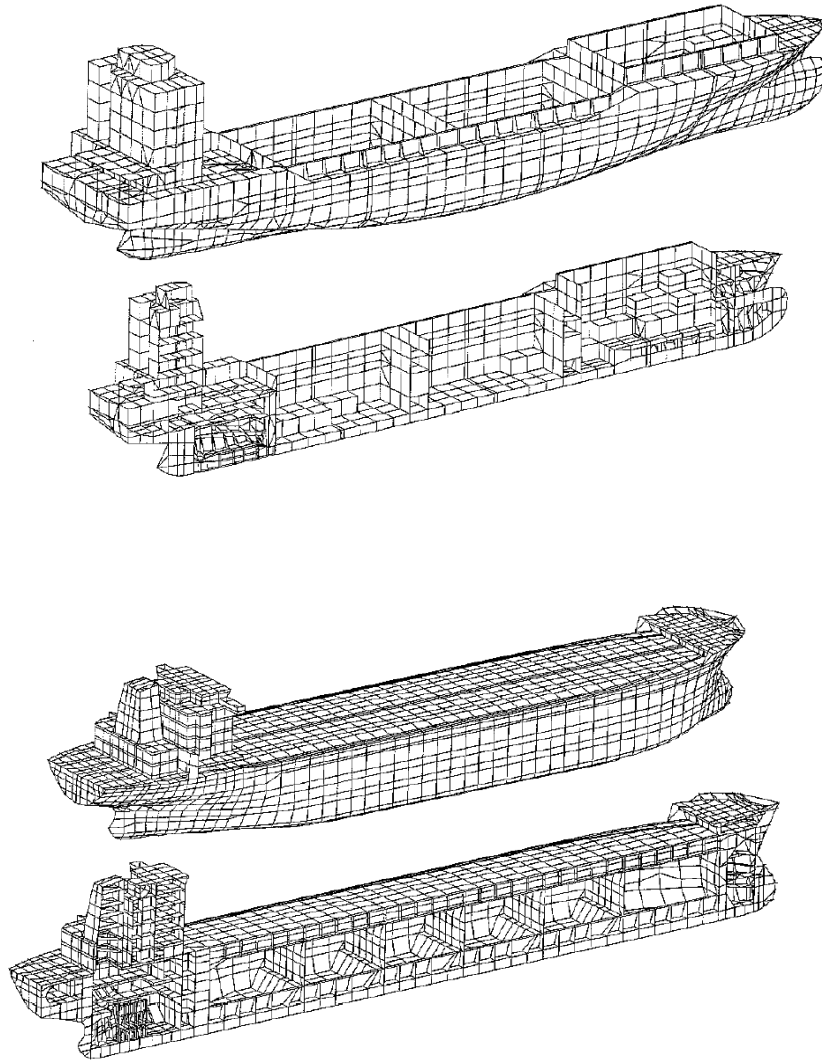


Figure 2.5 FE models of various types of ships

In global vibration analyses, it is not necessary to model the middle and the forward part of the ship with the level of detail shown. However, the global models are mostly used for strength analyses too, which require a more accurate modelling of the structure in these areas. If the bending stiffnesses of the deck grillages are also to be included in the global model, the representation of the transverse and longitudinal girders of decks is necessary, at least in the form of beam elements. Normally, these models possess 40 to 80 thousand degrees of freedom and have 300 to 500 natural frequencies in the range up to 20 Hz. An alternative for taking account of deck grillages in the form of beam elements is *to model the webs of girder by means of the plane elements and the flanges by truss elements*. Fig.2.4 shows a FE model of the yacht where webs of the deck grillages are modelled three-dimensionally. For larger ships this procedure would have led to unnecessary large and complicated models.

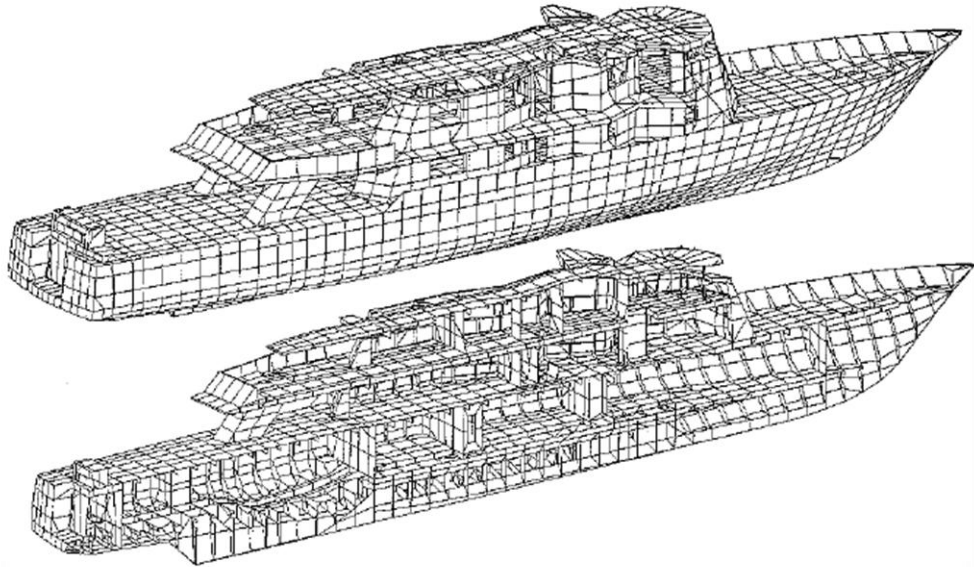


Figure 2.6 FE of 60m long yacht

In the computation of global vibration of ships, it must be borne in mind that natural frequencies are highly dependent on the loading condition. From a draught variation of about ± 1.0 meter, it should be considered to take a further loading condition into account. For cargo vessels, therefore, at least two or three mass distributions have to be considered. In contrast to strength analyses, no extreme cargo distributions should be selected, but rather homogeneous ones typical for the expected ship operation.

The following masses must be taken into account:

- Ship structure
- Outfitting and equipment
- Tank filling
- Cargo
- Hydrodynamic masses

In FE techniques, a distinction is drawn between node masses and element masses. Node masses are concentrated at the respective nodal points of the FE model. This arrangement of masses is advisable for heavy parts of equipment whose centres of gravity are not automatically evident from the model geometry. For the arrangement of structure masses,

as well as for “distributable” part of equipment masses, the existing geometric information of the FE model should be used (element masses).

The masses of tank contents are distributed over the nodes of the relevant tank structure, taking correct account of the centres of gravity. If nodes are available, the same applies to cargo masses. However, in many cases, for example for container masses, auxiliary structures must be provided to introduce masses into FE model in a realistic manner. It must be ensured that auxiliary structures do not unacceptably stiffen the ship’s hull.

To determine hydrodynamic masses, separate computations must be performed. The procedures used are still often based on the method of Lewis, which involves a 2D theory derived for elongated, slim bodies. The associated set of potential-theory formulas is based on conformal mapping of a circular cross-section. The water flow in the ship’s longitudinal direction is taken into account by correction factors that depend mainly on the length-to-width ratio, and also on the natural mode being considered. Because hydrodynamic masses have to be determined prior to the calculation of natural vibrations, the selection of correction factors should be co-ordinated with the expected frequency range of natural modes. Strictly speaking, it is possible to accurately determine only the natural frequency of the particular mode used as the basis to select correction factors.

The Lewis method offers the advantage that the hydrodynamic mass matrix to be used for the eigenvalue solution contains terms on the main diagonal only. Thus the same numerically effective algorithms can be used for solving the eigenvalue problem, as those used for problems involving only structured masses.

More comprehensive methods to calculate hydrodynamic inertia effects take into account the fact that acceleration of a point on the wetted shell also causes changes in the hydrodynamic pressure at adjacent points. This coupling leads to the introduction of terms on the secondary diagonals of the mass matrix, which leads to a considerably more effort-intensive calculation of the eigenvalues.

2.1.2 Calculation

If stiffness and mass matrices are known, natural vibration calculation can be performed. For this purpose, numerically effective approximation methods, such as the Ritz procedure, are used. For the eigenvalue solver, starting vectors must be specified, the superimposition of which permits as accurate a representation as possible of expected vibration modes. However, only mode shapes can be calculated for which corresponding starting vectors have been specified.

As starting vectors the Lanczos method for instance, selects in an automated manner unit load cases that act in every degree of freedom of the system. This leads to the computation of all existing natural frequencies in the desired frequency interval. At present, the natural vibration analysis of a large global model takes several hours on high-performed workstation.

To illustrate the situation, some typical fundamental natural vibration modes calculated for the previous FE models shown in Fig.2.5

In each case, the first torsional vibration mode and the second vertical bending vibration mode are presented together with the computed natural frequencies. Because of the large deck-opening ratio, the natural torsional frequencies for container ships are low. As a result of the comparatively short deckhouses there is no significant stiffening effect on the ship's hull.

For the other ship types, on the other hand, it can be assumed that the superstructures contribute considerably to hull stiffness.

Vibration modes of ship hulls lie in the lower frequency range. Because of the usual higher excitation frequency their contribution to the vibration level is small. Nevertheless, knowledge of these vibration modes is important for validation purposes.

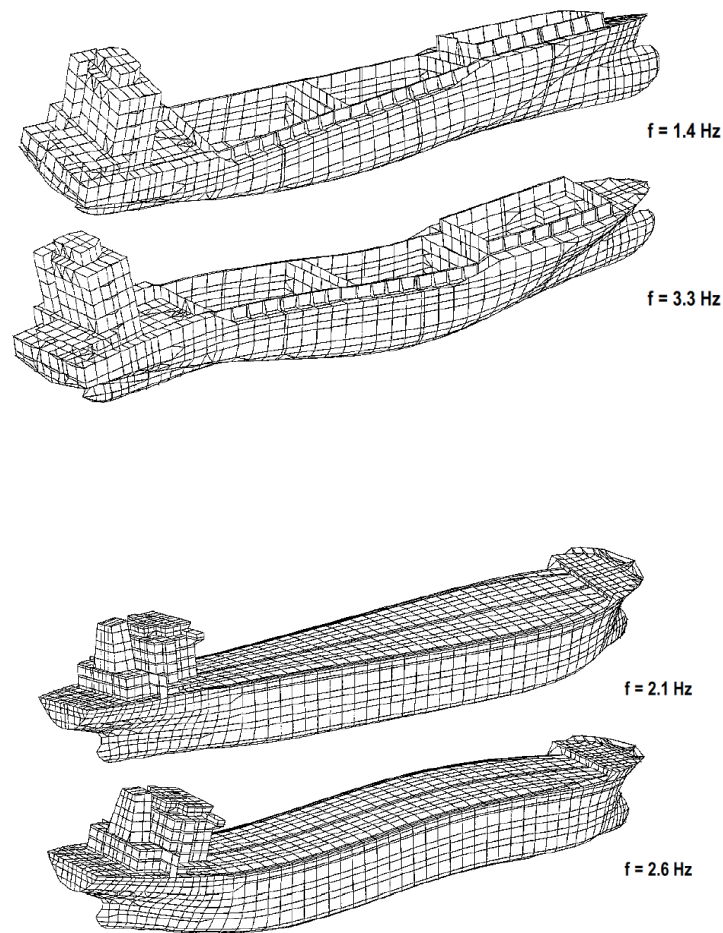


Figure 2.5 Natural torsional and vertical bending modes of various ship types

2.2 Local structures

Because of comparatively high natural frequencies of local ship structures, FE models for their calculation must be detailed. In particular, bending stiffnesses of local structures must be considered as realistically as possible, in contrast to their representation in global computations. The aim of local vibration investigations is usually to limit vibration magnification relative to the global level. Thus, for example, vibration amplitudes at the centre of a deck grillage of an accommodation deck should not be much larger than at stiffly supported edges. This can be achieved only if freedom from resonance exists for all structural components of the deck.

In calculation practice, a distinction is drawn between vibrations of plate fields, stiffeners and panels (grillages) – see also Fig. 2.6. The amount of the effort needed for the creation of FE models of such structures should not be underestimated. In spite of parameterized input possibilities and extensive graphic support, experience has shown that

this type of analysis can hardly be carried out within the given time schedule. In addition, other important parameters of influence, such as rotational stiffnesses at plate field edges and effective mass distribution (including hydrodynamic added mass) also have to be taken into account here.

There is a strong interaction between local vibrations of structures and ship's acoustics. This relationship is manifested by the fact that a ship whose local structures have been consistently designed in respect to vibration also gains acoustic advantages.

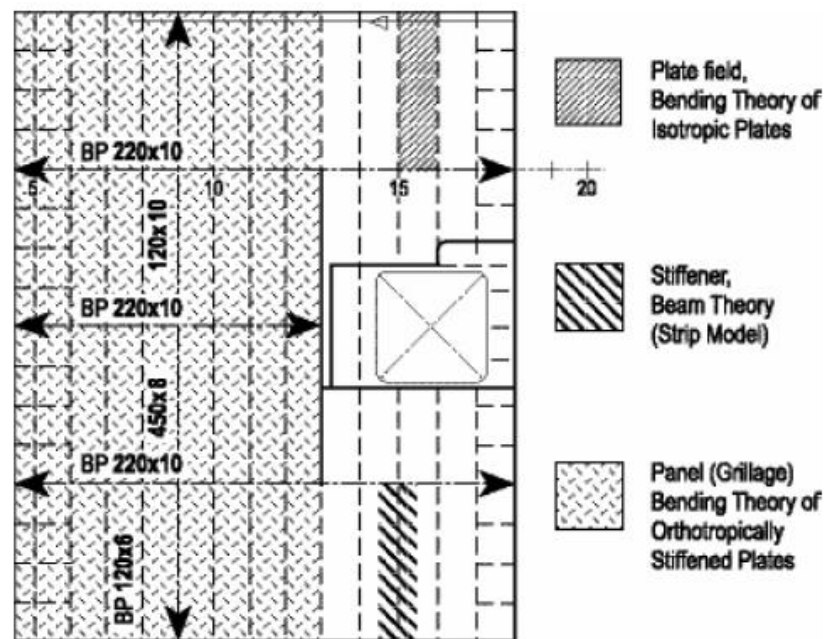


Figure 2.6 Structural components in local vibration calculations.

2.3 Substructures

In the transition between global and local vibrations, vibrations of large subsystems are interest in practice too. Here subsystems are structures of equipment items, whose natural vibration characteristics can be regarded, for the sake of simplicity, as being independent of the vibration behavior of the structure surrounding them – which is the case a vibrating radar mast for example. However, in the analysis of subsystems, the surrounding structure must not be ignored, because it defines the connecting stiffness, i.e. the supporting conditions. *The vibration of the main diesel engine can be referred to that level of ship vibrations.* But at first basic aspects of the superstructure natural vibrations will be shown as sometimes engine excitation causes high amplitude displacements of a deckhouse.

2.3.1 Deckhouses

The aim of analyses of this type is the avoidance of resonance between fundamental vibration modes and main excitation frequencies. A typical example of a substructure is a deckhouse when considered as an isolated system. Fig.2.7 shows such a model with the calculated fundamental vibration modes.

The longitudinal and transverse vibration modes, in particular, are significantly affected by the vertical stiffness in the supporting area. Therefore, an attempt must be made to incorporate, in a simplified manner, an appropriate part of the ship's hull in the region of the deckhouse into the model.

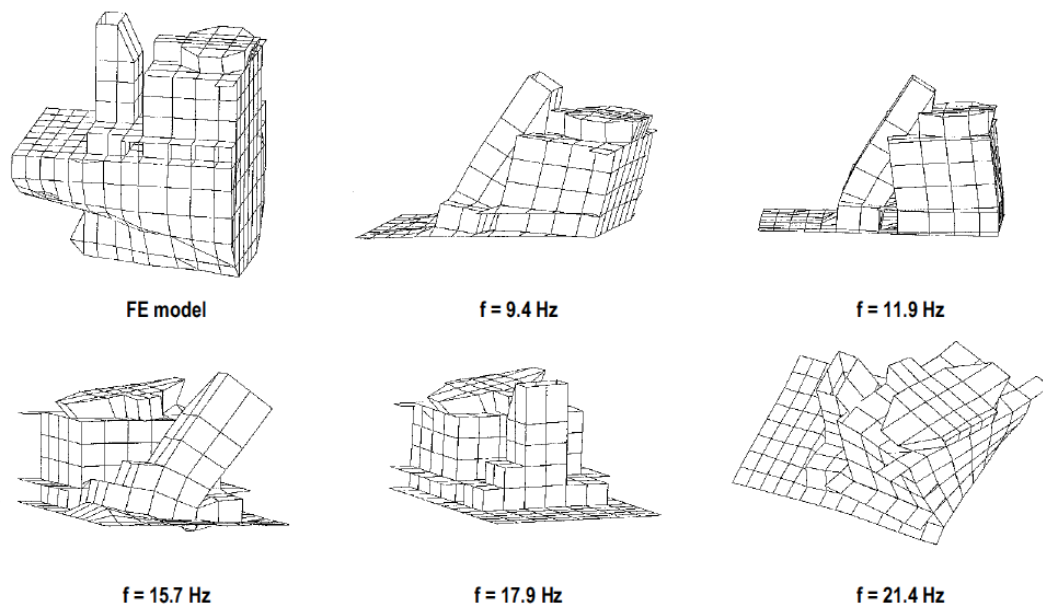


Figure 2.7 Natural vibration of a deckhouse

In this way, it is also possible to investigate the effect of design changes in the deckhouse foundation on the vibration behavior. As can be seen from the natural vibration modes presented, the foundation is stiffly constructed. There are two coupled natural modes for longitudinal vibration of the deckhouse and funnel. The low-frequency vibration is the in-phase vibration, whereas these subsystems in the following vibration mode vibrate in the anti-phase mode at 11.9 Hz. Because of the stiff foundation, the natural frequency is defined by shear stiffness of the deck-house.

Global transverse vibration of the deckhouse does not exist in the considered frequency range. The supporting structure governs the vibration behavior of the funnel as well, leading to a natural frequency of 15.7 Hz for the transverse mode. Vibration of the upper region of the deckhouse occurs at 17.9 Hz. This natural frequency is defined mainly by the grillage stiffness of the bridge deck.

The natural torsion vibration frequency is found to have a comparatively high value of 21.4 Hz because of the large external dimension of the deckhouse.

The design proves to be advantageous from the point of view of vibration because the basic recommendation had been adopted:

- Minimum possible height and maximum possible length and width of the deckhouse
- Stiffly designed foundation, especially the arrangement of bulkheads or wing bulkheads under the fore and aft bulkheads of the deckhouse (alternatively: support of the longitudinal deckhouse walls on longitudinal bulkheads in the ship's hull)
- Maximizing the longitudinal shear stiffness of the deckhouse by means of continuous longitudinal walls having as few and small cut-outs as possible

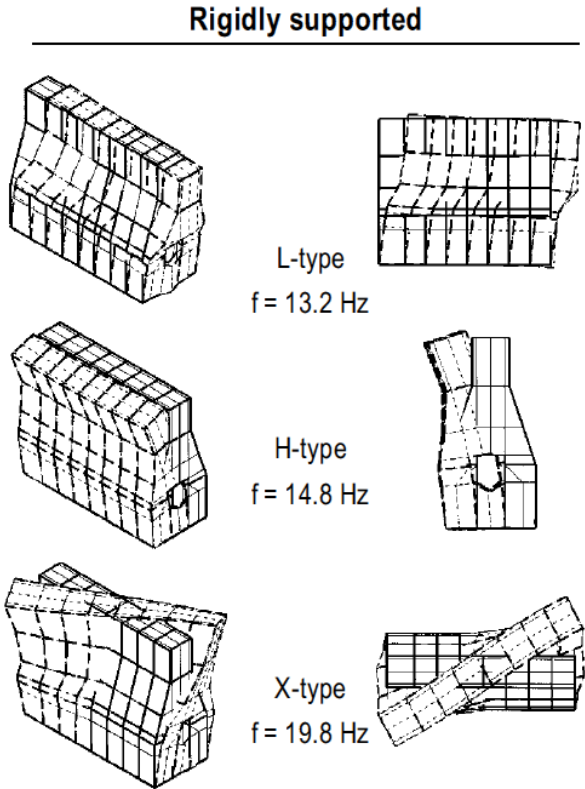
For container ships and bulk carriers, in particular, the first two of these recommendation are often unachievable, since deckhouse are designed to be both short and tall to optimise stowage space. For the same reason, deckhouse are additionally often situated far aft in the vicinity of the main sources of excitation. Thus a risk of strong vibration exists in many cases.

However, it is not possible to assess, on the basis of such models, whether resonance situation may lead to unacceptably high vibration, since coupling with hull vibration cannot be taken into account. Thus, for example, vertical vibrations of the aft part of a ship lead to longitudinal vibrations in the upper region of the deckhouse. These vibration attain a significant level in many cases. This situation can only be investigated in a forced vibration analysis by taking into account of stiffness and mass characteristics of the entire hull and by considering excitation forces realistically. It is not least due to this fact that an isolated consideration of deckhouses is increasingly giving way to complete global vibration analyses.

2.3.2 Engine/foundation system

The natural vibration of ship's main engines is described.

Fundamental natural frequencies of main engine vibrations depend on the distribution of stiffnesses and masses of the engine itself, but they are also determined to a large extent by the stiffness of adjoining structures. The effect of the double-bottom stiffness is more marked for slow-running engines than for medium-speed ones. Fig. 2.8 shows natural modes of a slow-running, rigidly mounted 7-cylinder engine, compared to those of the engine supported realistically in the ship. Furthermore, corresponding natural frequencies are given for an infinitely rigid engine structure supported on a realistic ship foundation. The global stiffness of the engine housing is represented in a simplified form by means of plane stress elements.



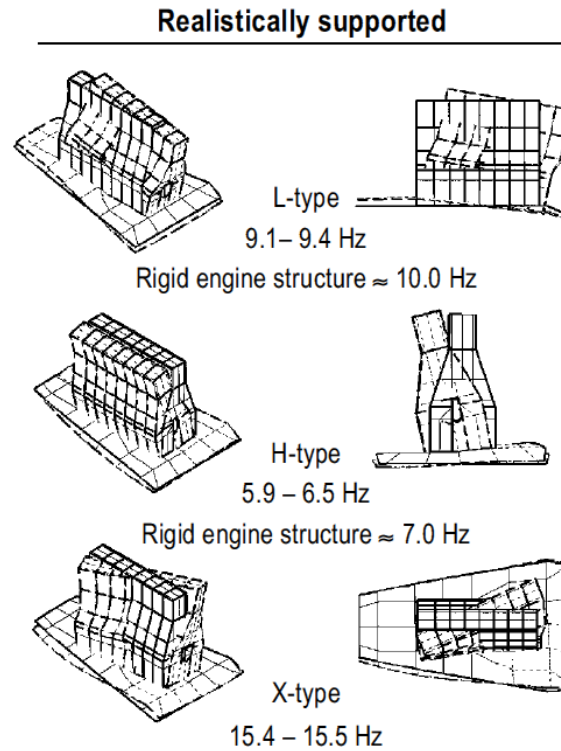


Figure 2.8 Natural vibrations of slow-running main engines for various boundary conditions.

Fundamental vibration modes of housing – called “*H*”, “*X*” and “*L*” modes – depend mainly on the doublebottom stiffness. Since doublebottom designs for slow-running main engines do not differ significantly, bands for the probable natural frequencies can be derived for engines having a certain number of cylinders.

For slow-running engines resonance situation can be experienced for all three fundamental modes, which typical combination of number of cylinder and speed.

In the case of medium-speed engines this is true only for the H-type vibration mode, which might be in resonance with the ignition frequency. Corresponding computation models should contain at least the doublebottom structure in the engine room area and the structure up to the next deck. However, the engine housing must be included in the model too. Because the effect of the engine’s frame stiffness is more marked for medium-speed than for slow-running engines, the engine structure must be simulated with greater accuracy. A computation model with a typical level of detail of engine and ship structure is presented in Fig. 2.9. This shows the port half of the engine room area of a RoRo trailer ferry powered by two 7-cylinder, 4.400 kW main engines driving two propellers.

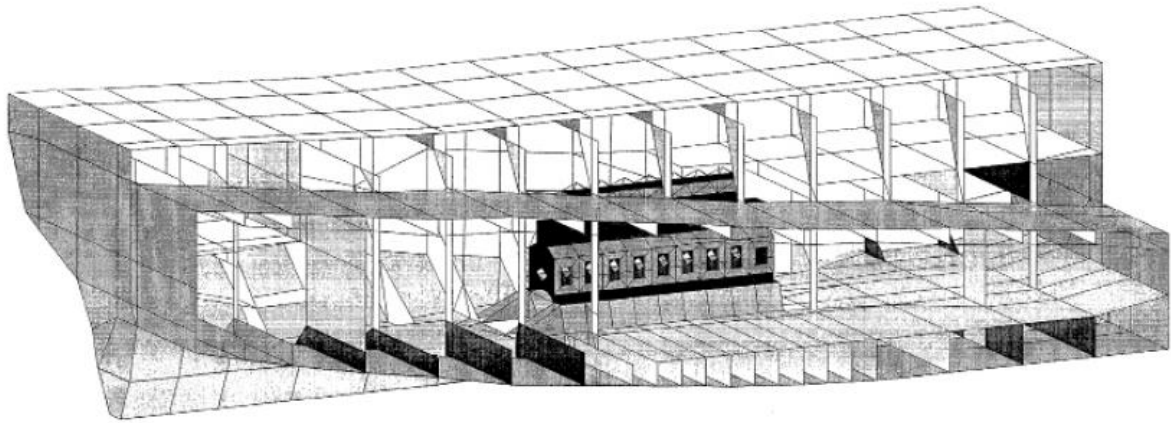


Figure 2.9 Computational model for determining the natural transverse bending frequencies of medium-speed engines

Because the transverse members in the ship's aftbody, which tapers off in a catamaran-like manner, are not very stiff, the task was to check the risk of resonance between transverse modes of the engines and the ignition frequency. Because H-moment also leads to vertical vibration of the doublebottom, hydrodynamic masses act on the ship, which have to be considered.

Large tank-filling in the vicinity of the main engines are taken into account as well.

For this example various natural frequencies were determined, reflecting coupled vibrations of the port and starboard engines.

Fig. 2.10 shows three corresponding vibration modes. Depending on coupling conditions of the port and starboard engines, H-types transverse vibration modes occur at 17.9, 20.5 and 22.5 Hz. *It is worth to notice an existence of several H-types modes for one structure.* The design was, therefore, supercritical relative to the ignition frequency of 30 Hz. Consequently, there was no need to install an elastic or semi-elastic mounting. Computations of X-type vibration modes of the main engines revealed frequencies in a band between 34 and 38 Hz, thus indicating an adequate safety margin to the ignition frequency as well.

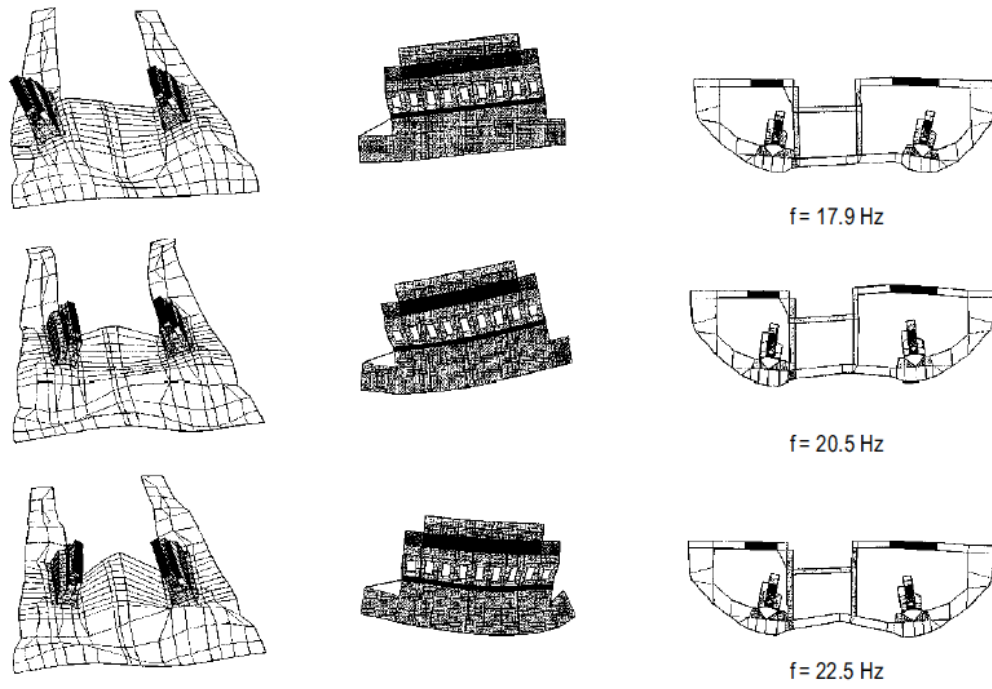


Figure 2.10 H-type natural vibration of two 7-cylinder engines

3. EXCITING FORCES

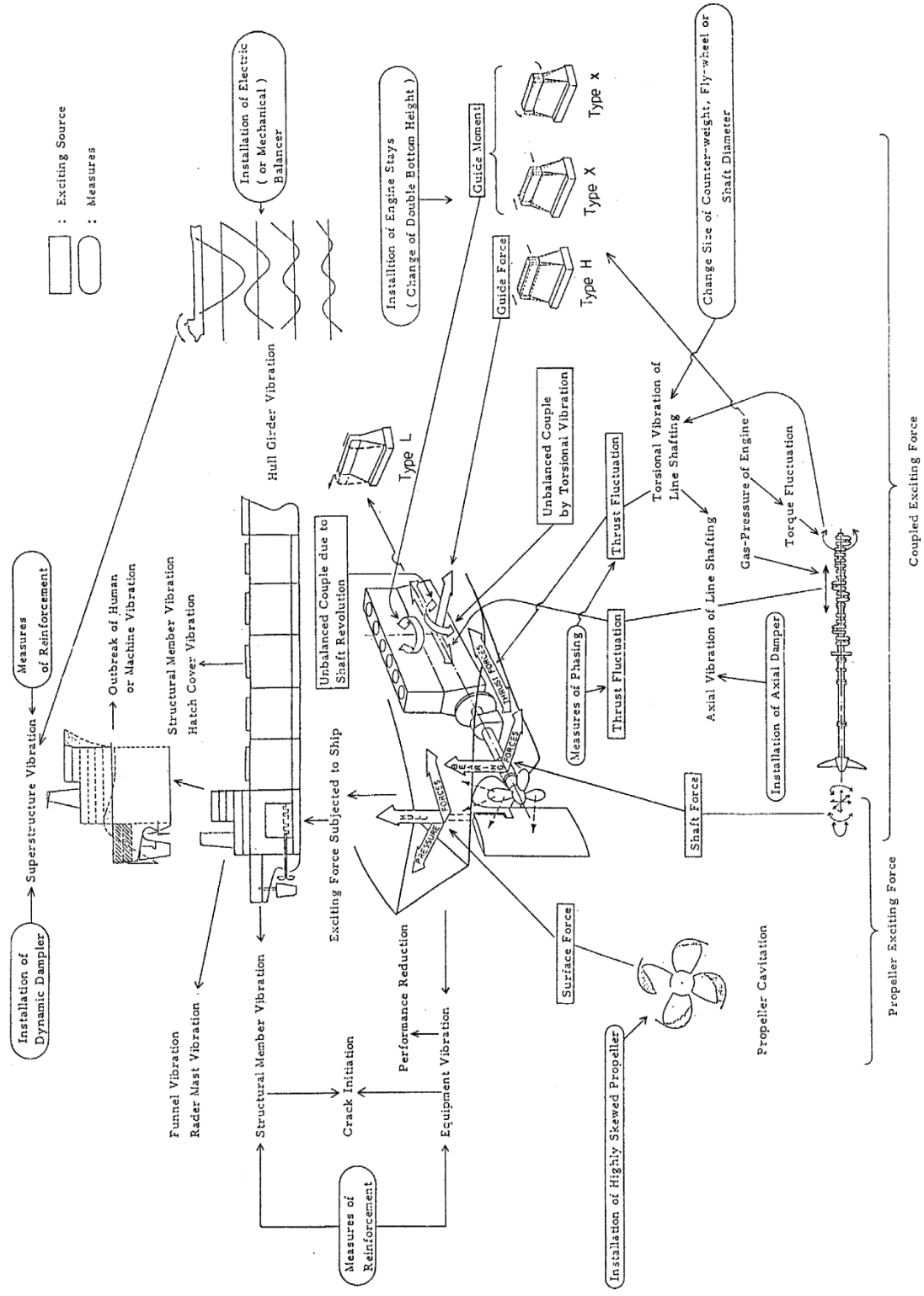


Figure 3.1 Overview of ship excitation forces

Figure 3.1 illustrated the relationship between the exciting force and ship structural vibration, induced by the propeller and main engine. In the figure, an item boxed by a rectangle indicates an exciting force and an item circled by an ellipse shows a countermeasure suppressing the structural vibration which is induced by the exciting force.

The exciting forces of a diesel engine can be categorized by the following components according to the mechanism by which they are produced:

- Unbalanced forces or unbalanced moments induced by inertia forces due to the movement of pistons, etc.
- Guide forces of guide moments which are generated by the explosive (combustion) pressure of gas. These are transmitted to the cylinder of main engine.
- Longitudinal exciting force which is induced by the inertia force of longitudinal deflection on the crankshaft due to gas pressure.
- Fluctuation in thrust force which comes from torque variation in line shaft: Torque variation due to gas pressure causes torsional vibration of line shafting, including the propeller and propeller shaft, and torsional vibration of shafting causes cyclic fluctuation in flow velocity of propellers, which results in thrust fluctuation.

The magnitude of the unbalanced force and the unbalanced moment is so large, in general that they can produce hull girder vibration if they resonate with the natural frequency of the hull girder vibration. Wartsila six-cylinder engines generate second order unbalanced vertical moment with rather significant amplitude. If it makes problems, an electrical balancer (Fig. 3.2) is often installed at the aft of the hull.

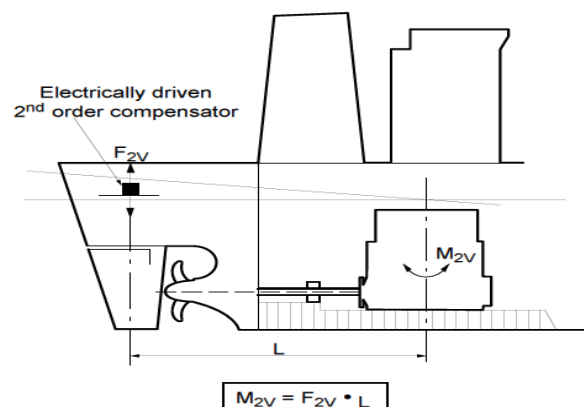


Figure 3.2 Electrical compensator

Guide forces and moments are the exciting forces which induce lateral vibration in the main engine structure. Guide forces cause lateral vibration of the main engine structure which is called “H” type vibration or rocking. Guide moments can cause torsional vibration, that is “X” type, and it can also cause horizontal bending vibration, that is x type. When natural frequency of the main engine structure resonates with the exciting of the guide force or moment, the exciting force is amplified extremely because the engine structure acts as a resonator. Such an increased exciting force is transmitted to the structure members in the engine room and these may cause severe vibration. In order to avoid the lateral vibration of the main engine, it is normally practice to provide engine stays at the top of the engine structure.

Once longitudinal vibration in a crankshaft happens, it causes a longitudinal exciting force on the crankshaft which is transmitted to the double bottom structure of the engine room via the thrust block of the main engine. This exciting force may cause a vibration problem in the superstructure or cause structural failures of panels, stiffeners, etc. To prevent such vibration trouble, a longitudinal vibration damper is usually installed at the fore end of the main engine crankshaft. This damper, which is a kind of oil damper, can reduce the longitudinal vibration amplitude by absorbing the vibration energy.

The fluctuating thrust force due to torsional vibration of the line shafting will cause longitudinal vibration of the main engine structure. If the natural frequency of the longitudinal vibration of the engine agrees with the frequency of the thrust fluctuation, the fluctuating thrust is magnified by this resonance, and this vibration is transmitted to structural members through the thrust block. In such cases, the installation of a torsional damper is a quite effective countermeasure to suppress torsional vibration of the line shafting.

As a problem of H-type engine vibration is being studied so guide forces should be examined more properly. Just to remember that “H” type lateral vibration are characterized by a deformation where the driving and free end side of the engine vibrate in the same phase.

Figure 3.3 reveals how the gas pressure is transmitted through the crosshead to the engine block.

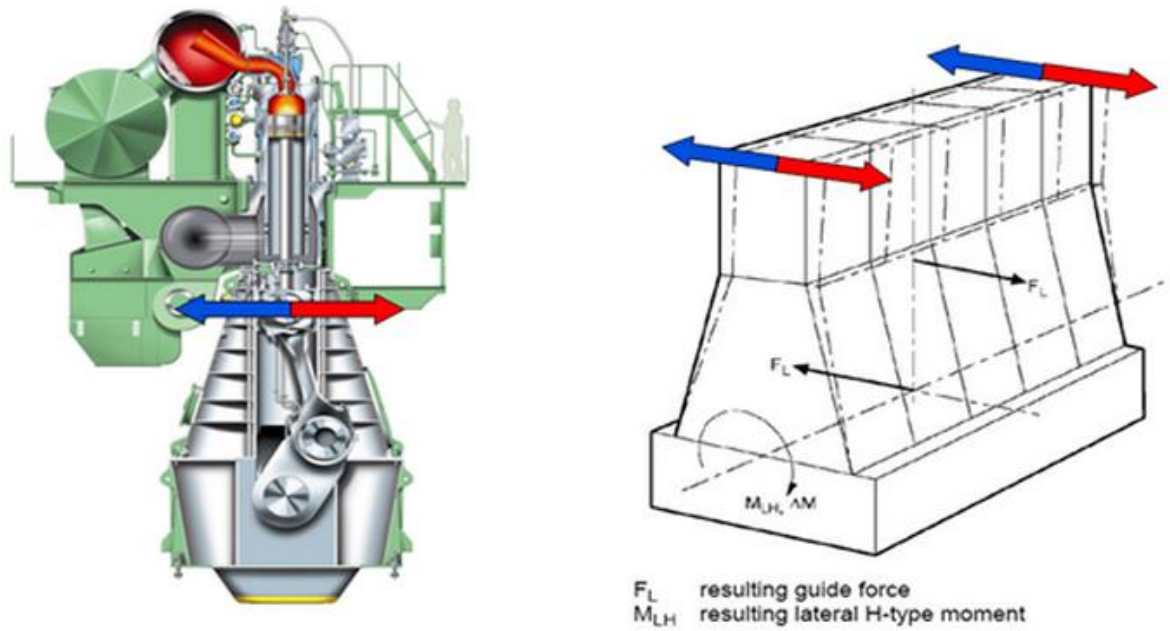


Figure 3.3 Resulting guide forces

Wartsila Company provides information about forces and moments amplitudes. For six-cylinder RTA48T-B total lateral H-moments equals to 498 KN^*m . The frequency is *six-order* one. It means that this frequency is 6 times larger than rotational speed of the crankshaft.

For example if rotational speed of the low-running main engine is around 100 rpm (or 1.66 Hz) then the frequency of the lateral H-moment is 10 Hz.

Forces induced by a propeller are not considered as vibration problems which can be caused by them were not observed on board.

There is also H-moment of 12-order but its amplitude is relatively low (22 KN^*m).

Sometimes it is useful to make a distinction between external and internal forces and moments. The external ones will act as resultant on the engine and thereby also on the ship through the foundation and top stays. The internal force and moments will tend to deflect the engine as such. From a practical engineering point of view H-type moment should be applied to the engine frame as an external moment.

4. BASIC THEORETICAL ASPECTS OF VIBRAION

In this chapter some theoretical information about vibration processes will be presented. This data helps to explain some significant points in the analysis and also reveals limitation of the simulation which was performed.

4.1 Fundamentals of vibration protection

Let us consider the problems of protection against vibration using the simple example of a system with single degree of freedom. This is a body with the mass “m” making forced rectilinear oscillations.

The necessity of protection against vibration emerges in two cases:

- 1) When it is necessary to reduce the vibration impact on the foundation of some machine which appears during its operation.
- 2) When it is necessary to protect some device (or the crew, the instrument, etc.) from the harmful impact of vibrations uprising during transportation or as a result of the operation of machines and equipment which are near.

In the first case the exciting force is applied directly to the body itself (see pic.4.1a) – so called force excitation of vibration. In the second case the kinematic excitation takes place due to the base vibration (see pic.4.1b).

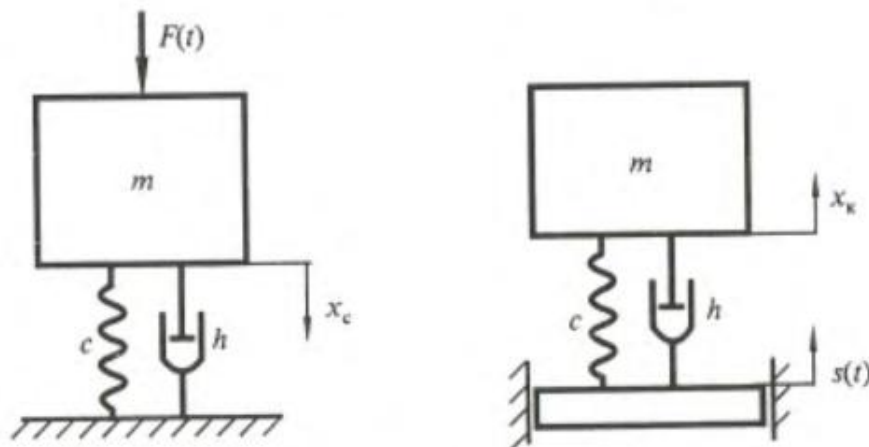


Figure 4.1 Scheme for force excitation and kinematic excitation

In case of force excitation of vibration the low impact on the foundation is needed if possible. As a rule all sorts of engines, motors, machines are mounted on the special foundation with the shock absorbing pads (crash pads) made of specific sorts of ribbon which has both elasticity and high internal inelastic resistance. Such pads can be approximated by a spring with the stiffness “c” and a damper with the viscous resistance (drag) coefficient “h” (see pic.4.1a). Then the dynamic impact on the foundation is the following:

$$R(t) = cx_C + h\dot{x}_C, \quad (4.1)$$

where x_C – body movement (displacement) relative to the position of static equilibrium.

In case of kinematic oscillations excitation we need to make the absolute body displacement x_K as small as possible (see pic. 4.1b).

Because the gravity force is compensated by the static deformation of the spring, for the case of force excitation the equation of the body motion is the following:

$$m \ddot{x}_C + h\dot{x}_C + cx = F(t), \quad (4.2)$$

and for the case of kinematic excitation:

$$m \ddot{x}_K + h[\dot{x}_K - \dot{s}(t)] + c[x_K - s(t)] = 0,$$

or

$$m \ddot{x}_K + h\dot{x}_K + cx_K = cs(t) - h\dot{s}(t), \quad (4.3)$$

where h - the viscous resistance coefficient in the material of spring or specially mounted damper (for example automobile shock absorber); c – suspension spring stiffness.

We assume that exciting force and foundation vibration are changed harmonically. According to the complex amplitude method the solution for the equation (1.73) is defined as $x_C = Ge^{ipt}$.

After substituting x_C into (1.72) and (1.73) we obtain the following:

$$R(t) = (c + hip)Ge^{ipt};$$

$$F(t) = (-mp^2 + hip + c)Ge^{ipt}.$$

Therefore the ratio of the dynamic impact on the foundation to the acting force is:

$$\frac{R(t)}{F(t)} = \frac{(c + hip)}{(c - mp^2 + hip)} = \frac{\omega^2 + 2\varepsilon ip}{(\omega^2 - p^2 + 2\varepsilon ip)}.$$

In our case we are interested only in the ratio between the amplitudes $R(t)$ and $F(t)$. We are not interested in the phase shift.

Let us denote the ratio between the amplitudes $R(t)$ and $F(t)$ as β_C :

$$\beta_C = \frac{\sqrt{\omega^4 + 4p^2\varepsilon^2}}{\sqrt{(\omega^2 - p^2)^2 + 4p^2\varepsilon^2}}. \quad (4.4)$$

In case of kinematic excitation after substituting $x_K = Ge^{ipt}$ and $s(t) = s_0e^{ipt}$ into (4.3) we obtain the following:

$$(c - mp^2 + hip)x_K(t) = (c + hip)s(t),$$

whence

$$\frac{x(t)}{s(t)} = \frac{(c + hip)}{(c - mp^2 + hip)} = \frac{\omega^2 + 2\varepsilon ip}{(\omega^2 - p^2 + 2\varepsilon ip)}.$$

As well as in the case of force excitation we are interested only in the ratio between the amplitudes $x_K(t)$ and $s(t)$.

Let us denote the ratio between the amplitudes $x_K(t)$ and $s(t)$ as β_K :

$$\beta_K = \frac{\sqrt{\omega^4 + 4p^2\varepsilon^2}}{\sqrt{(\omega^2 - p^2)^2 + 4p^2\varepsilon^2}}. \quad (4.5)$$

Comparison of (4.4) and (4.5) shows that these two equations coincide.

Therefore the condition of vibration protection for both force and kinematic excitation is the following:

$$\beta = \beta_C = \beta_K = \frac{\sqrt{\omega^4 + 4p^2\varepsilon^2}}{\sqrt{(\omega^2 - p^2)^2 + 4p^2\varepsilon^2}} < 1. \quad (4.6)$$

Let us introduce the detuning factor $z = \frac{p}{\omega}$ and dimensionless damping coefficient $d = \frac{2\varepsilon}{\omega}$.

Then after dividing both numerator and denominator of (4.6) by ω^2 we obtain the condition of vibration protection in the following form:

$$\beta = \frac{\sqrt{1 + d^2 z^2}}{\sqrt{(1 - z^2)^2 + d^2 z^2}} < 1$$

whence

$$(1 - z^2)^2 + d^2 z^2 > 1 + d^2 z^2,$$

or

$$z^2(z^2 - 2) > 0.$$

This equation takes place when $z > \sqrt{2}$ or $p > \sqrt{2} \cdot \omega$.

Functional relationship $\beta(z)$ is shown in the pic.1.27. It can be seen that independently from the type of excitation and the value of viscous resistance coefficient for protection against vibration the natural frequency of the system oscillations must be considerably lower (at least $\sqrt{2}$ times) than the excitation frequency.

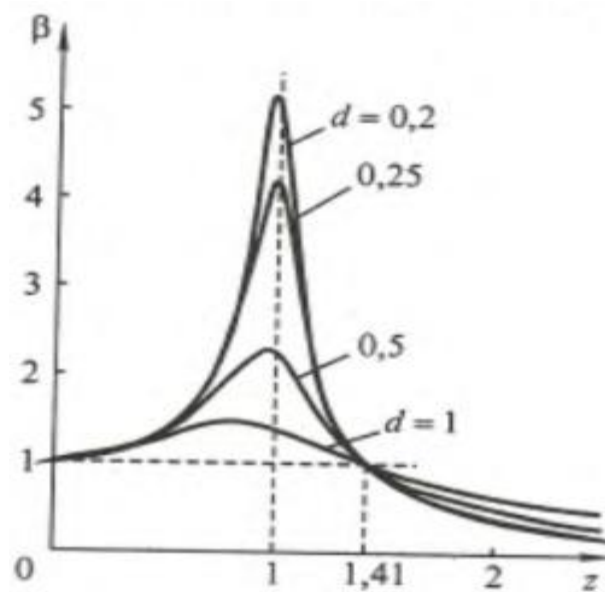


Figure 4.2 Relation $\beta(z)$

The graph $\beta(z)$ shows that when $p > \sqrt{2} \cdot \omega$ (in vibration protection area) damping has a negative effect because the less the damping - the greater the vibration protection. It looks like the damping should be reduced but it is not always true.

We should take into consideration the fact that in case of force excitation every machine after the start goes through the spin-up condition and the stall condition before the stop. During this process the excitation frequency changes from 0 to p and back, which means that the system goes through resonance. That fact forces the designer to increase the damping in order to reduce the amplitude of the resonance oscillations even though it harms vibration protection. And in case of kinematic excitation there is a possibility of the abrupt movement of the base (hurdle hit, wheel falling into the road hole, etc.). If the damper is absent this can cause unacceptable displacement and lead to the overload on protected against vibration object as well as too long process of free decaying.

4.2 Coulomb friction

As main engine can be completed with friction type side stays it is important to get a view in some basic aspect of vibration process where dissipation force is Coulomb (dry) friction. There is one-degree of freedom system which includes a mass and a spring. The mass moves forward and backward on the coarse surface (see Fig. 4.3).

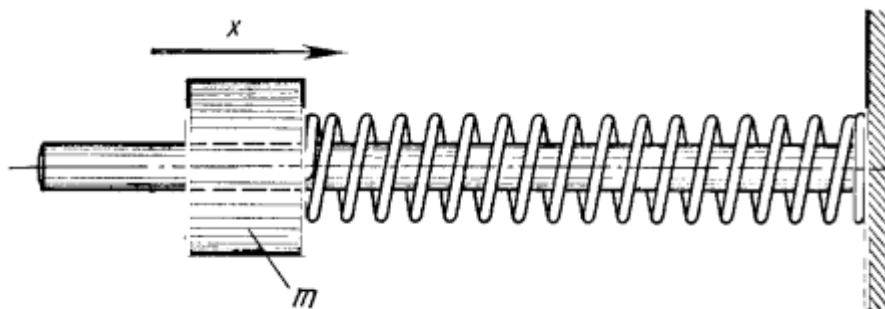


Figure 4.3 One degree of freedom system

Friction force has constant amplitude and has opposite direction relative to mass displacement. Free vibration equation:

$$m\ddot{x} + cx \pm R_0 = 0$$

Where sign plus corresponds to the case of positive velocity and sign minus correspond to the negative one.

Total force as a function of displacement:

$$F = cx \pm R_0$$

Rewrite equation () as:

$$m\ddot{x} + cx + R_0 * \operatorname{sgn}\dot{x} = 0 \quad (4.7)$$

Where $\operatorname{sgn}\dot{x}$ is a singular function with a sign of an argument (see Fig. 4.4).

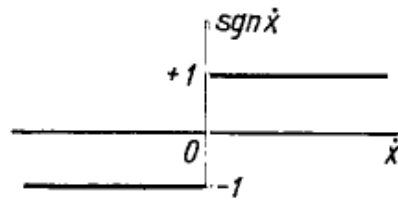


Figure 4.4 Function $\operatorname{sgn}\dot{x}$

Equation (4.7) has *nonlinear component*. Therefore it is impossible to perform forced linear harmonic analysis with Coulomb friction force.

Phase portrait of the one degree of freedom system is shown on Fig. 4.5. It is worth to notice diapason between $-a$ and a . This region is called a stagnation zone. If initial displacement is less than a then vibration does not occur. If in the final of the process displacement of the system comes in that region process stops.

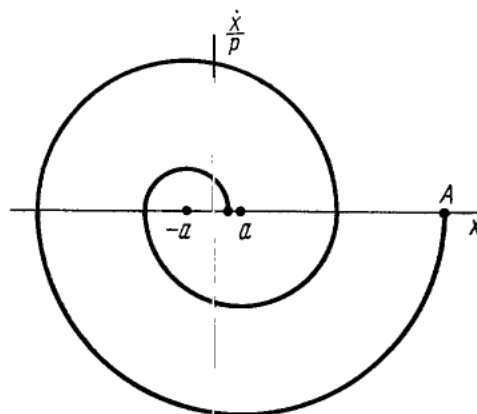


Figure 4.5 Phase portrait

4.3 Passing through the resonance

Big amplitudes of vibrations under the resonance condition are the problem and it is better to avoid it. At the same time it happens that a system has to be operated under a frequency of excitation force which is larger than its own natural frequency. In that case system passes the resonance during acceleration and run out

During the passing through the resonance less amplitudes appears than in case of stationary regime as required for oscillation built up energy is delivered in short period.

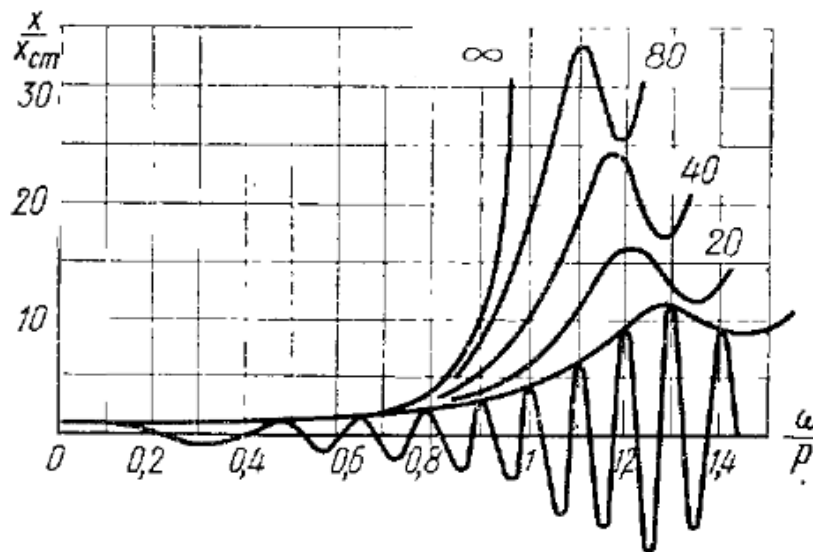


Figure 4.6 Amplitudes for different rate of frequency rise

Curves on Fig. 4.6 show that faster acceleration (lower numbers on the Figure) corresponds to less maximum oscillation amplitude and bigger moment frequency when maximum displacement occurs. All calculations for this Figure were done for a non-damping system.

This task can be classified as a transient analysis type. In this research only stationary regime of vibrations are modeled.

5. FINITE ELEMENT MODEL PREPARATION

To implement numerical vibration calculation it is necessary to have an accurate finite element model of a studied structure. Current model includes several types of elements: solid, shell and beam elements. Following chapters show and reveal main sub-elements of the ship structure and their FE representations.

Accomplished FE model is used for both: *modal* and *harmonic* analysis.

Table 1 shows total number of different elements.

Model features	
Number of nodal points	233831
Number of 8-node solid elements (SOLID185)	41053
Number of 4-node shell elements (SHELL63)	207506
Number of beam elements (BEAM4)	21979
Number of concentrated mass elements (MASS21)	4776

Table 5.1 Size of the whole FE model

Shipyards drawings were used to create numerical model of the hull structure. Information from the manufacturer was used to model main diesel engine. For initial hull modeling Poseidon GL software was used. Engine, generators, shaft line were built in ANSYS Preprocessor directly.

5.1 Main engine

The bulk carrier m/s Miedwie is equipped with the main engine of 6RTA48T-B type produced by Wartsila Company. It is a low-speed, direct-reversible, single-acting, two-stroke engine comprising crosshead-guided running gear with 6 cylinders. This engine is designed for running on a wide range of fuels from marine diesel oil (MDO) to heavy fuel oils (HFO) of different qualities.

Main features are shown in Table 5.2:

Number of cylinders:	6
Cylinder arrangement	In-line
Operation:	2-stroke
Cylinder bore:	480 mm
Piston stroke:	2000 mm
Load, nominal (at Rx):	7800 kW
Speed, nominal (at Rx):	118 rpm
Dry weight:	205000 kg
Wet weight:	225800 kg
Turbocharger (ABB type):	1 x TPL73B12
Scavenge air cooler:	1 x SAC43F

Table 5.2 Main features of the engine

Ships which are equipment with low-speed engines do not have gear box and rotation speed of a crankshaft equals to rotational speed of a propeller. Frequency of shaft rotation is called *first-order frequency*.

Low-speed engines have been started to use because of their fuel-efficient. Also such engines have larger operating life and less noise characteristics.

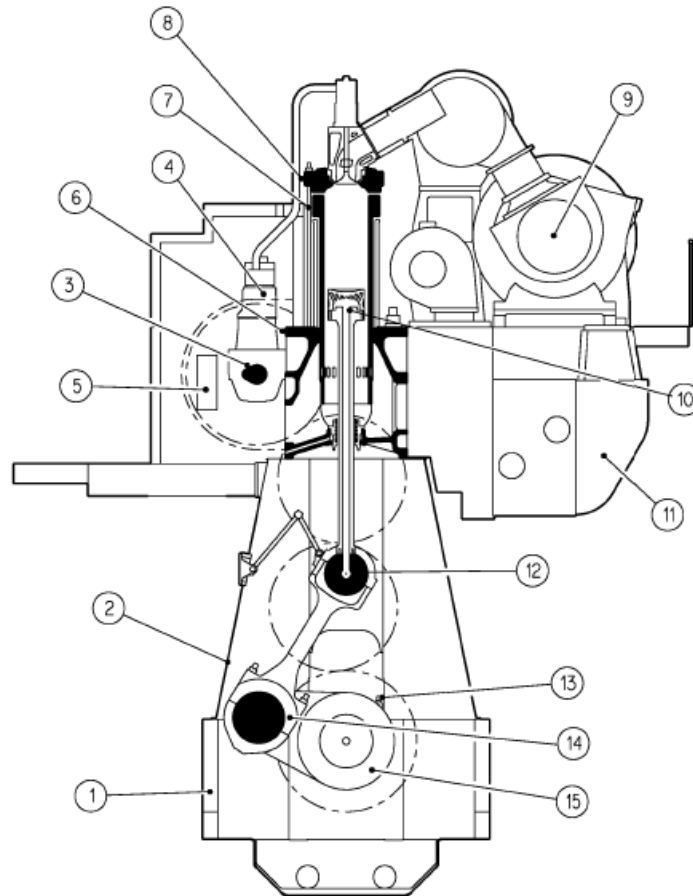


Figure 5.1 RTA48T-B cross section

- 1) Welded bedplate with integrated thrust bearings and large surface main bearing shells
- 2) Sturdy engine structure with low stresses and high stiffness comprising A-shaped fabricated double-wall columns and cylinder blocks attached to the bedplate by pre-tensioned vertical tie rods.
- 3) Fully built-in camshaft driven by gear wheels housed in a double column located at the driving end.
- 4) A combined injection pump and exhaust valve actuator unit for two cylinders each. Camshaft driven fuel pump with double spill valves for timing fuel delivery to uncooled injectors. Camshaft-driven actuator for hydraulic drive of poppet-type exhausts valve working against an air spring.
- 5) Standard pneumatic control – fully equipped local control stand
- 6) Rigid cast iron cylinder monoblock or iron jacket moduls bolted together to form a rigid cylinder block.

- 7) Special grey cast iron, bore-cooled cylinder liners with load dependent cylinder lubrication
- 8) Solid forged or steel cast, bore-cooled cylinder cover with bolted-on exhaust valve cage containing exhaust valve.
- 9) Constant-pressure turbocharging system comprising exhaust gas turbocharger and auxiliary blowers for low-load operation.
- 10) Oil-cooled pistons with bore-cooled crowns and short piston skirts.
- 11) Uniflow scavenging system comprising scavenge air receiver with non-return flaps.
- 12) Crosshead with crosshead pin and single-piece white metal large surface bearings. Elevated pressure hydrostatic lubrication.
- 13) Main bearing cap tightened with down bolts for easier assembly and disassembly of white-metalled shell bearings.
- 14) White-metalled type bottom-end bearings.
- 15) Semi-built crankshaft

A crosshead plays major role in excitation of H-type vibration so it is useful to study it more properly.

Crossheads are used in big slow 2-stroke engines with cylinder diameter more than 450 mm. The aims are to reduce piston's normal pressure on cylinder, to increase gap between piston and liner, full separation volume of casing and work volume of cylinder. Also such scheme of engine has the second work volume below a piston. Crosshead provides bigger reliability and operating life to the parts of piston block.

Connection of a crosshead with a piston by means a rod allows installing a gasket to separate casing and cylinders and to provide leak resistance. This is especially important if lower grade fuels are use for example with higher sulphur content.

To ensure that the crosshead reciprocates in alignment with the piston in the cylinder, guide shoes are attached either side of the crosshead pin. These shoes are lined with bearing material and they reciprocate against the crosshead guide, which are bolted to the frame of the engine.

Using the crosshead design of the engine allows engines to be built with very long strokes – which means the engine can burn a greater quantity of fuel per stroke and develops more power.

The advantages of the crosshead design are:

1. Guide faces take side thrust; this is easily lubricated, wears little and takes side force off the piston and liner running surfaces.
2. Uniform clearance around piston allows for better lubricating oil distribution reducing wear.
3. Simplified piston construction designed for maximum strength and cooling. Extended load bearing skirts found on trunk pistons unnecessary.
4. Due to gland lubricating oil may be optimized for crankcase and cylinder. High alkalinity oils used in cylinder allow poorer quality fuels to be burnt.

The type of crosshead which is used in the engine Wartsila RTA48T0-B is shown on Fig. 5.2

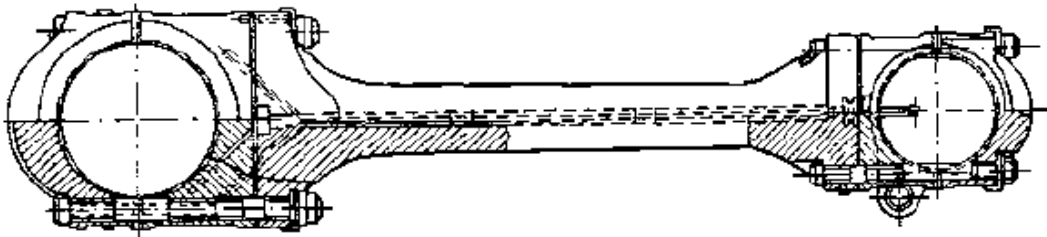


Figure 5.2 Crosshead

To create an accurate FE model of the engine it is necessary to represent mass/inertial and stiffness characteristic as close as possible.

Using geometrical dimensions of the engine from Installation Manual (see Appendix 2, Fig.A2.1 and Fig. A2.2) and knowing position of center of gravity it is possible to assign different densities of the materials to control proper position of CG.

Center of gravity (from Installation Manual):

- Distance from crankshaft flange (fly wheel): 3185 mm
- Vertical distance above crankshaft centreline: 2110 mm
- Transverse distance from crankshaft centreline: 110 to Starboard

Six materials with different densities were used to achieve correct position of the CG (see Fig. 5.3. Different colors mean different material models). The defining of the densities is iterative process and it is better to code the search procedure.

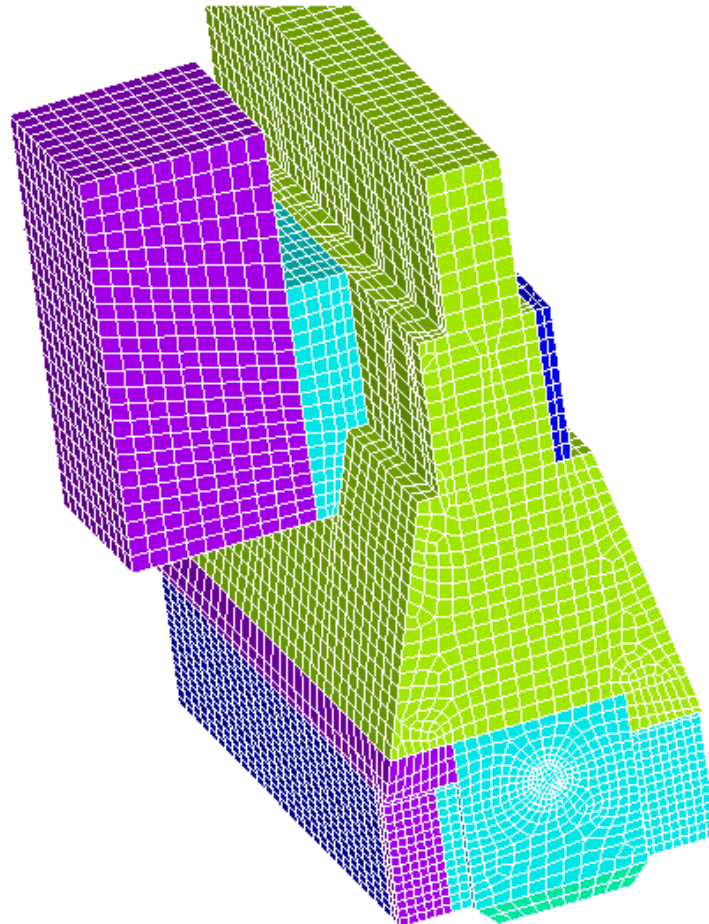


Figure 5.3 Material position

In order to match the engine stiffness it is crucial to create almost exact 3D geometrical model with all details. Internal combustion engine are too complicated and the accurate 3D modeling process can take a lot of time. To pass over this problem it is possible to implement the same trick with several materials properties as in case with CG. Different Young's modulus can be used for longitudinal and transverse structural members. Unfortunately, stiffness data for this engine are unknown (actually such information should have manufacturer, but I could not find it in open sources). Nevertheless using data about H-type natural frequency (obtained in the company) for the mounted engine, proper stiffness parameters were found (see below).

Anyhow engine was modeled as accurate as available drawing allows (see Fig. 5.4). Type of used element for meshing was SOLID 185.

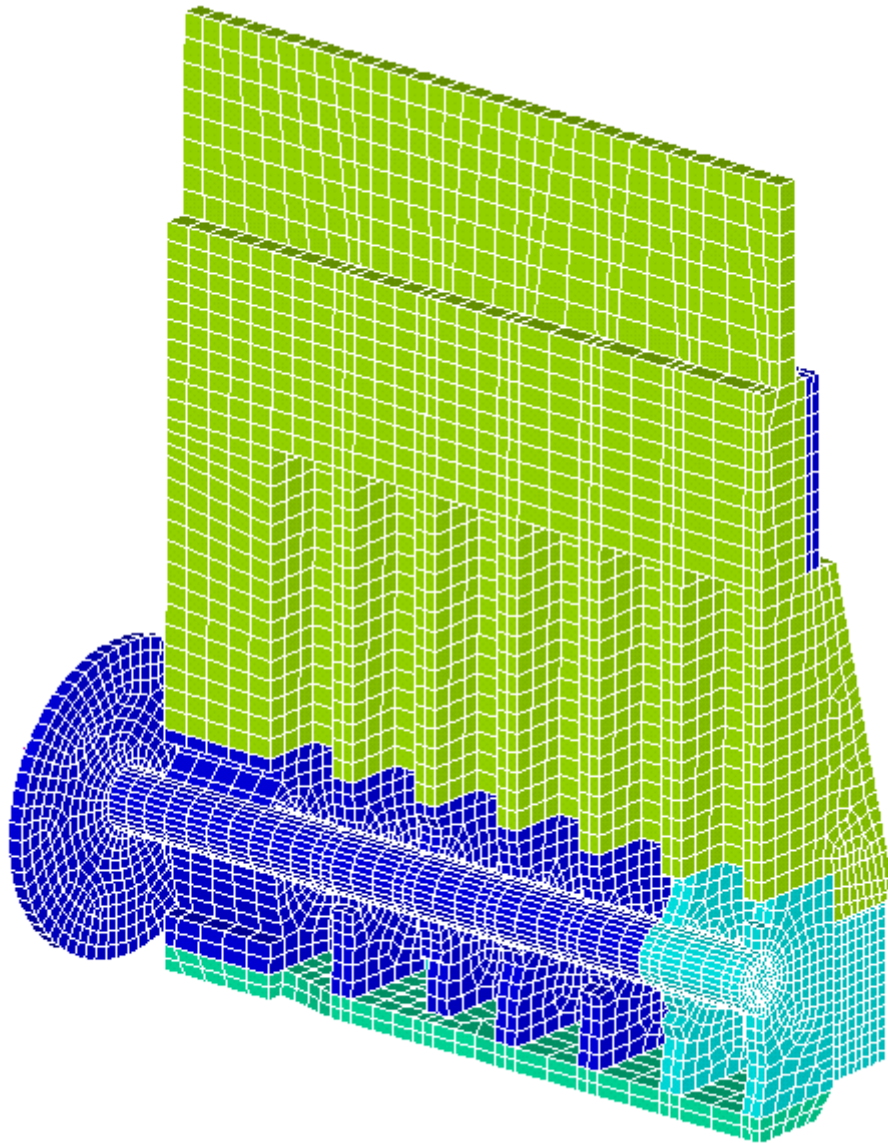


Figure 5.4 Internal structure of the engine

There are the fly wheel and the crankshaft on the Figure 5.4. In spite of the fact that crankshaft is modeled very simplified (just straight cylinder) its presence will ease procedure of the force applying and allow transmitting loads more realistic. Also it is possible to see all internal structure of the engine. Engine is rigidly attached to the doublebottom.

5.2 Engine's platform

Current engine is fitted with several platforms which provide access to the engine from different decks (all ladders are attached to those platforms). Also some equipment is placed on them (see Fig. 5.5). Moreover side stays are connected to the upper platform.

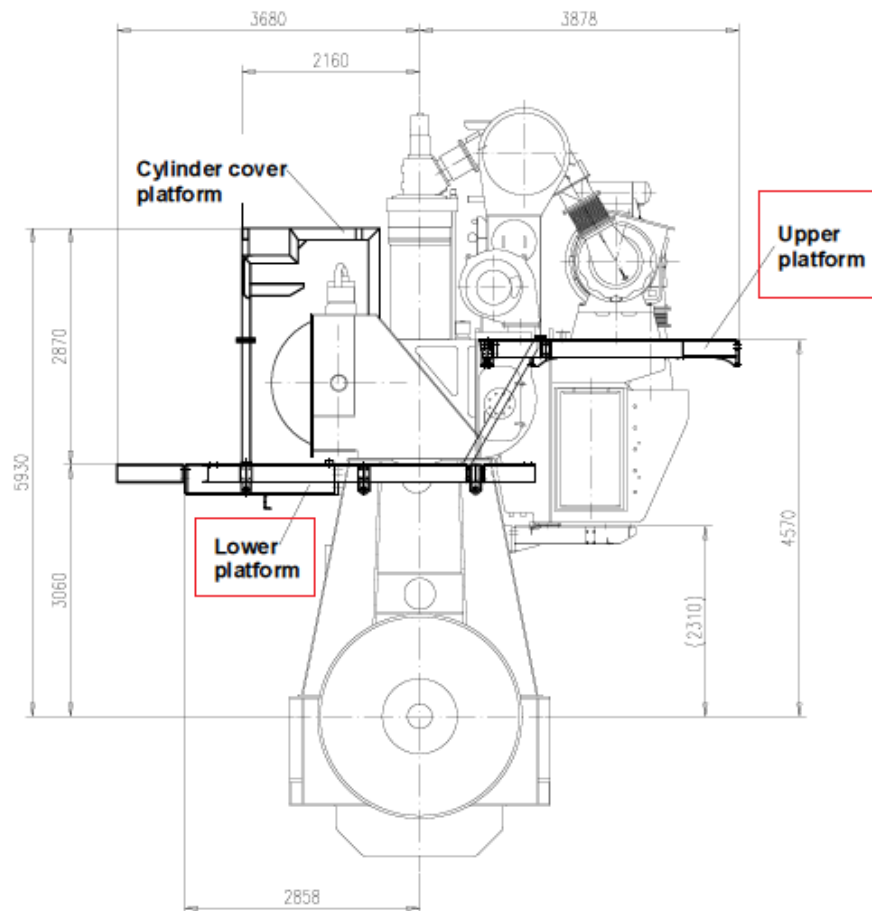


Figure 7.5 Platform arrangements

Wartsila Company offers some typical structure for such platforms. However, the shipyard used own design for the platforms (both for shape of the plate and grillage). The models of the platforms were created by using mainly photos from the board (see Figures in Appendix 3).

Obviously that a platform and its grillage should be modeled by means of shell elements. Indeed for this purpose element SHELL63 was used. SHELL63 is 4-node element and has 6 degree of freedom at each node (see Fig.5.6 and Fig.5.7). A triangular option of this element exists.

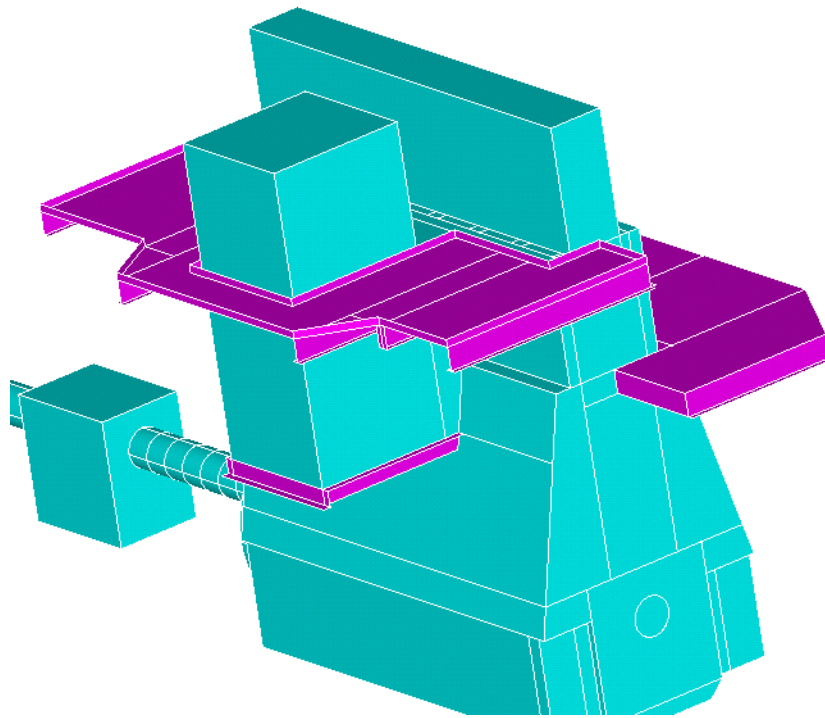


Figure 5.8 Platform position

Stiffness of the platform is rather important for the vibration process when side stays are installed. So the optimal construction parameters of the platform may be investigated.

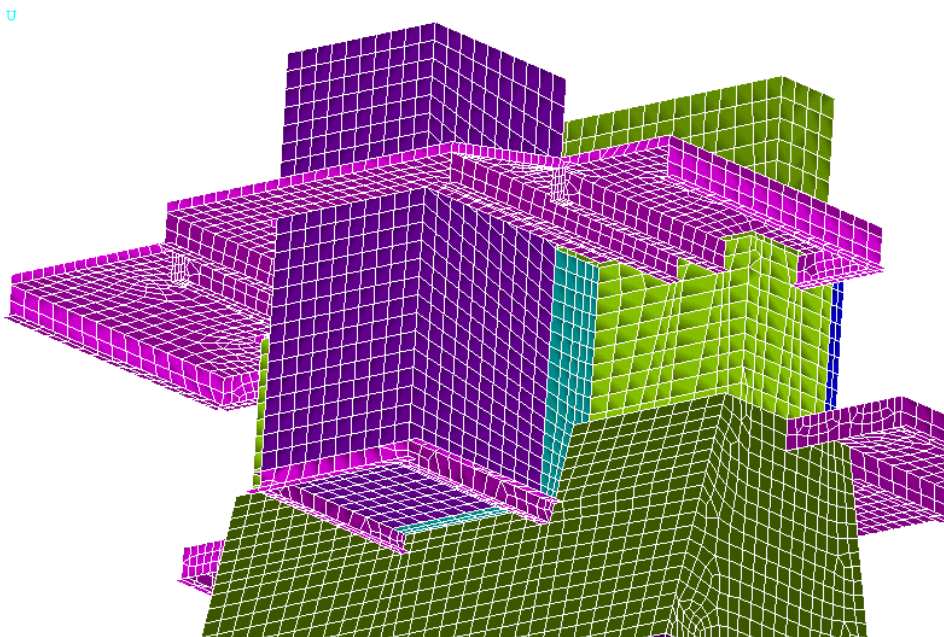


Figure 5.9 FE model of the platforms and the engine

5.3 Electrical generators

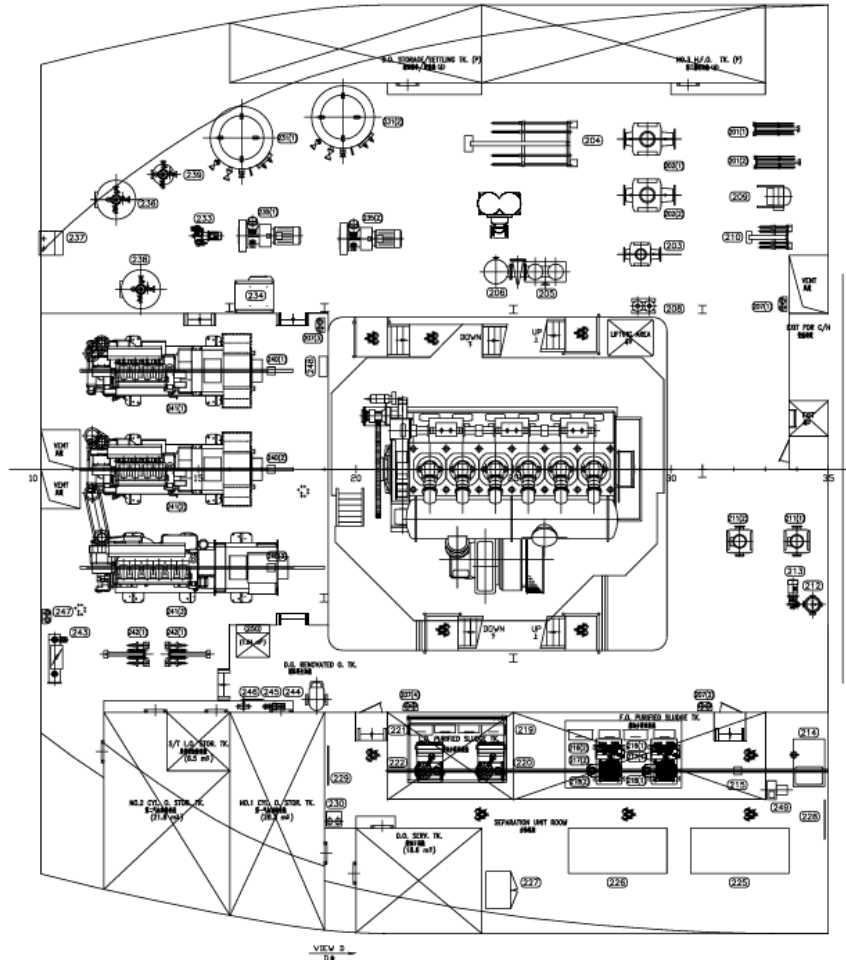


Figure 5.8 Position of the generators on the board (top view on deck 7000)

Electrical generators located on board play very important role. Their task is to produce electrical energy to supply different systems (light, heating, etc.). Due to high humidity, temperatures fluctuations, interaction with salt water marine generators must have special characteristics diverse from land ones.

The bulk carrier is equipped with 3 generators produced by “Wartsila” company. Two of them are Auxpac 645W4L20 with output power 645 kW and the last one is 875W6L20 with the power equals 875 kW. They are placed on the deck 7000 (Fig. 5.8 and Fig. 5.9) behind the main engine. Such generator is an integrated system which includes 4-stroke marine diesel engine with turbocharger and heavy-duty marine-design alternator.

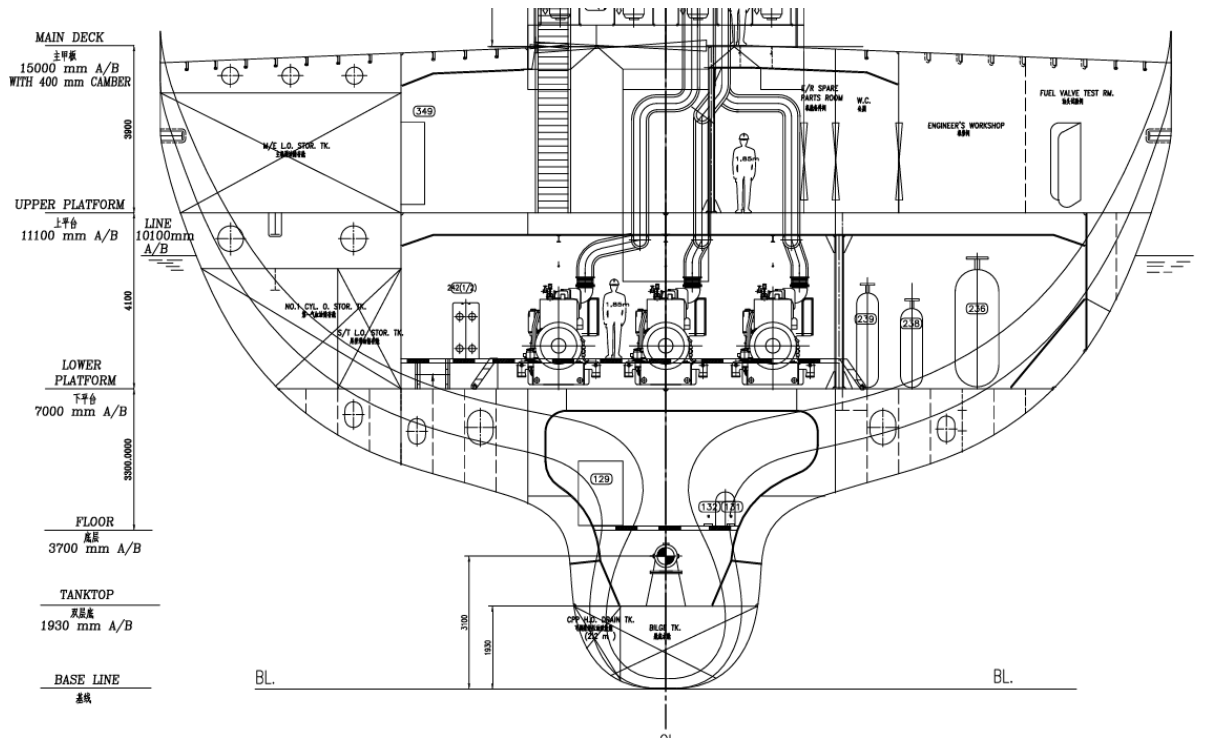


Figure 5.9 Position of the generators on the board (transverse view on deck 7000)

Wet mass and main geometrical dimensions for installed generators are shown in Table 5.10.

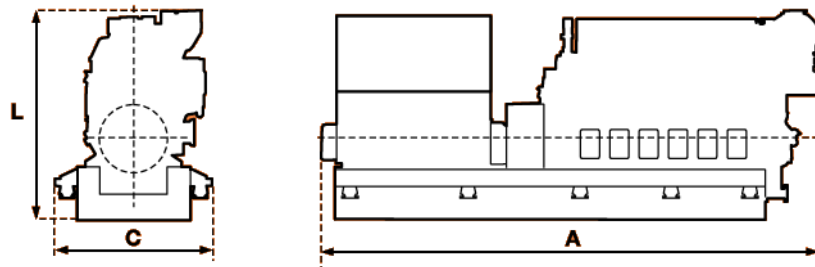


Figure 5.10 Main dimensions of the Auxpac generator.

Type	Wet weight (t)	A (mm)	C (mm)	L(mm)
645W4L20	14.7	4537	1920	2248
875W6L20	17.9	5062	1920	2248

Table 5.3 Main parameters of the Auxpac generators

It is easy to notice that generators have significant mass characteristics and their total mass is 50.5 tones. Of course their presence on the board of the ship influence on the vibration process excited by main engine. So it is necessary to incorporate models of the generators in whole model of hull to have more realistic results. Actually as the generators consist of 4-stroke marine diesel engines it also excites some high frequency vibration in the structures but this process is not simulated and only additional inertial-mass characteristics of 3 electrical generating sets is modeled.

As stiffness of the electrical generators is non-important in that type of numerical simulation a box shape was used to represent 3-items electrical power set.

Knowing main dimension parameters from Table 1 and proper generators positions from Fig. 1 and Fig. 2 three boxes were created in ANSYS Preprocessor. As already was mentioned that the main influence on the modeled vibration process is made by inertial-mass characteristic of electrical machinery. It was difficult to find proper information about center of mass and axis inertial parameters for such complex structure so main attention was directed on the mass characteristic. As a volume of each generator is already known it is necessary only to apply *proper density values* for materials of the boxes.

Other crucial task is to mesh these three volumes and to provide exact coincidence of the nodes in place of the contact machinery with deck 7000. For that task a special macros in APDL was coded. After that nodes were merged to model rigid connection between generators and foundation (deck 7000). Probably such connection is not realistic, but this representation is enough for the numerical simulation which will be performed.

Type of used element for meshing was SOLID 185. Such element is used for 3-D modeling of solid structures. It is defined by eight nodes having three degrees of freedom at each node: translations in the nodal x, y, and z direction. Only linear models of material were utilized with Young's modulus equals to 2.06E+011 (the same as for steel).

Material number for 645W4L20 is 1001 and density is 748 kg/m³

Material number for 875W6L20 is 1002 and density is 810.5 kg/m³

Fig. 5.11 shows the geometrical model of generators set on the board.

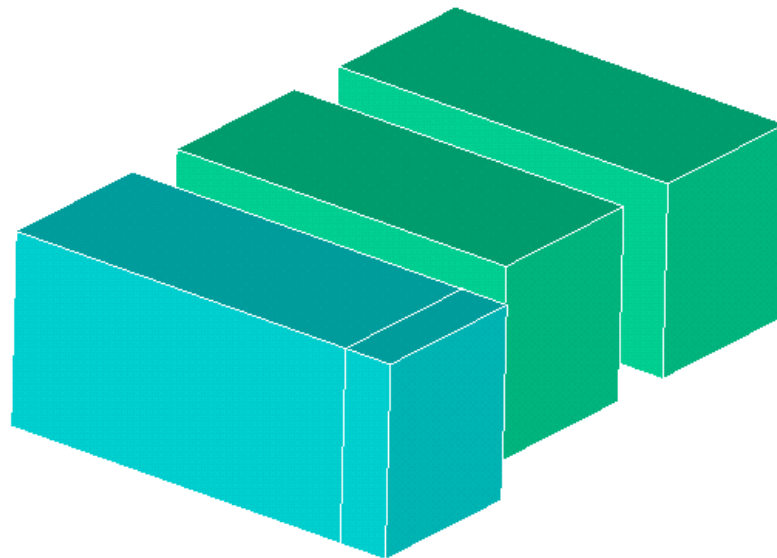


Figure 5.11 Geometrical model of the generators set

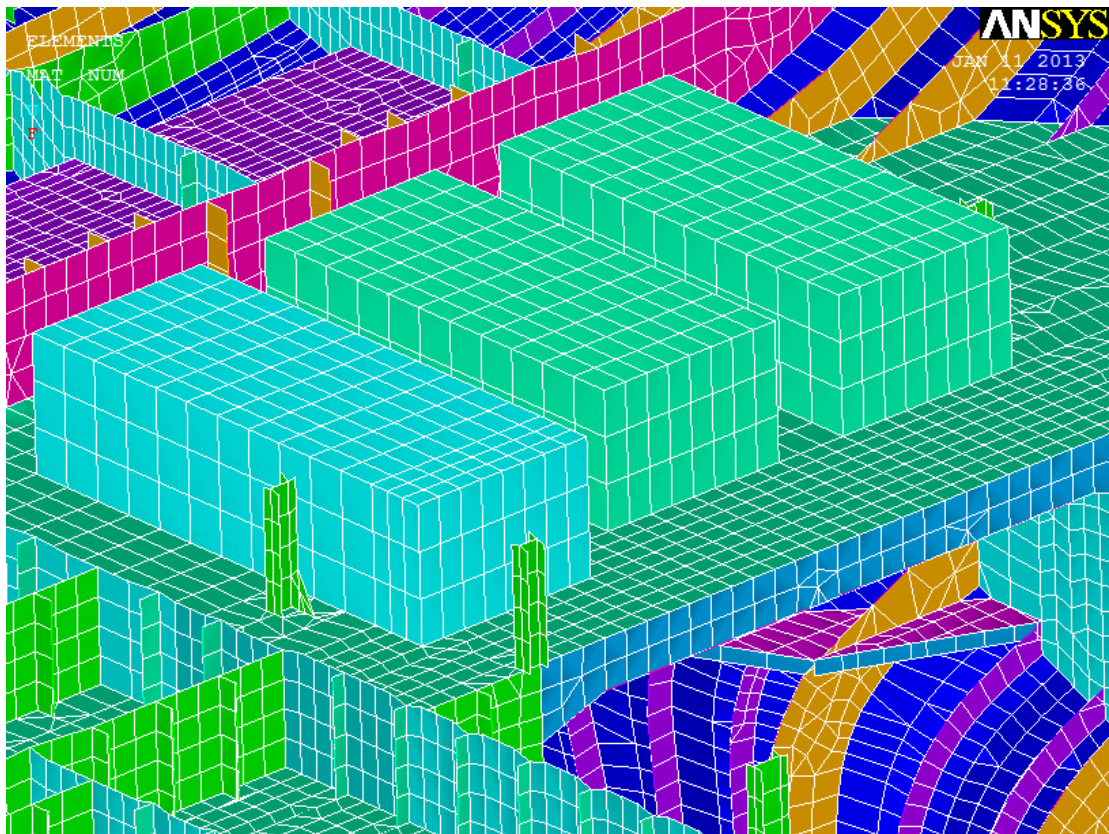


Figure 5.12 FE model of the electrical generators

Fig. 5.12 presents finite-element model of the electrical generators on board. Different colors of the elements mean different material number of structures. It is possible to see that all nodes in the contact region are merged. All generators elements are box-shape.

5.4 Turbocharging system

Current main diesel engine Wartsila RTA48T-B is equipped with one turbocharger TPL73B12 produced by ABB company (see Fig. 5.13).

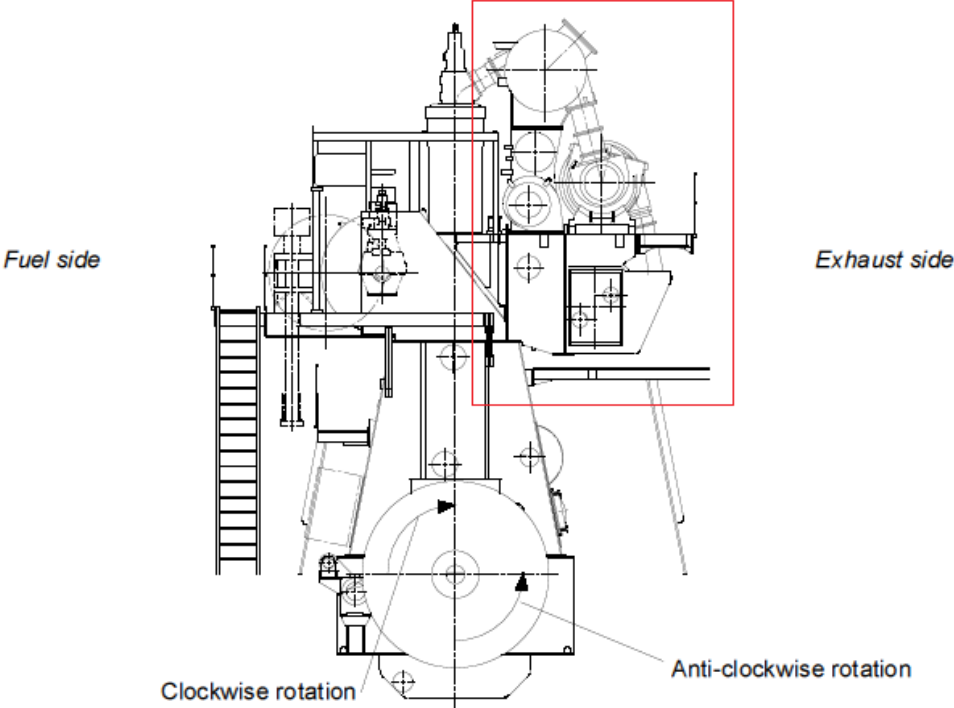


Figure 5.13 Position of the turbocharging system

Increasing of engine power per liter is called forcing. One of the common ways is average effective pressure forcing. Bigger pressure creates due to input into the cycle more heat. But more fuel needs more oxidizing compound for full combustion. Therefore it is necessary to increase amount of fresh air charge which is pumped into the cylinder. Such process has name an engine boost.

The most popular is gas turbine charging or *turbocharging* where energy of exhaust gases is used. Principal diagram is shown on Fig.5.14.

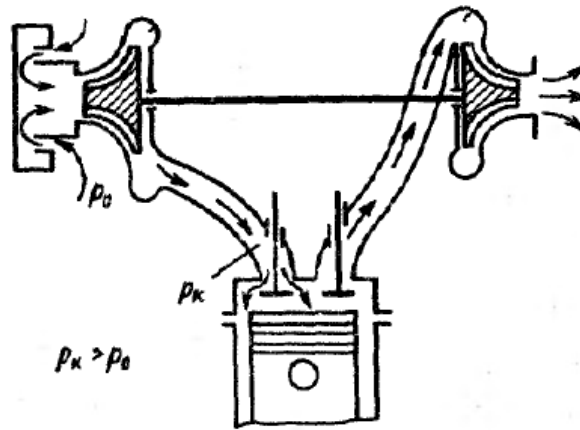


Figure 5.14 Principal diagram of the turbocharger

As exhaust gases have small energy under low power there is not enough fresh air charge in the beginning of acceleration process. To solve this problem two electrical driven auxiliary blowers are used.

After compressor air temperature is about 80° - 180° C. Air cooling is used to increase mass cylinder filling and power goes up. Therefore scavenge air cooler is an integral part of the turbocharging system.

Turbochargers, SAC, receiver, auxiliary blowers were modeled as one simplified sub-assembly in ANSYS Preprocessor. Weight of this construction is 20 tones and knowing approximate geometrical dimensions from Installation Manual it is also possible to calculate density for the material model.

Fig. 5.15 and Fig. 5.16 show geometrical and finite-element models respectively.

Turbo charging system is located on the platform which is attached to the engine column.

Node connection also is provided for correct simulation. Attendance of such massive structure on the one side of the engine moves Y-coordinate of the center mass from the center line, but this question will be discussed below.

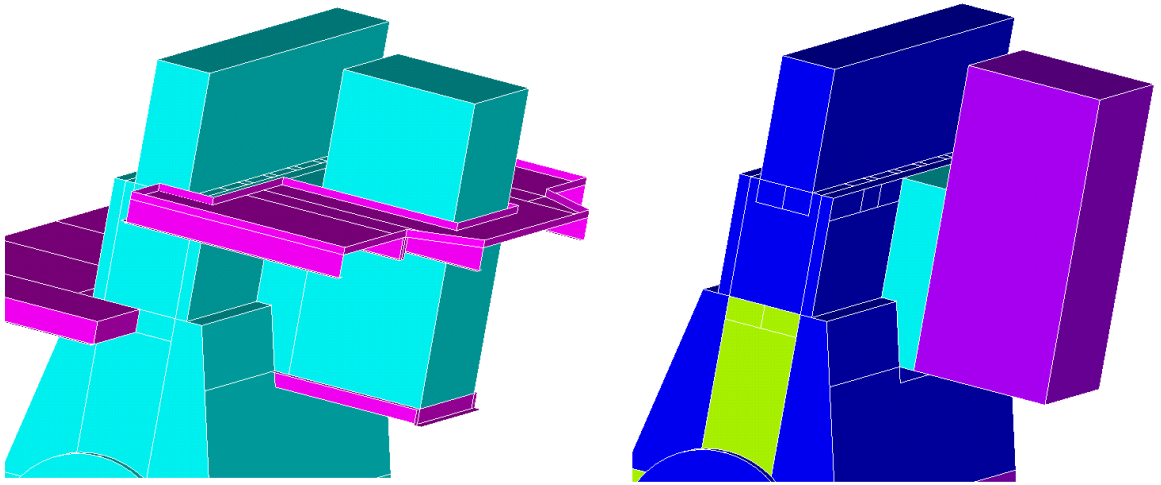


Figure 5.15 Geometrical model of the turbocharging system

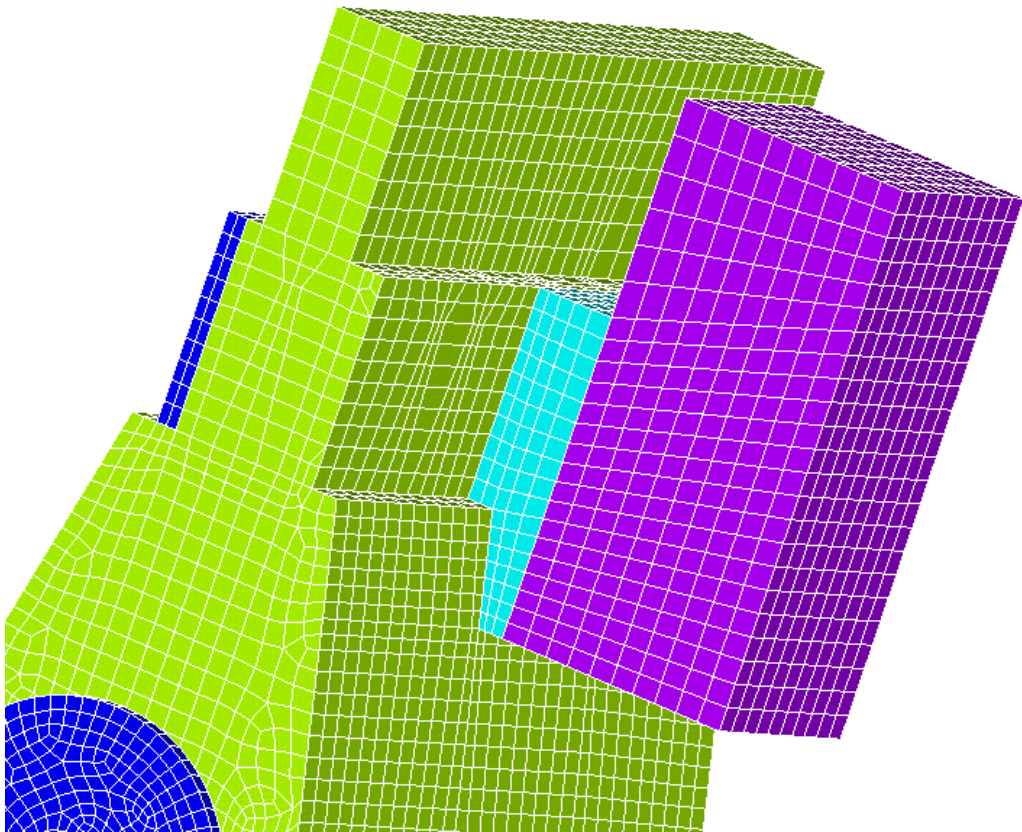


Figure 5.1610 F-E model of the turbocharging system

5.5 Shaft line

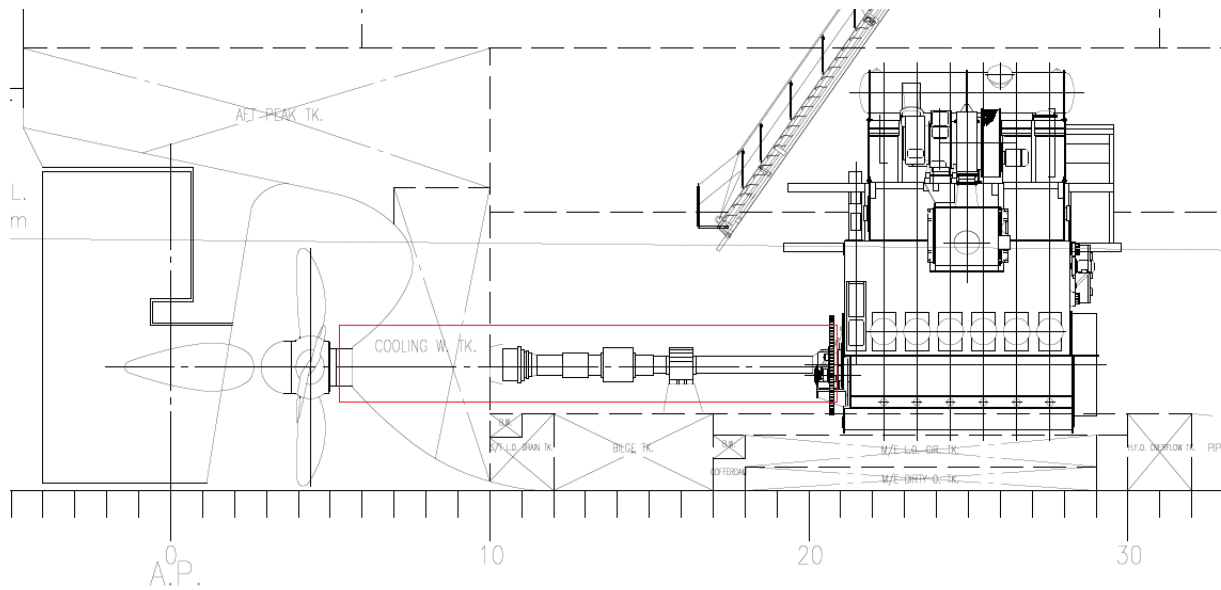


Figure 5.17 Location of the shaft line

There is the shaft line on the board. It connects slow-running main diesel engine driving end with controllable pitch propeller (4 blades, diameter is 5900 mm.). Scheme of shaft location with position of plumber block is presented on Fig. 5.17.

The finite-element model of ship aft includes simplified shaft line representation (long solid cylinder with constant diameter without propeller). It is made as a direct continuation of simplified crankshaft and it is attached to the fly wheel. As thrust bearing is integrated in engine's bedplate so there is only one intermediate shaft support bearing which is also presented in the numerical model. Details are shown on Fig. 5.18 and Fig. 5.19.

In spite of the fact that sometimes a shaft line can transmit vibration due to thrust fluctuation of the propeller to the engine through thrust bearing, such excitation force sources are out of this research frame.

Therefore shaft line is modelled as static solid structure which creates additional connection between double bottom, sterntube and the main engine thereby increasing stiffness of engine/foundation system.

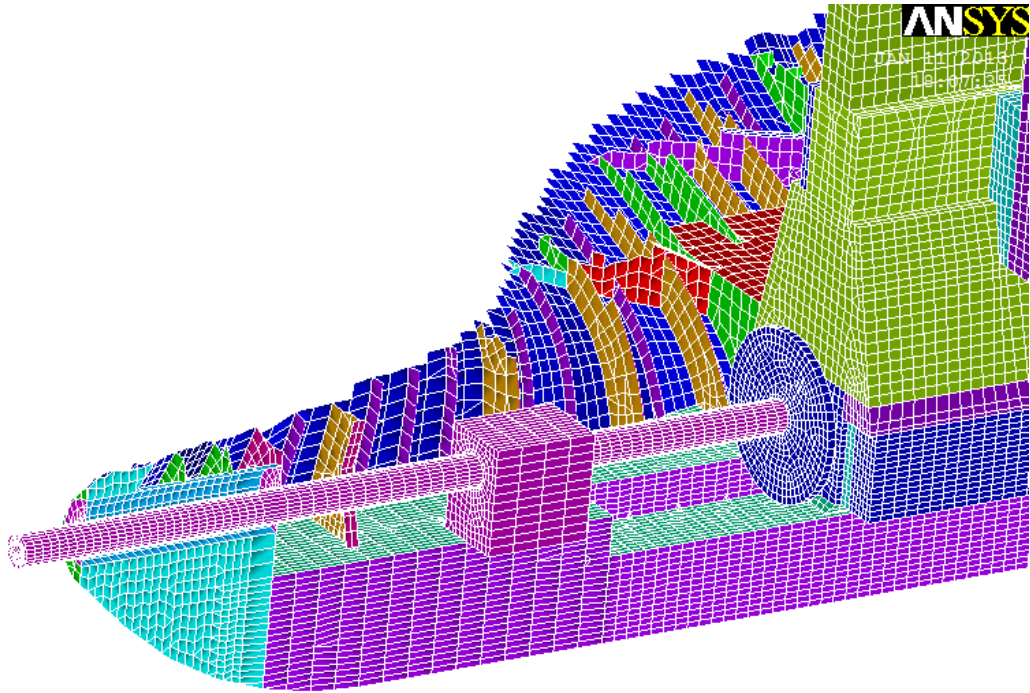


Figure 5.18 FE simplified model of the shaft line

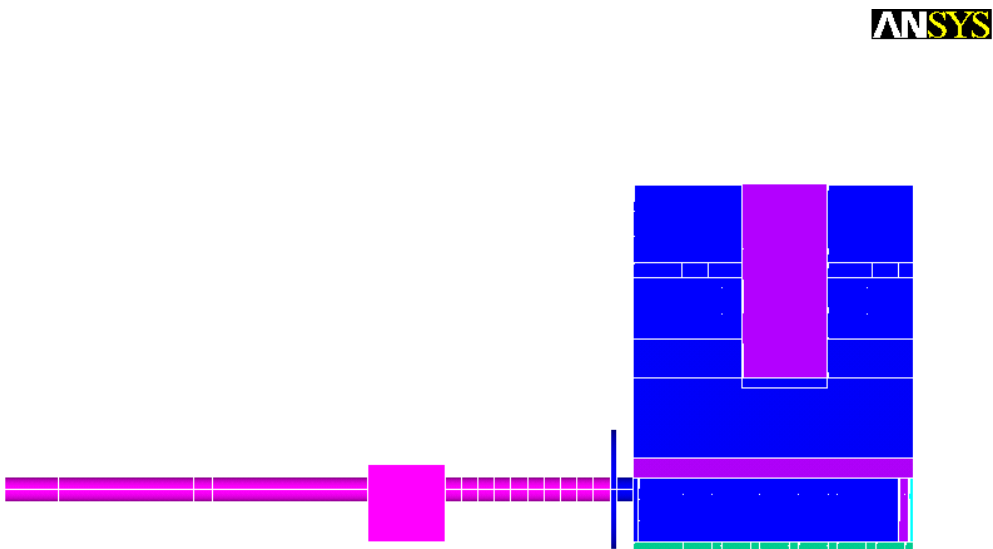


Figure 5.19 Side view of the main engine with the shaft line

5.6 Superstructure (deckhouse)

The superstructure is located on the aft part of ship straight above the engine rooms. This is very typical for modern bulk carriers. There are living rooms, stores, radars etc. Such architecture solution allows increasing load capacity due to bigger holds (also there is a better trim condition). Also a speed of loading and unloading operation goes up due to free space above the hold hatches.

Of course there are several disadvantages. One of them is worse ship handling but this is easily solved by means of using video cameras. Another problem linked with close location of the engine room and propeller which are main excitation sources of vibrations.

As deckhouse has rather large weight so it can influence studied vibration process. Mass of superstructure equals to 350 tones (obtained from statistical data in the company). Mass distribution on the main deck is considered uniform. As only transverse modes are modeled (there is no big distortion of the main deck) influence of the deckhouse stiffness is minimal. Rotational inertial of the superstructure is also ignored.

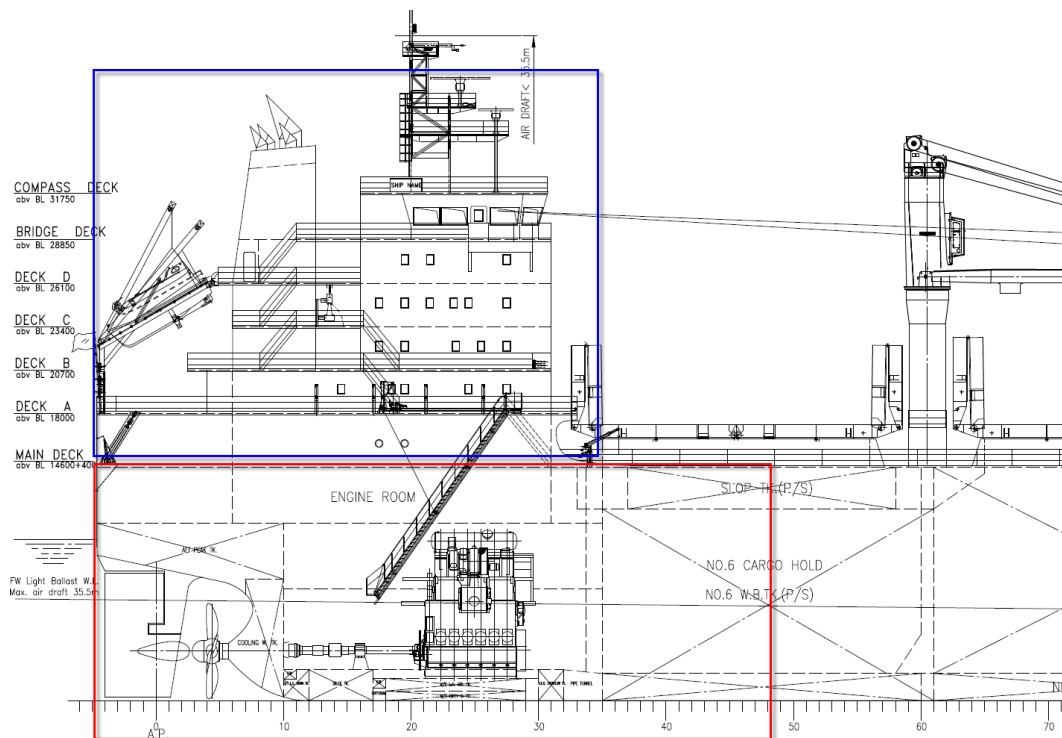


Figure 5.20 Ship parts covered by FEM modelling.

There are two ship parts involved (see Fig.5.20): PART A (red box) subject to detailed FEM modelling and PART B (blue box) incorporated in the model in approximate way to represent only the mass inertial interaction of the hull and superstructure.

Part B is modelled by means of point mass (one-node) elements (MASS 21). There are located on the main deck and connected with the existed nodes. Total number is 4776 and weight of each element is 74 kg. Purple dots on the Fig. 5.21 show such elements which model presence of the superstructure.

Below analysis of superstructure influence on the natural H-type modes of main engine will be shown.

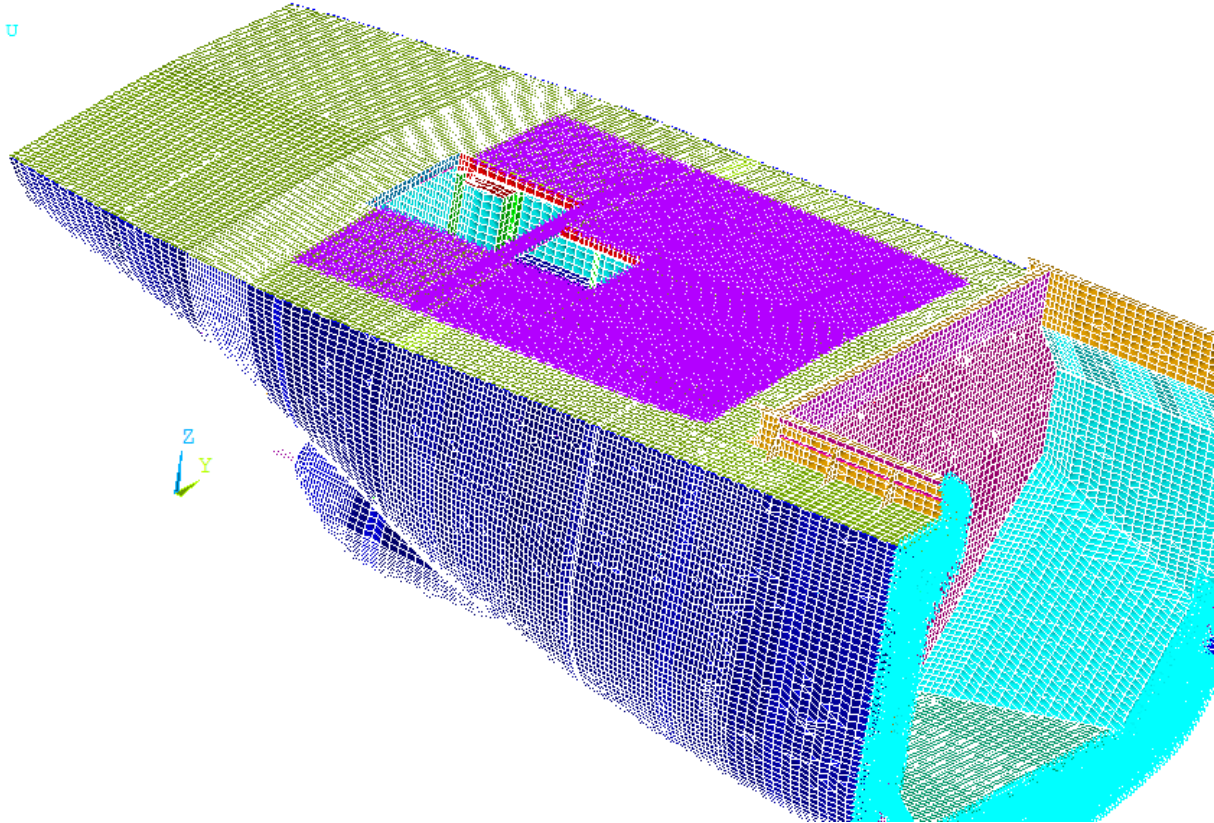


Figure 5.21 Representation of the superstructure

5.7 Hull

As engine structure vibration is studied it is possible to model only aft part of the ship with engine room. Actually, it is difficult to estimate at the beginning the extent of the hull structure really engaged in vibration of the system considered. Therefore, the hull part taken into analysis is a priori intended to be large enough to eliminate the influence of boundary conditions on the analysis result.

Initially the model was built in Poseidon GL software but after the FE model was almost fully modified in ANSYS and all errors were eliminated. The model is to be created in sufficient details to describe properly elastic and inertial coupling between all the components involved.

Roughly speaking, the big hull structure plays a role of the elastic foundation for the engine.

Later comparison between the rigid supported propulsion system and realistic supported propulsion system will be done and influence of the hull on the vibration process will be studied.

There are several tanks in the aft part: cooling water tank, bilge tanks, aft peak tank, etc. Their fullness (inertial characteristic) probably may influence the vibration process and it is possible to take into account by defying such densities of tanks materials to have the same weights as full tanks. Nevertheless empty tank condition is studied in that research.

Hydrodynamic added masses are very important and interesting question. Studied H-type vibration leads to torsional, vertical and horizontal vibration of the doublebottom so surround water should influence the process. The role of the added mass is in increasing inertial characteristic of the hull. It is possible to conduct calculation with inclusive of fluid-structure interaction as it is done for global or local vibration but it is very time consuming and the influence in case of engine system vibration may be not crucial.

Moreover if added masses are included in the calculation several runs are required as there are several drafts for different load conditions.

Therefore hydrodynamic added masses are out off the research topic, but anyway it is possible and necessary to study surround water influence additionally.

Shell and beam elements were used to create a finite-element model of the hull. Shell elements describe hull plating, deck plating and webs of frames, girders and stiffeners. Beam elements model flanges of frames and stiffeners. Such approach is very common in ship modeling.

As beam element is used BEAM4. BEAM4 is a uniaxial element with tension, compression, torsion, and bending capabilities. The element has six degrees of freedom at each node: translations in the nodal x, y, and z directions and rotations about the nodal x, y, and z axes.

Details of structure idealization are presented on model views shown in Figures below.

1

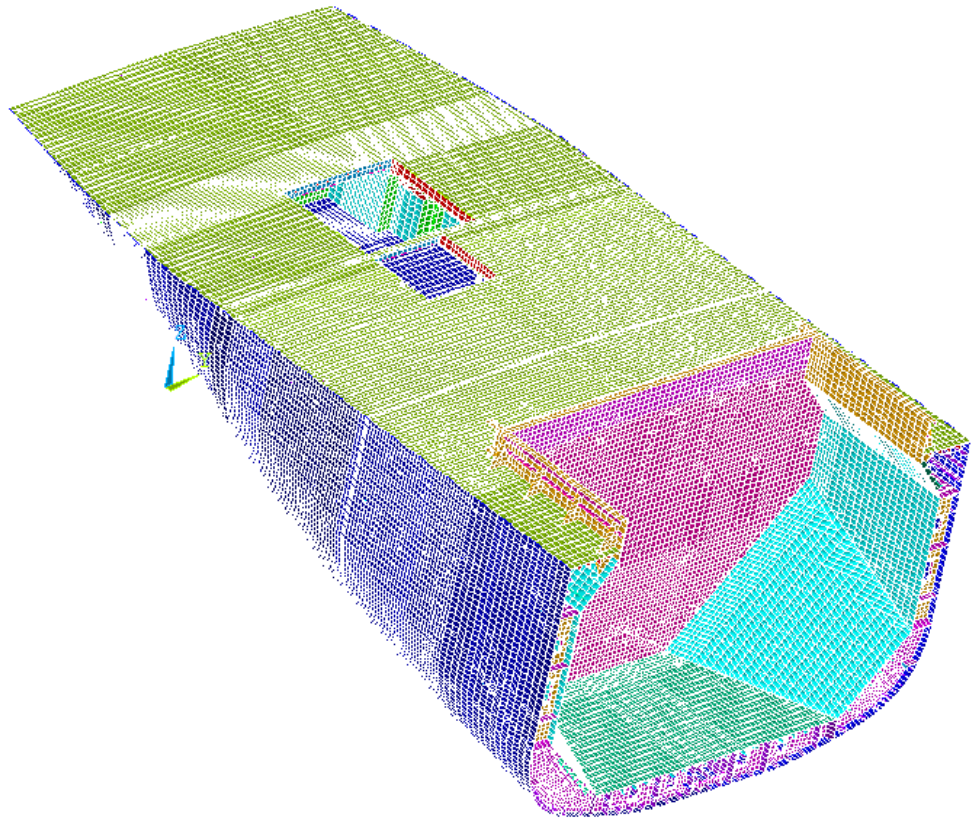


Figure 5.22 FE model – general view

The following coordinate system is defined in the model:

(X, Y, Z) – Cartesian coordinate system with axes parallel to axes of global ship coordinate system.

X – axis – positive in the bow direction,

Y – axis – positive in the port side transverse direction,

Z – axis – positive in the vertical upward direction.

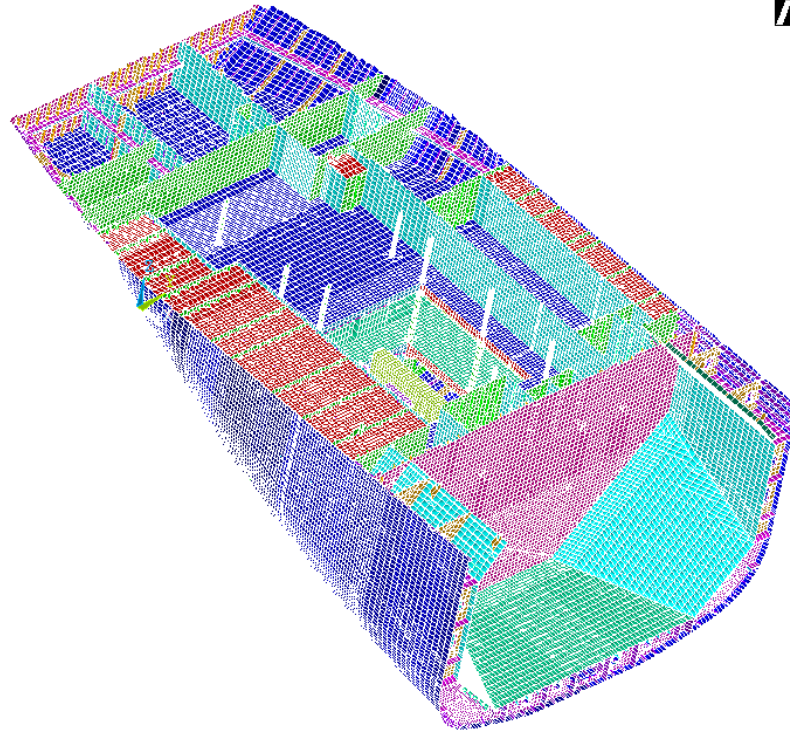


Figure 5.23 FE model – general view under Main Deck

1

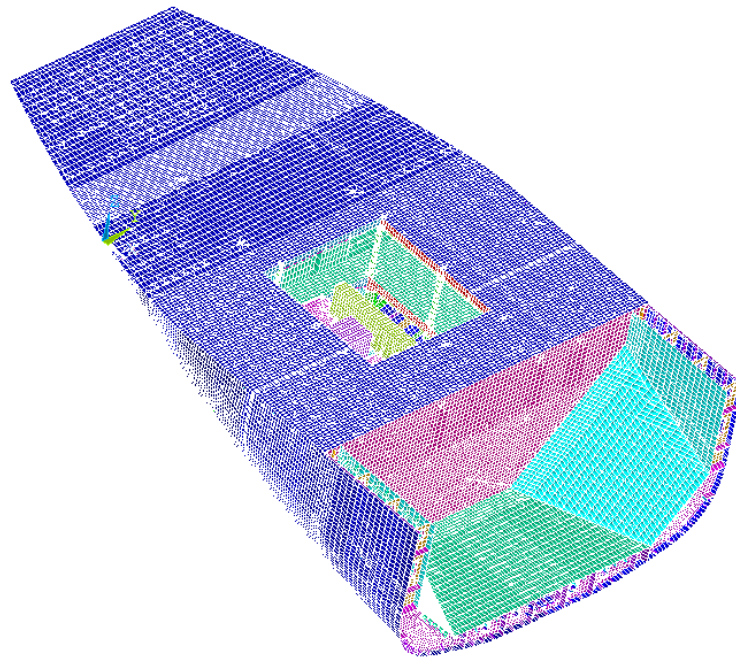


Figure 5.24 FE model – general view on Platform 11100

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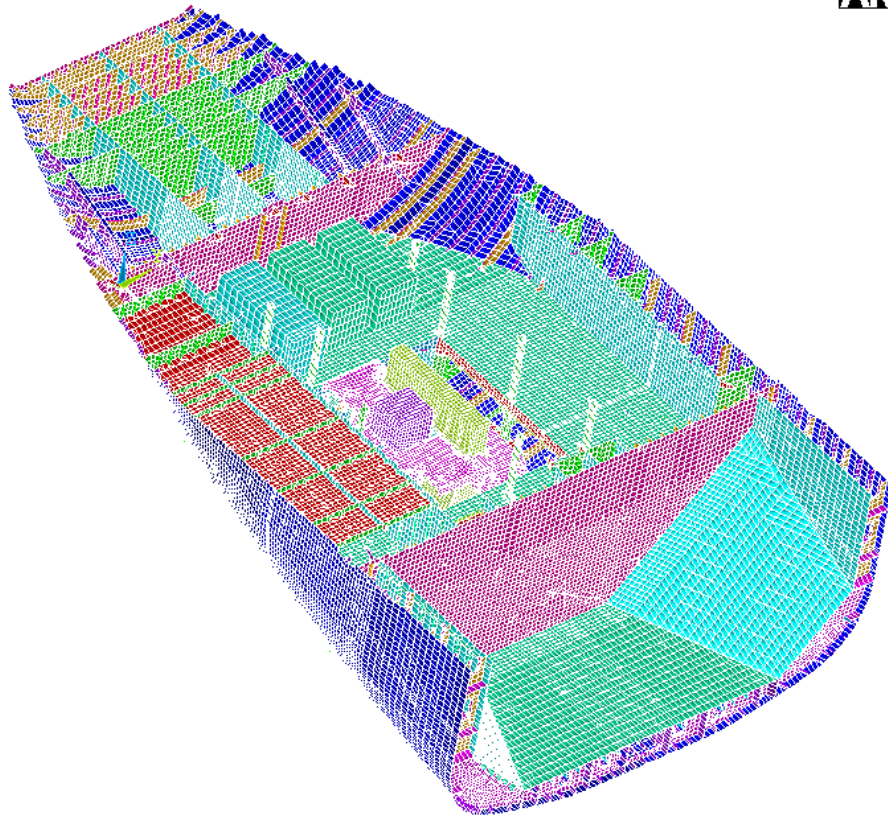


Figure 5.2511 FE model – general view under Platform 11100

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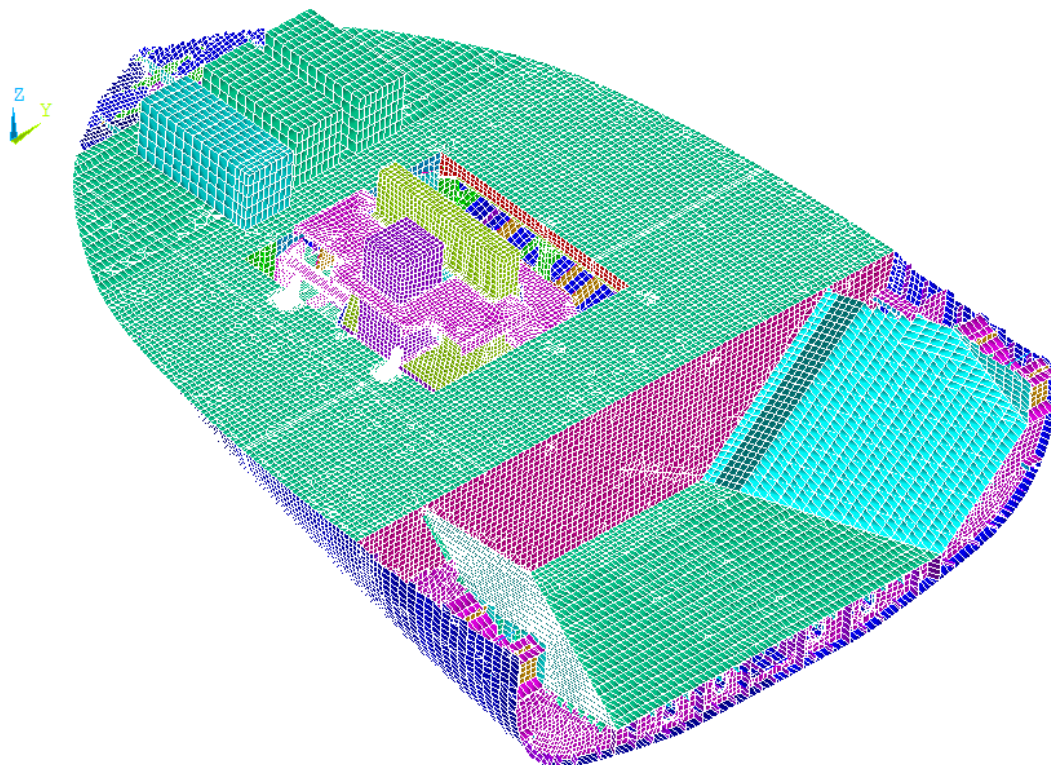


Figure 5.26 FE model – general view on Platform 7000

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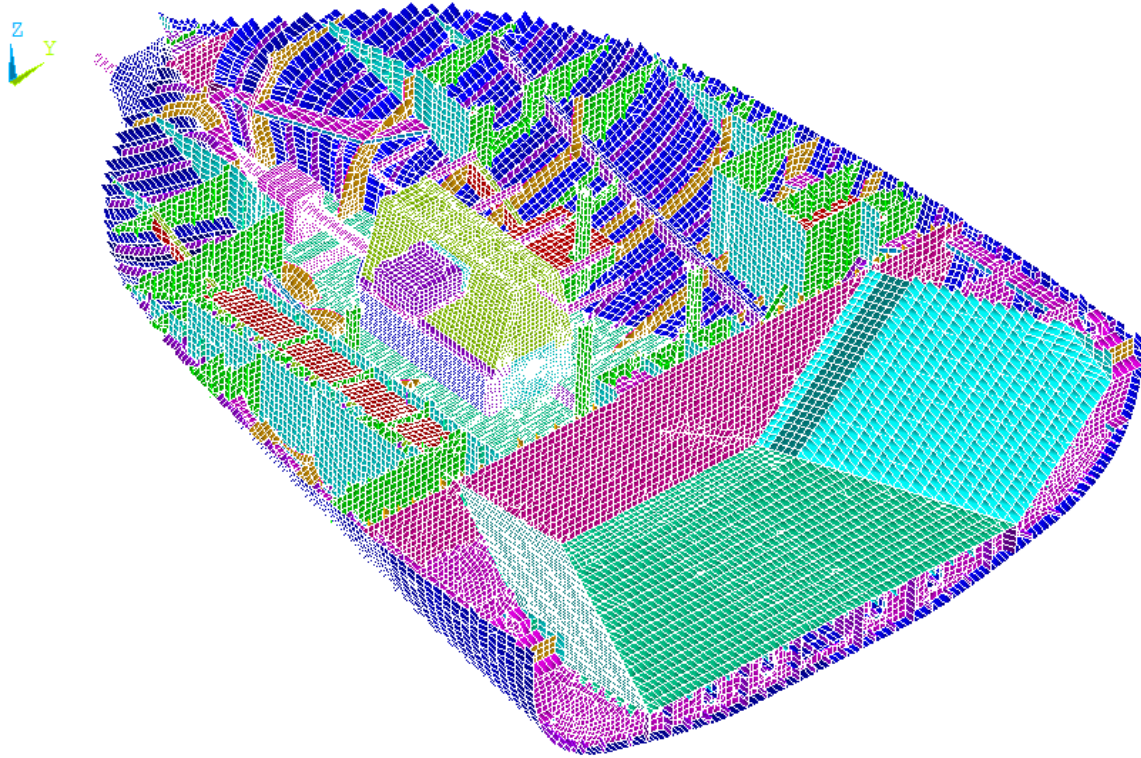


Figure 5.27 FE model – general view under Platform 7000

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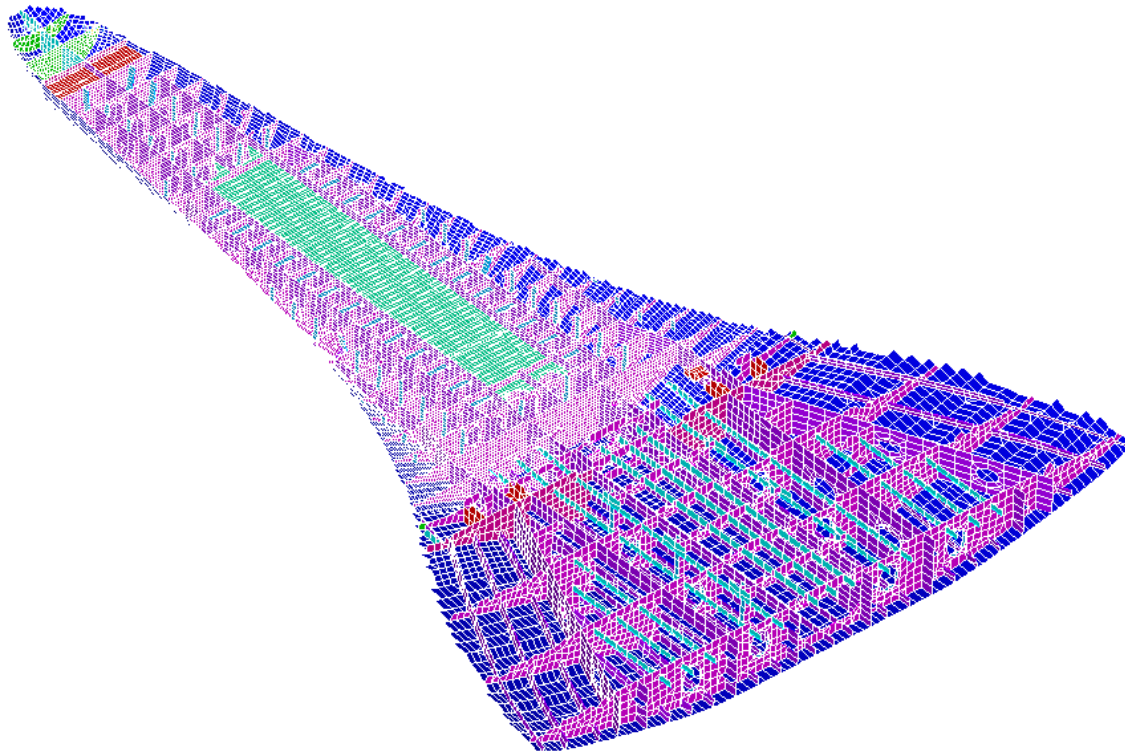


Figure 5.28 FE model – general view on Double Bottom

6 MODAL ANALYSIS

6.1 Constrains

At all node adjacent to model boundary at Fame 45, translation degree of freedoms have been fixed ($dx=0, dy=0, dz=0$)

This constrains, in some way, represents the “cut” forward part of the ship.

Figure 6.1 shows positions of constrained nodes (blue color)

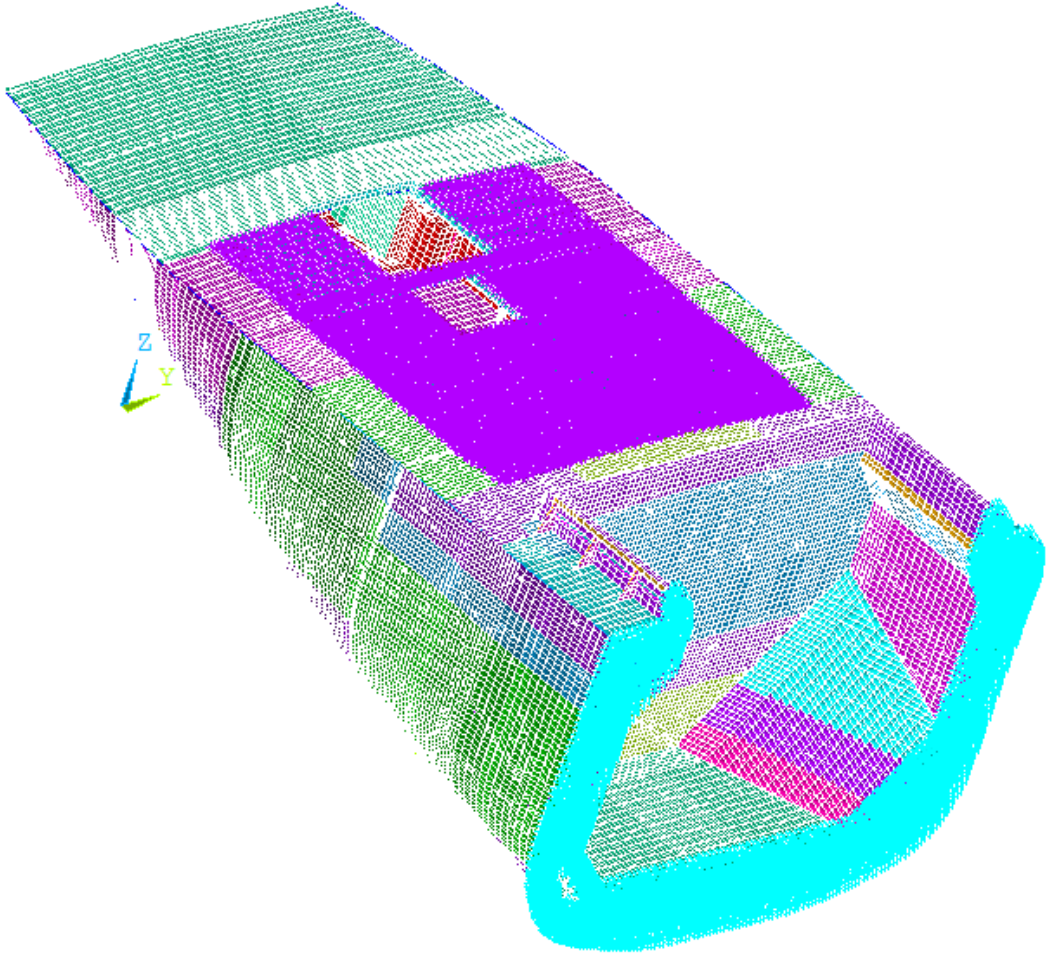


Figure 6.1 Boundary conditions

The same boundary conditions are used in harmonic analysis also. As free vibration analysis is performed so there are not any loads.

6.2 Modal analysis of the hull with embedded engine

The company has given information that the H-type natural vibration frequency of the mounded engine should be in the diapason between 6 and 7 Hz. Using an interactive procedure the necessary combination of materials stiffnesses was obtained. Extracted natural frequencies and corresponding modes are shown on Figure below. For better visualization only part with propulsion system and close surrounding elements will be presented.

The Block Lanczoc eigenvalue extraction method is used.

The range of searching is 4-10 Hz. *Eight natural frequencies* were found.

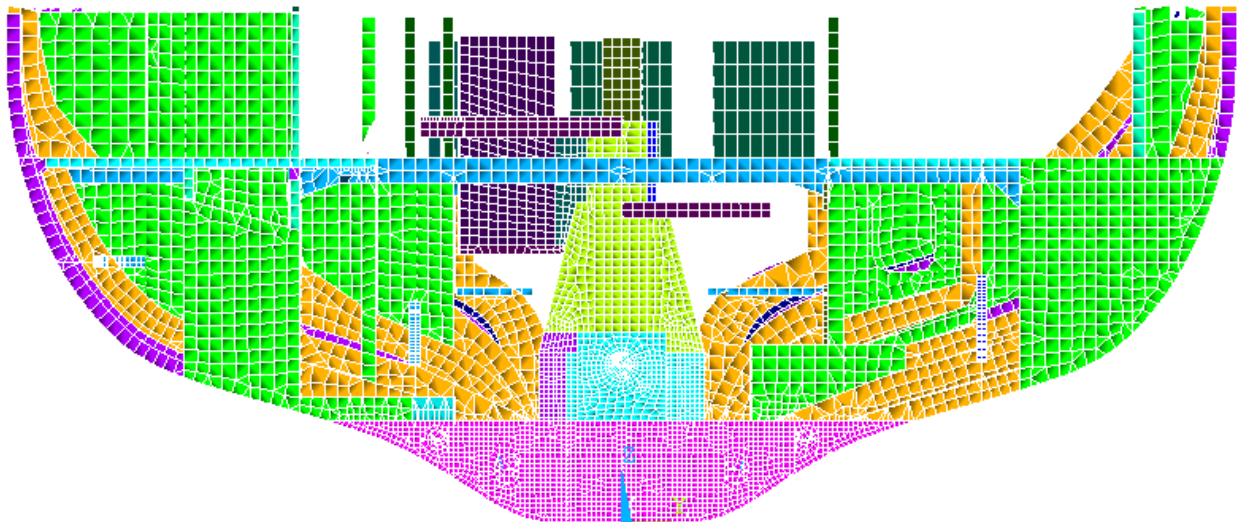


Figure 6.2 Area of interest (cross section view)

First calculated natural frequency is 4.97 Hz. Vertical mode of the engine structure vibration is corresponds to that frequency. (See Fig. 6.5) This mode is defined only by doublebottom stiffness which is rather low. If hydrodynamic added masses were considered in the simulation the natural frequency would be even lower.

As the exciting guide forces are lateral type such mode is out of the interest because direction of forces does not coincidence with direction of free vertical oscillation.

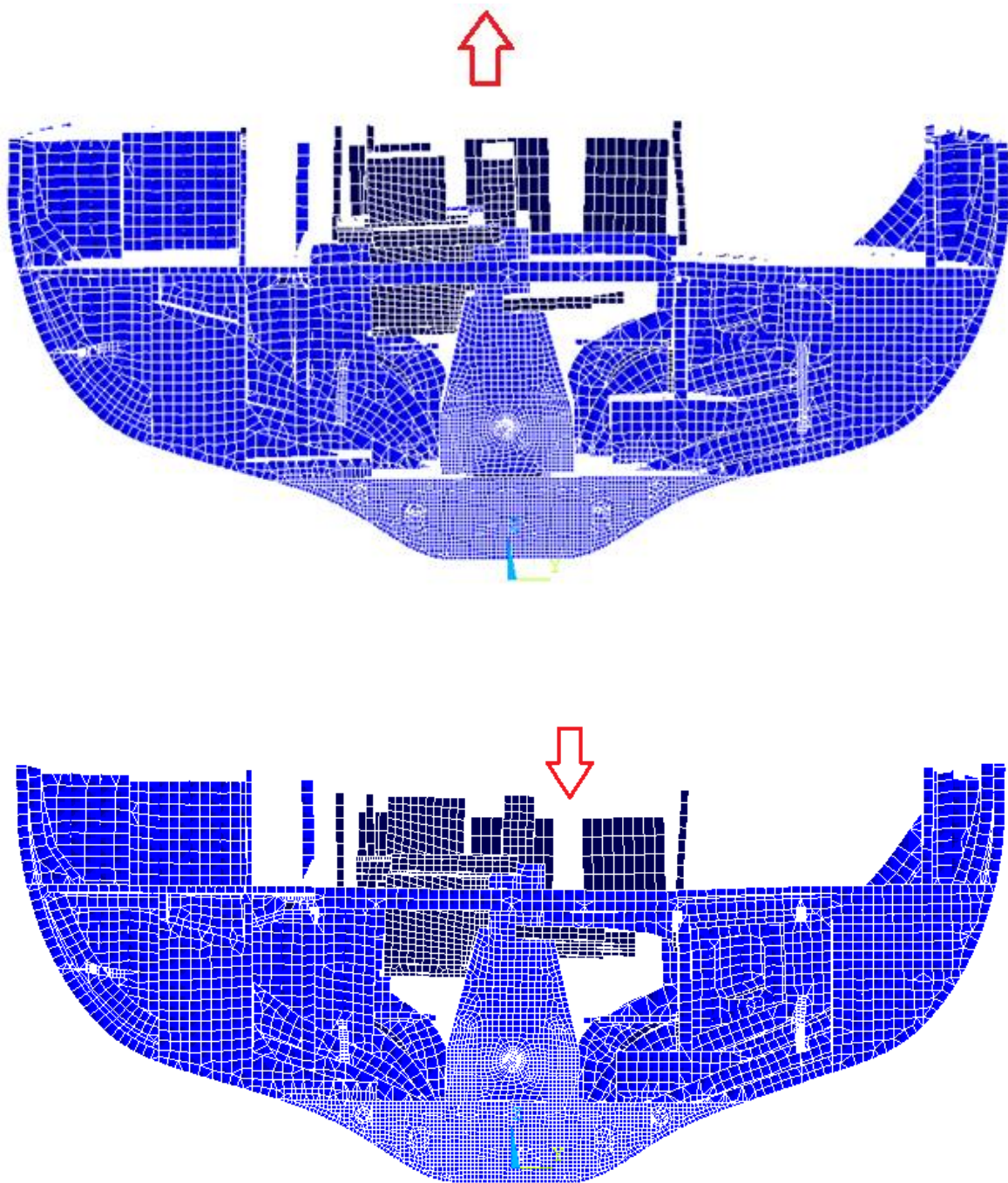


Figure 6.3 Vertical vibration mode

The second natural frequency is 5.895 Hz. Lateral mode of the engine structure vibration corresponds to that frequency (See Fig. 6.4). Such mode can be classified as H-type. It is important to notice that the hull and the engine oscillate in the same phase. Possibility of the resonance when frequency of lateral guide force coincides with this natural value will be investigated in forced vibration simulation (see Part 8).

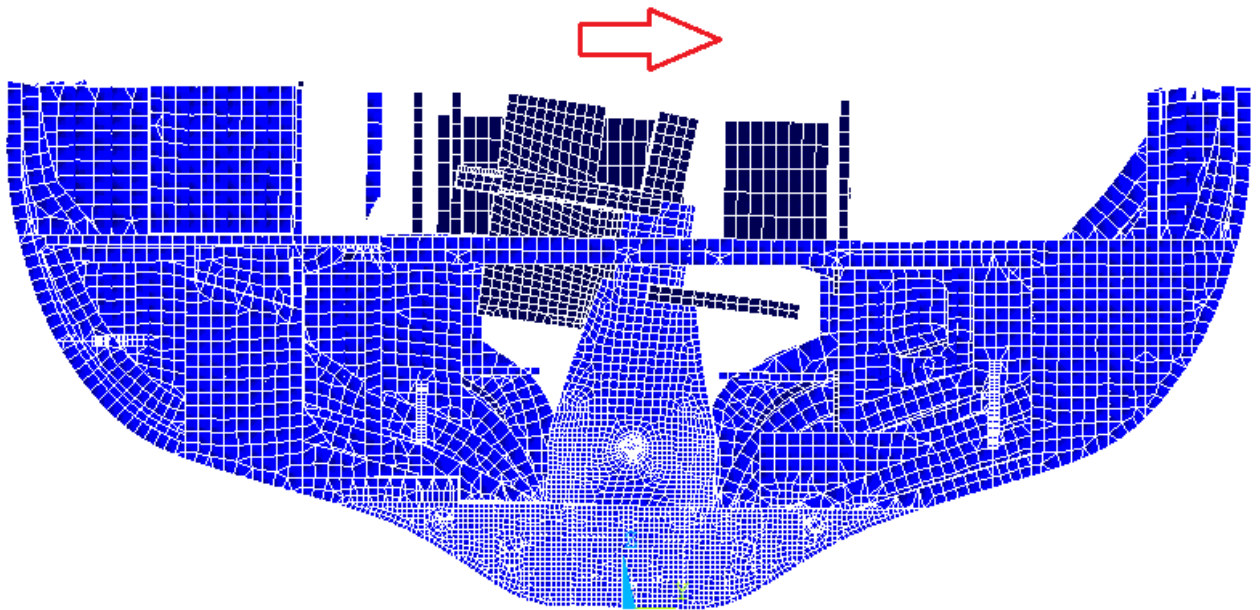
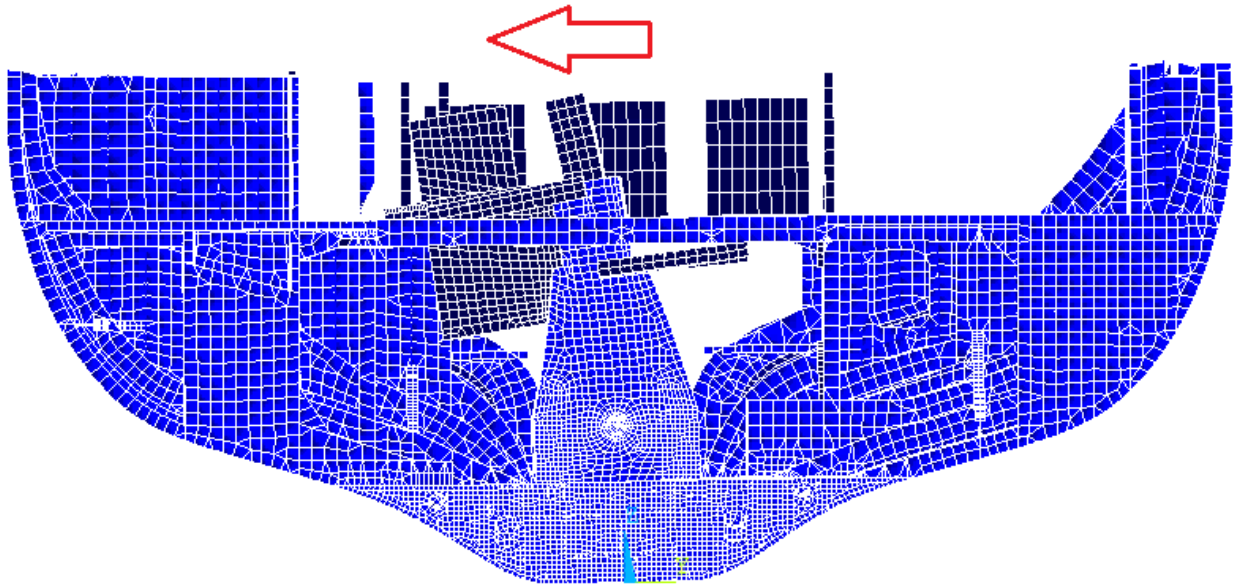


Figure 6.4 First lateral vibration mode (5.895 Hz)

The third natural frequency is 6.53 Hz. Vertical mode of the engine structure vibration is corresponds to that frequency. As already was mentioned, verticals modes are out of interest in that research.

Next natural frequency is 6.92 Hz. Again, lateral mode of the engine structure vibration is corresponds to that frequency. This mode is very close to the mode with frequency 5.895 but still has significant distinction. The hull structure and the engine structure oscillate almost in opposite phases (see Fig. 6.4). Such condition is perfect to install dampers as opposite displacements may counterbalance each other. Resonance at that frequency will be also studied in forced vibration analysis in Part 8.

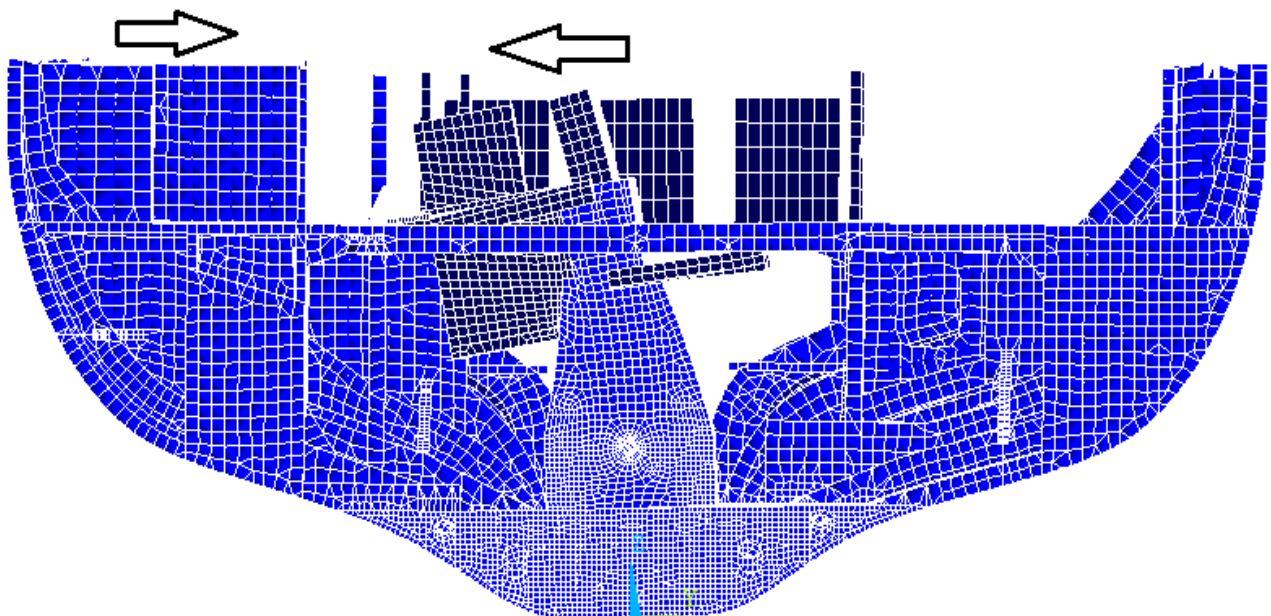


Figure 6.4 Second lateral vibration mode (6.92 Hz)

Lateral mode vibration also excites at frequency 7.15 Hz. Mode shape is very close to the previous one (see Fig. 6.4). The hull and the engine vibrate in anti phases. The main difference is that vibration affects upper part of the engine block. Behavior of this mode under forced condition (lateral guide forces) should show whether this mode real H-type or not.

After five natural frequencies empty space in 2 Hz was observed. Next calculated frequency is 9.3 Hz and so-called X-type vibration mode corresponds to it.

Two last frequencies 9.522 and 9.525 are responsible for so called L-type vibration (see Fig. 6.5). X-type and L-type modes are hardly affected by lateral exciting forces.

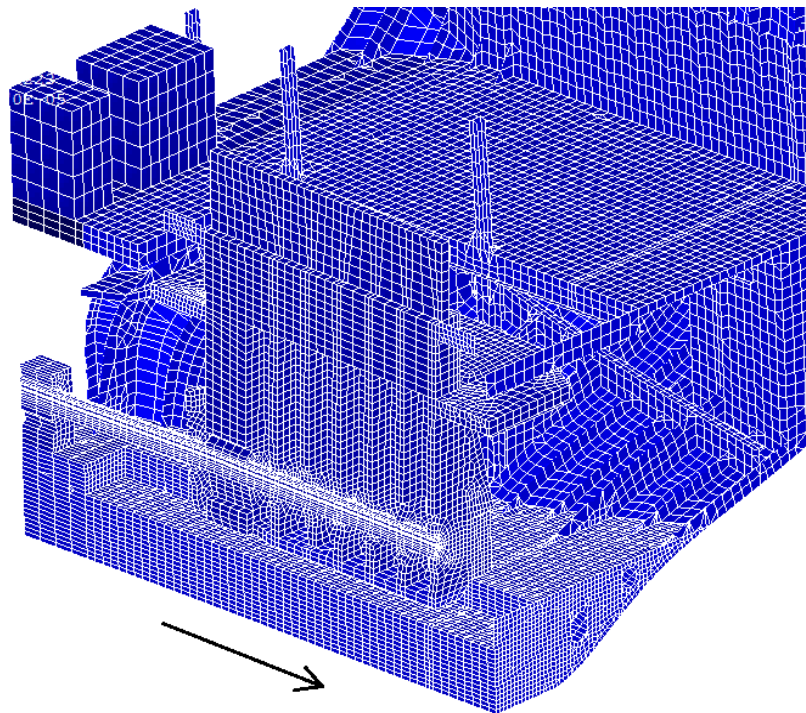
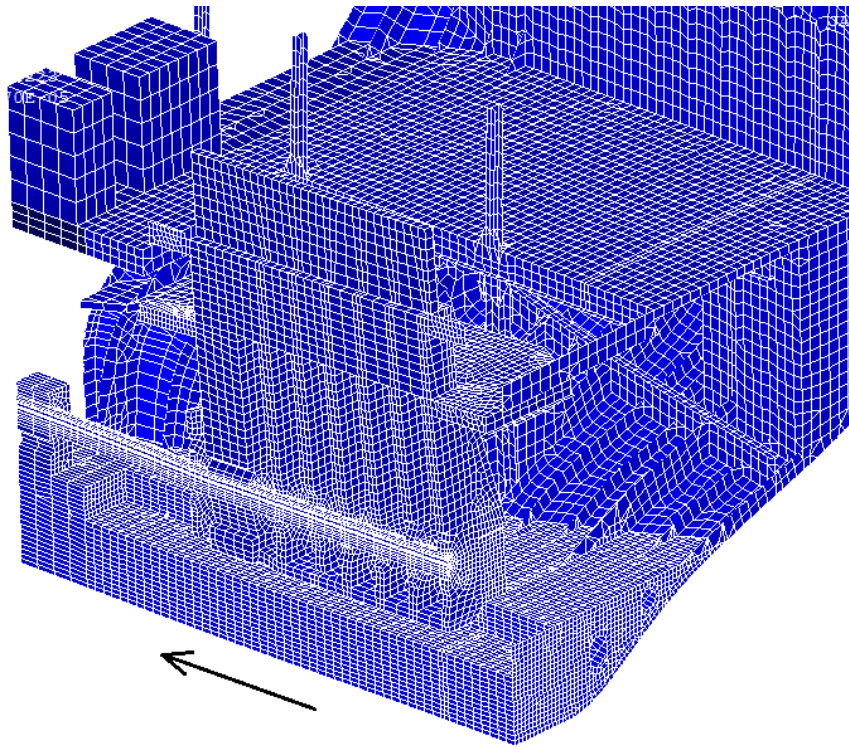


Figure 6.5 L-type mode

6.3 Modal analysis of the engine structure on the rigid foundation

In case if approximate range of the natural frequencies of mounted engine is unknown it is useful initially to perform a modal analysis of the engine which is fixed rigidly (for example engine mounted on big and stiff foundation). Engines fixed on some rigid foundation always have larger values of natural frequencies than ones which are attached to the elastic foundation (for example ship hull). Such initial simulation allows reducing searching filed and saving significant amount of time.

Moreover a performing of both simulations allows identifying the influence which makes hull presence (frequencies shift). It can be useful in case of a creating a simplified mass-spring-dampers models which can be used for nonlinear analysis.

Boundary conditions for rigidly supported engine are shown in Figure 6.6. Nodes with blue color mean that 3 translation degrees of freedom are constrained.

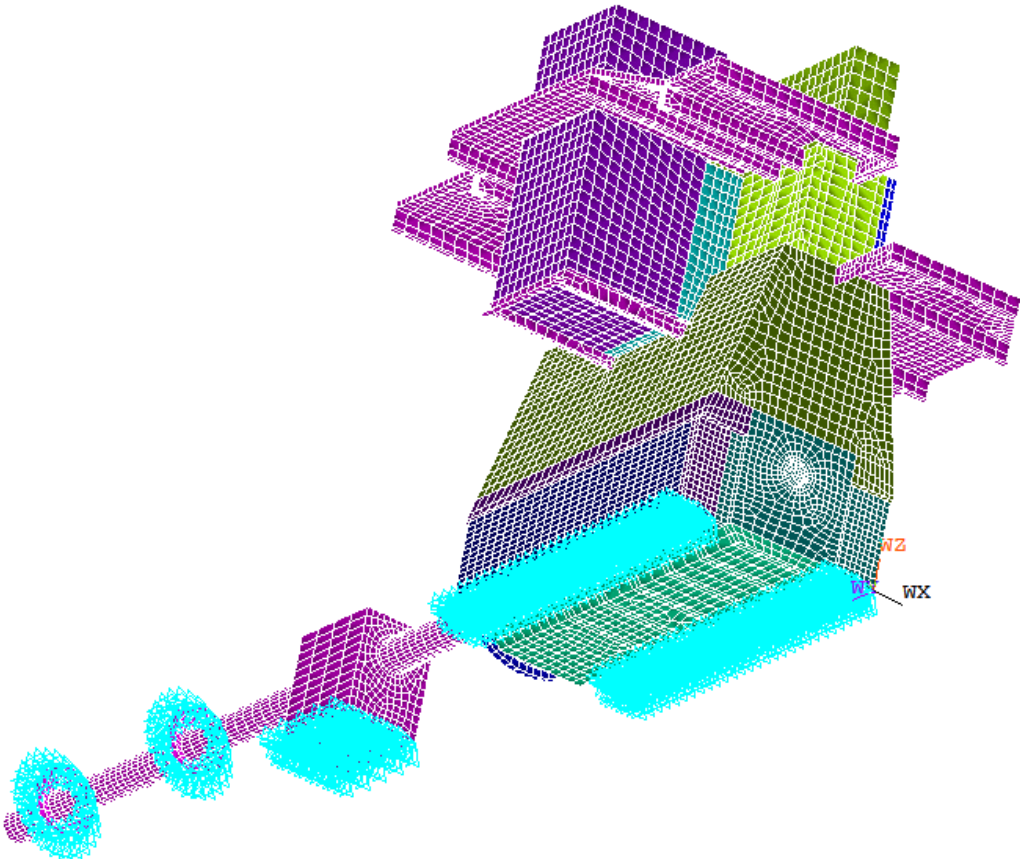


Figure 6.6 Boundary conditions for rigidly supported engine

Two out of the three main natural vibration modes are shown in Figures below.

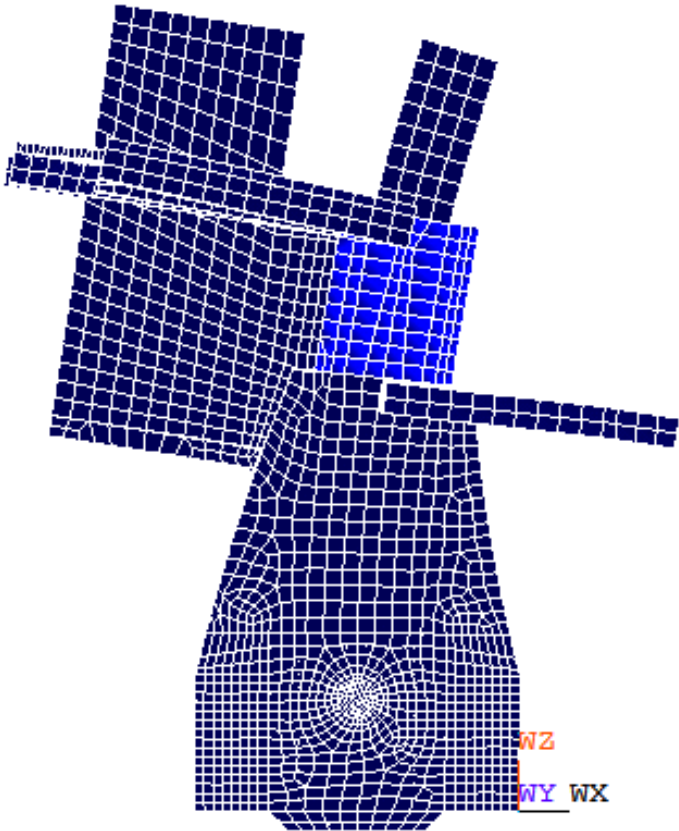


Figure 6.7 H-type mode. Natural frequency is 8.55 Hz

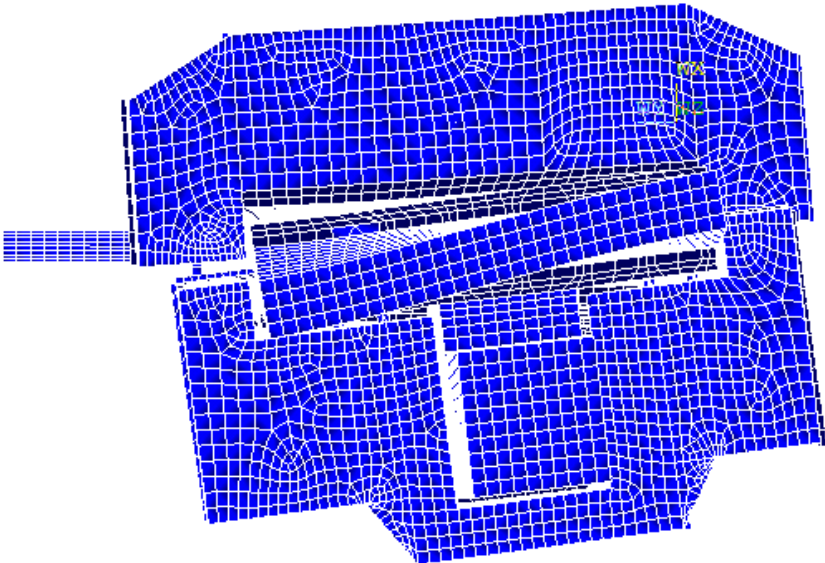


Figure 6.8 X-type mode. Natural frequency is 12.95 Hz

As was predicted the H-type natural frequency of rigidly supported engine has appeared higher than engine mounted on the hull housing (8.55 Hz against 6.92 Hz).

6.4 Modal analysis of the rigid engine on the elastic foundation.

To model stiffness characteristic of an engine properly is very complicated task. There is two ways. First one is to make very accurate and detailed geometrical models with all necessary material models. The second one is to create a simplified model and varying stiffness of the structural materials achieve desired results (used above) but for this case it is necessary to know experimental results or information from the manufacturer.

In this light it is interesting to know differences in natural frequencies when an engine is absolutely rigid and when it is modeled realistically. If results are close it means that we can avoid hard step of stiffness modeling and provide only adequate inertial/mass properties (including the position of gravity center). As the engine is rigid, consequently only stiffness of the doublebottom structure plays role.

The same structure as in Part 6.2 is investigated and the sane solver is used. Young's modulus for all material models used in the engine are set to $1E+013$ Pa. These values are big enough to consider the engine as rigid structure.

All modes except for H-type are ignored and not described here.

First frequency of H-type equals to 7.12 Hz. Hull and the engine vibrate in one phase. This mode is an equivalent of the first H-type mode (5.895 Hz) for engine with realistic stiffness.

The region which is more interesting locates above 9 Hz. There are as many as four natural frequencies and corresponding modes may be classified as H-type.

- 1) 9.18 Hz;
- 2) 9.46 Hz;
- 3) 9.52 Hz
- 4) 9.55 Hz

For all these frequencies the hull and the engine oscillate in opposite phases.

H-type mode with natural frequency 9.55 Hz is shown in Figure 6.9.

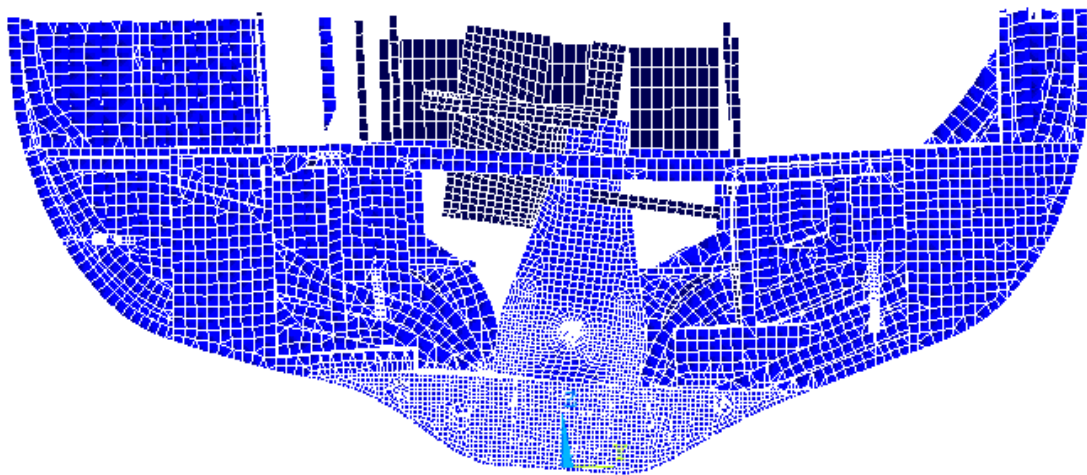
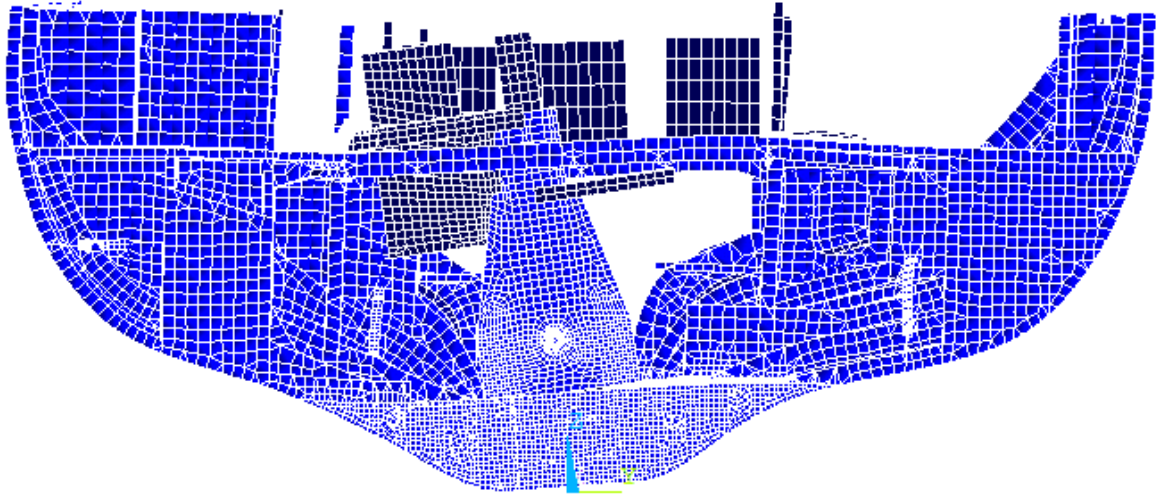


Figure 12H-type mode. Frequency is 9.55 Hz

It is easily to see a big distortion of the hull double bottom and it is logical as only double bottom stiffness defines the natural frequencies of the system.

If we compare natural frequencies of the realistic engine (6.92 Hz) and the absolutely rigid engine (9-9.5 Hz) we will see that frequency shift is (2-2.5 Hz). Therefore using rigid engine structure gives us overestimated results and, unfortunately, an accuracy of such approach is rather low. To obtain more or less valid results the engine structure with real stiffness parameters should be used.

Comparing of these two analysis shows one more time that engine vibration on board is complicated process which involves both engine and hull vibration interaction.

6.6 Influence of the superstructure on the engine natural frequency

It is difficult to obtain proper information about superstructure weight and especially rotation inertia. This is the reason why only mass influence by means of point elements is considered in that research. It is useful to know how big is affect of the superstructure mass on the vibration in the engine room. If it appears small then it will be possible to ignore additional mass distribution on the main deck and it affords us to reduce model.

For that task an analysis without mass element on the main deck was performed. FE model is the same as in Part 6.2 and boundary conditions are not changed.

Extracted natural frequencies of H-type vibrations:

- 6.12 Hz
- 7.1 Hz

Comparing with natural frequencies (5.895 Hz and 6.92 Hz respectively) from Part 6.2 we can see that frequency shift is about 0.2 Hz. Such value is not very big especially comparing with 2-2.5 Hz. Nevertheless as information about weight of superstructure is available model with distribute mass on the main deck is used.

After all analysis it is possible to make a following conclusion. Stiffness of the engine structure plays very important role and it is necessary to reproduce it with big accuracy. Influence of the superstructure is rather small and sometimes can be neglected. Natural frequencies between 5.895 Hz and 6.92 Hz correspond with information from the company.

In next chapter a forced vibration analysis will be revealed.

7 FORCED VIBRATION ANALYSIS

Modal analysis solves free vibration problem (Eq. (7.1)) and gives only mode shapes and corresponding natural frequencies.

$$[m]\ddot{x} + [b]\dot{x} + [c]x = 0 \quad (7.1)$$

If it is necessary to calculate certain response of a mechanical system or it means to obtain amplitudes of displacements, velocities, etc. we need to solve forced vibration problem (Eq. (7.2))

$$[m]\ddot{x} + [b]\dot{x} + [c]x = (F) \quad (7.2)$$

Where (F) is a vector of exciting forces.

It is known that H-type vibration is excited by lateral guide forces and moments (see Part 3). Such forces may be approximated as harmonic forces. Harmonic forces have a following equation:

$$F = F_0 * \sin(\omega t + \varphi)$$

F_0 is an amplitude value;

ω is a frequency of excitation;

φ is a phase.

If in equation (7.2) right side is a harmonic function then problem is linear and can be easily simulate in ANSYS APDL (Analysis type – Harmonic)

Installation Manual provides us information about induced forces. It states that amplitude of the lateral H-moment is 498 KN*m and that moment has six-order frequency. Six-order frequency means that exciting frequency in six times more than shaft rotation. If the speed range of m/s Miedwie is 80-110 rpm then six-order frequency is 8-11 Hz.

A figure 7.1 shows a location of guide forces which create H-type exciting moment. One part is applied to the center line of a crankshaft and the other to the engine block in the mean position of the crosshead.

For the simplicity it is considered that amplitude of the forces is constant for all frequencies.

Response is investigated in the frequency range between 0 Hz and 12 Hz.

Damping is the crucial parameter is forced response analysis. For the simplification, a constant damping coefficient of 1.5 percent of the critical damping is used (ABS recommendation)

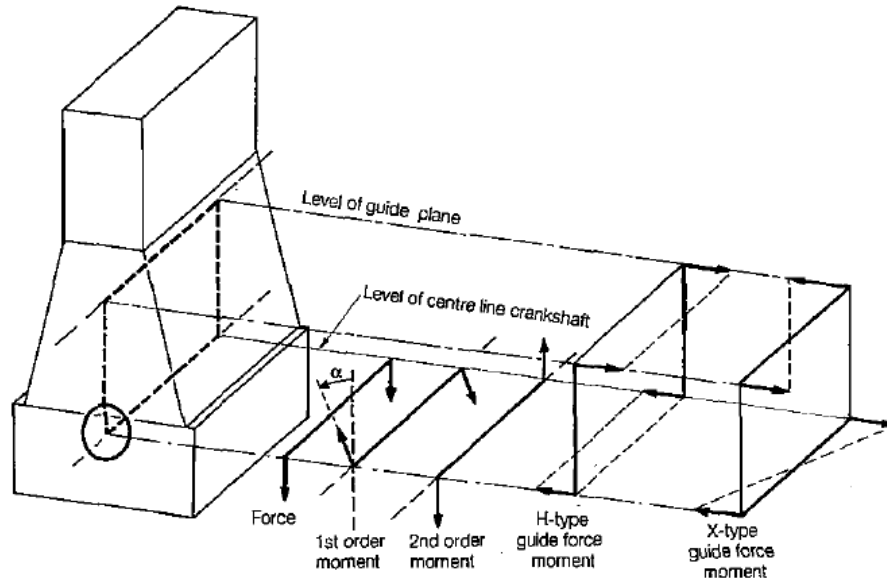


Figure 7.1 Direction of different exciting forces

Equivalent nodal loads for finite element model are shown in Figure 7.2. Forces on the engine block and on the crankshaft have the same phases but opposite initial amplitudes. That allows describing the lateral H-moment.

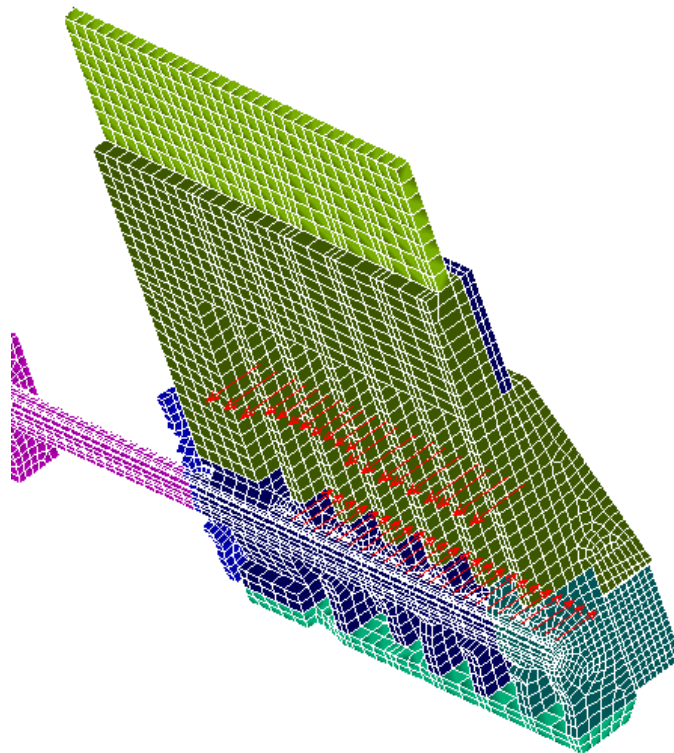


Figure 7.2 Nodal forces (half of the engine model)

7.1 Forced analysis of the engine without the side stays

The structure which was analyzed in Part 6.2 is the object of the forced vibration simulation. Boundary conditions are the same.

Applied forces are described above. Frequency is change from 0 to 12 Hz.

Solution will be presented in the view of *amplitude frequencies curves*. As the whole top part of the engine vibrates in one phase and has almost identical displacements, one node in the middle of the top was chosen for result representation. Node has a number 222412. Position of this node is shown in Figure 7.3.

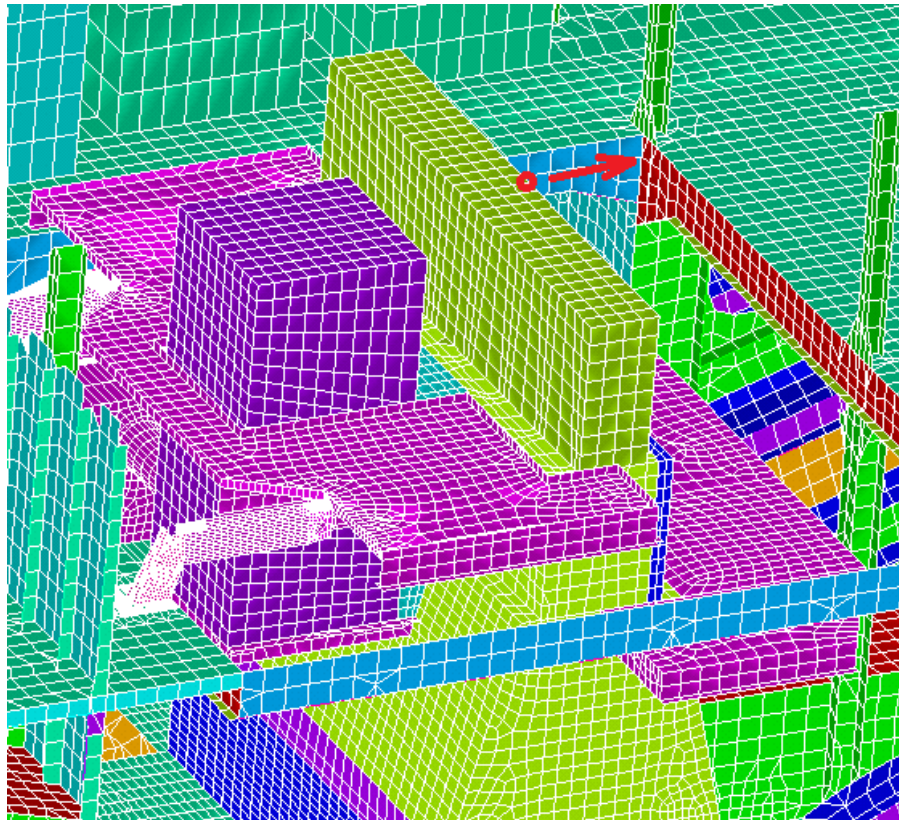
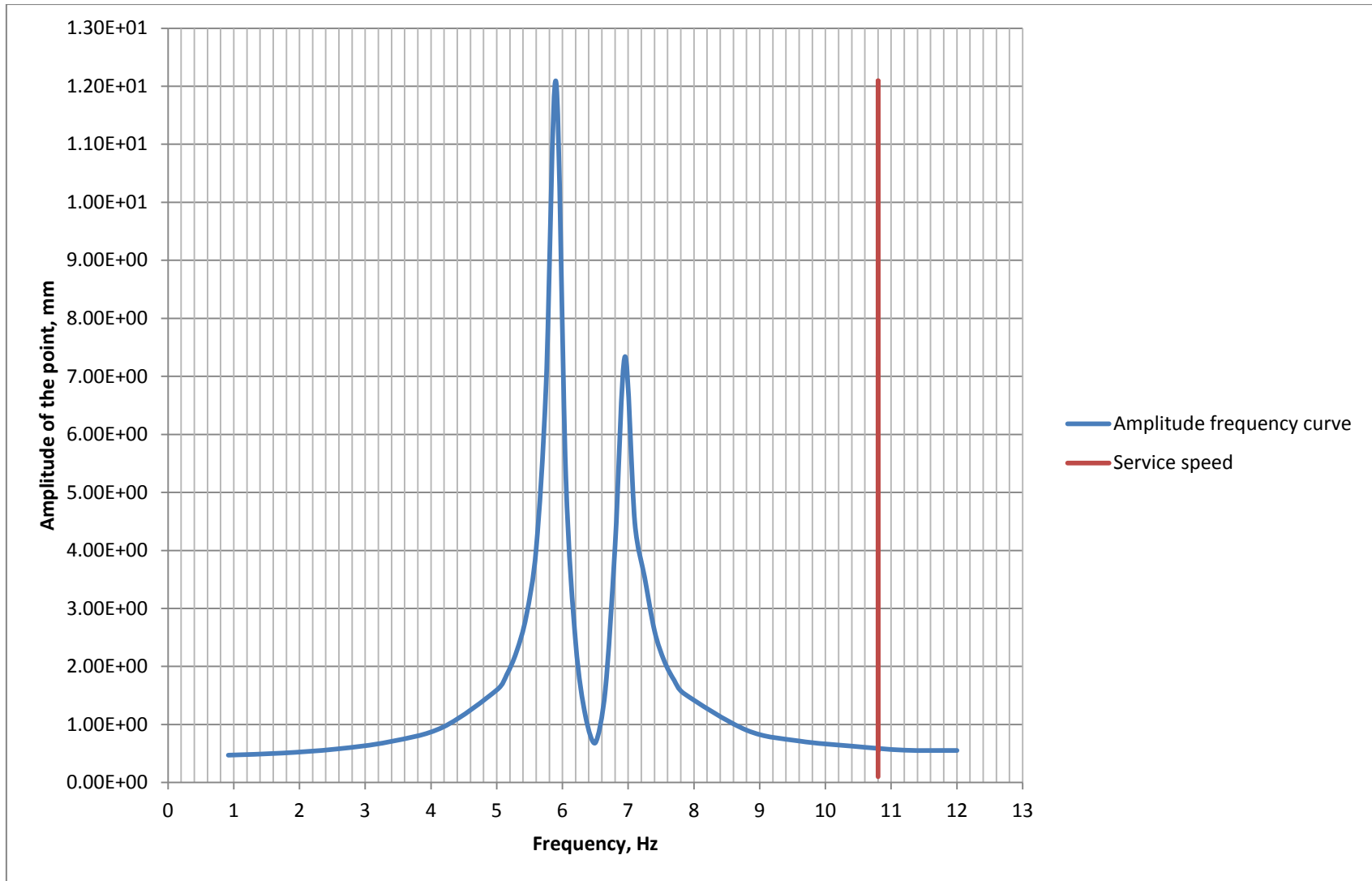
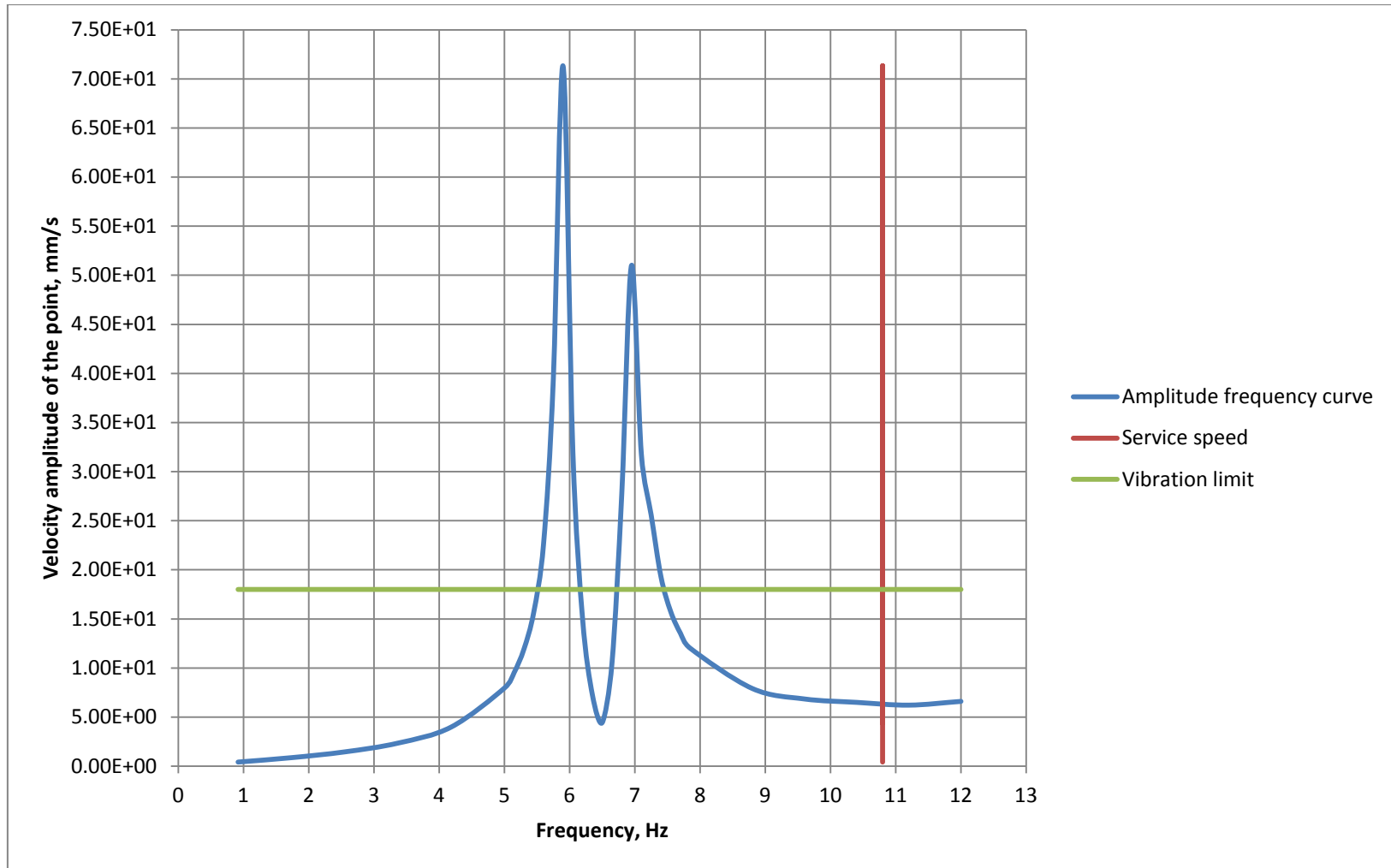


Figure 7.3 Location of node 222412



Plot 7.1 Amplitude frequency curve for the Y-displacement



Plot 7.2 Amplitude frequency curve for the Y-velocity

It is easy to observe two peaks of resonances on the plots. They correspond to natural frequencies 5.895 Hz and 6.92 Hz. Therefore it is possible to conclude that only 2 modes are really H-type engine vibration modes. Mode with frequency 7.15 is not excited by lateral guide forces.

Resonances occur at frequencies which are rather far from the operating speeds region (8-10.7 Hz) and it explains low level of amplitudes in this area (12 mm/s for 8 Hz and 7 mm/s for 10.8 Hz). Such values are under the limit which for slow-speed diesel engine equals to 18 mm/s (see Table 7.1)

<i>Propulsion Machinery</i>	<i>Limits (rms)</i>
Thrust Bearing and Bull Gear Hub	5 mm/s
Other Propulsion Machinery Components	13 mm/s
Stern Tube and Line Shaft Bearing	7 mm/s
Diesel Engine at Bearing	13 mm/s
Slow & Medium Speed Diesel Engine on Engine Top (over 1000 HP)	18 mm/s
High Speed Diesel Engine on Engine Top (less 1000 HP)	13 mm/s

Table 7.1 Vibration limits for Main Propulsion System

The main problem is the process of acceleration and run out as during these operations the system is passing the two resonances. All loads go into the double bottom structure. It is necessary to do such operations as fast as possible and probably implement some intelligent system of the acceleration control.

There is no strong indication on the necessity of installing side stays, however the forced response vibration levels (7-12 mm/s) may be considered as too high. The main idea of side stays using is to bring enough dissipation into the system. Perfect case is not to change stiffness of the system (detuning factor is ok) but just add big amount of the damping to reduce resonances peaks.

7.2 Forced analysis of the engine with installed side stays

Fitting stays between the engine and the hull should reduce the engine vibrations and the vibration transmission to the ship's structure.

Lateral stays are either of the hydraulic or friction type.

- *Hydraulic stays:*

two by two installed on the exhaust and on the fuel side of the engine,

- *Friction stays:*

installed on the engine exhaust side.

Bulk carrier m/s Miedwie is equipped with two friction stays. Position, structural elements and main dimensions of the stays are shown in Figure 7.4.

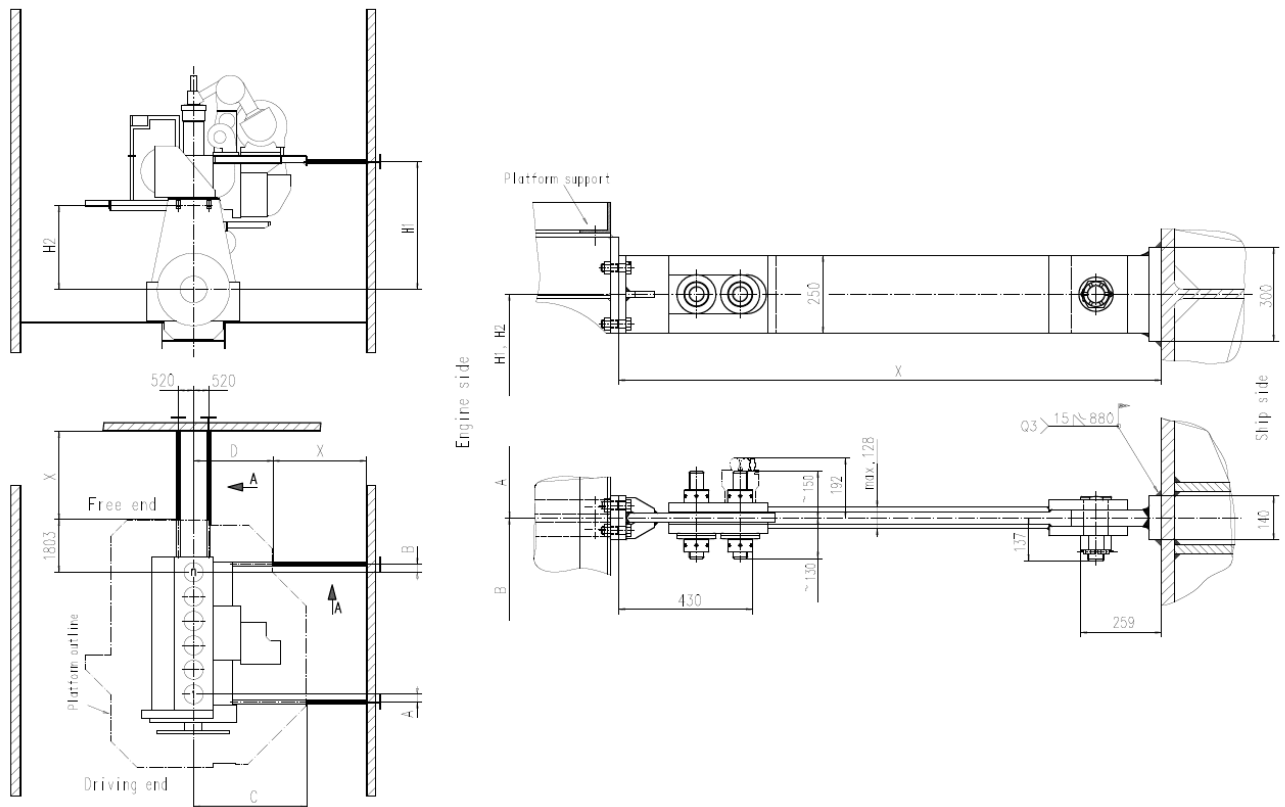


Figure 7.4 Friction stays

The stays are represented by finite element as realistic as possible – plate and solid elements are used (see Fig.7.5).

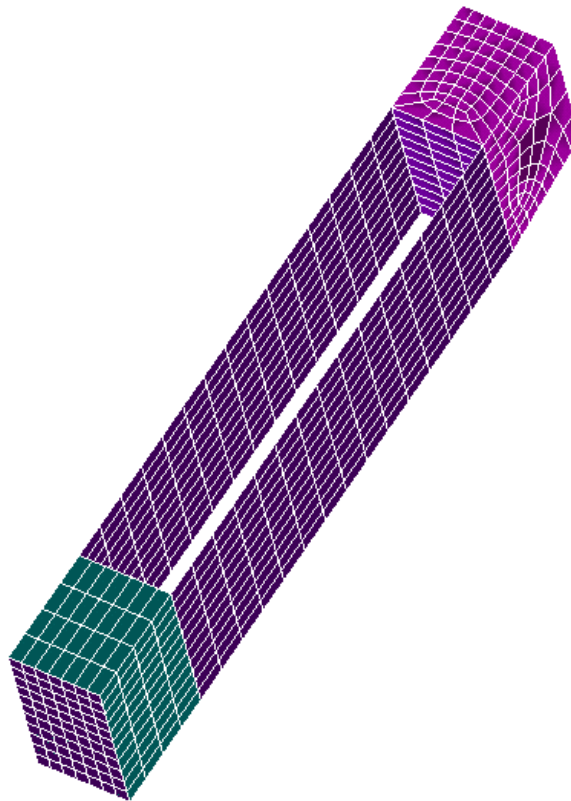
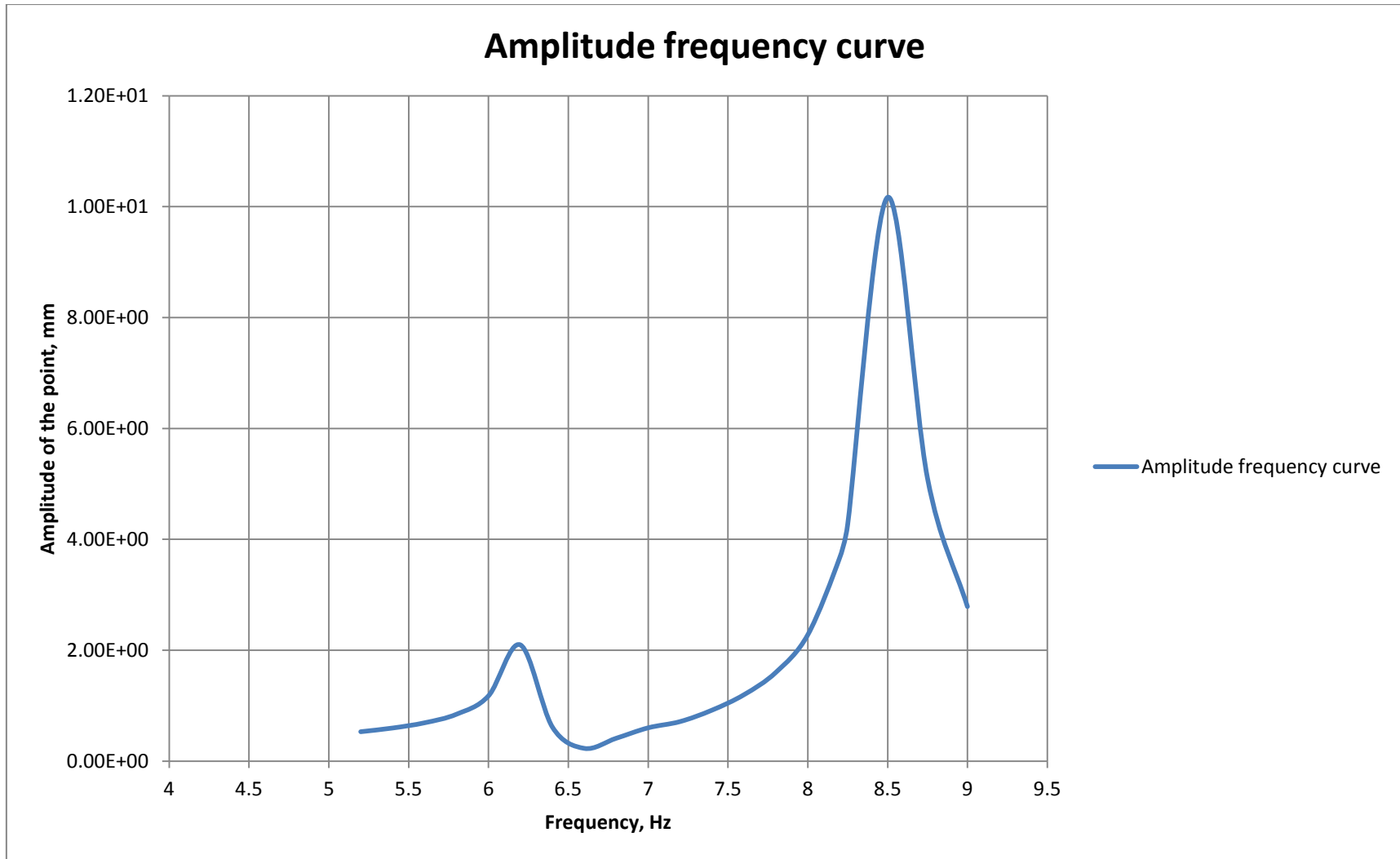


Figure 7.5 FE model of the friction type stay

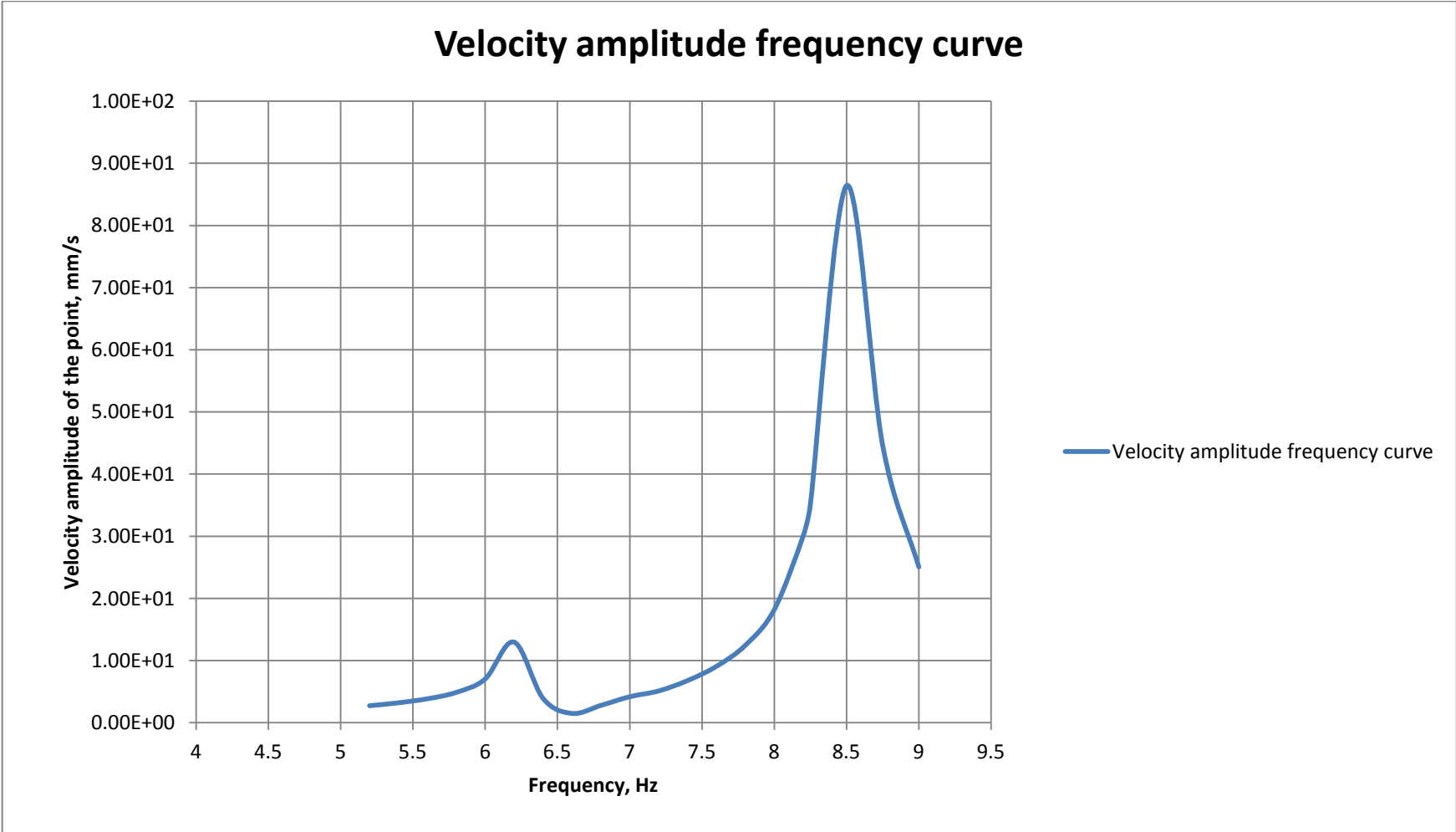
This finite element model is intended to be implemented in the global finite element model. As the analysis is linear, performed in frequency domain the friction force could not be modeled. In this case the stays are investigated as stiff trusses connecting the engine block and the hull supporting structure. This is equivalent to the case when side stays are regulated to tight and friction force is very high.

To investigate the influence of the friction dissipation force on the vibration process it is necessary to perform nonlinear analysis as friction force has a nonlinear nature (see Part 4). Such calculations are too heavy for computational power which I possessed.

Alternative way is to create simplified two mass model using parameters of masses and stiffnesses from linear analysis and then implement nonlinear analysis.



Plot 7.3 Amplitude frequency curve for Y-displacement



Plot 7.4 Amplitude frequency curve for the Y-velocity

Results of the simulation are shown in Plots 7.3 and 7.4. Amplitude frequency curves were obtained for upper part of the engine (node 222214 in the middle of the top).

Curves also have two resonance peaks but only one is dominated.

Forced vibration analysis shows that H-type vibration mode of the engine with side stays has natural frequency around 8.5 Hz. This value is already in the operating speed diapason and it can have dangerous effects. Velocity amplitude near 8.5 Hz is 85 mm/s. It is much higher than acceptable limit (see Table 7.1).

This result explains high level vibration in the engine room under reduced speed (80-90 rpm) reported by crew.

Increased stresses in the areas near hull supporting structure (see Fig. 7.6) are the reasons of the fatigue cracks.

It happened because stiffness of the system was increased but no damping was put into it.

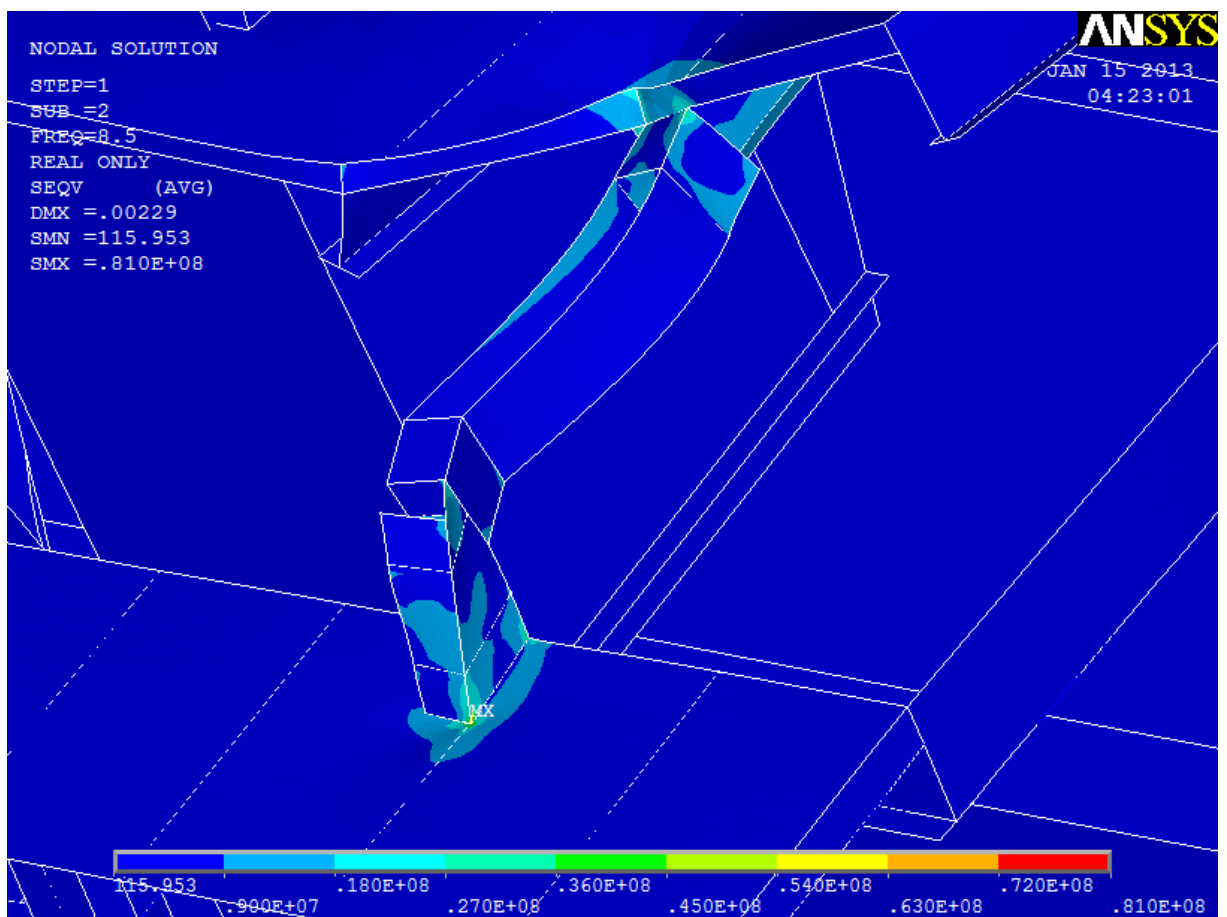


Figure 7.6 Stress distributions. Maximum value is around 80 Mpa.

Figure 7.6 reveals shape of the H-type engine vibration mode with side stays with natural frequency equals to 8.51 Hz.

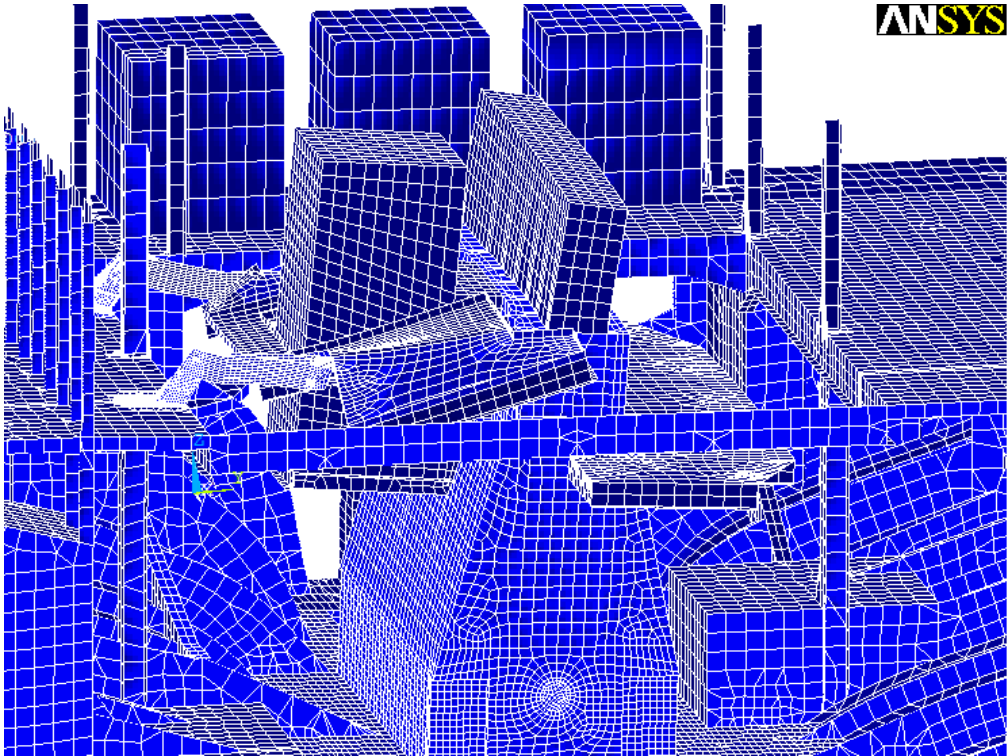
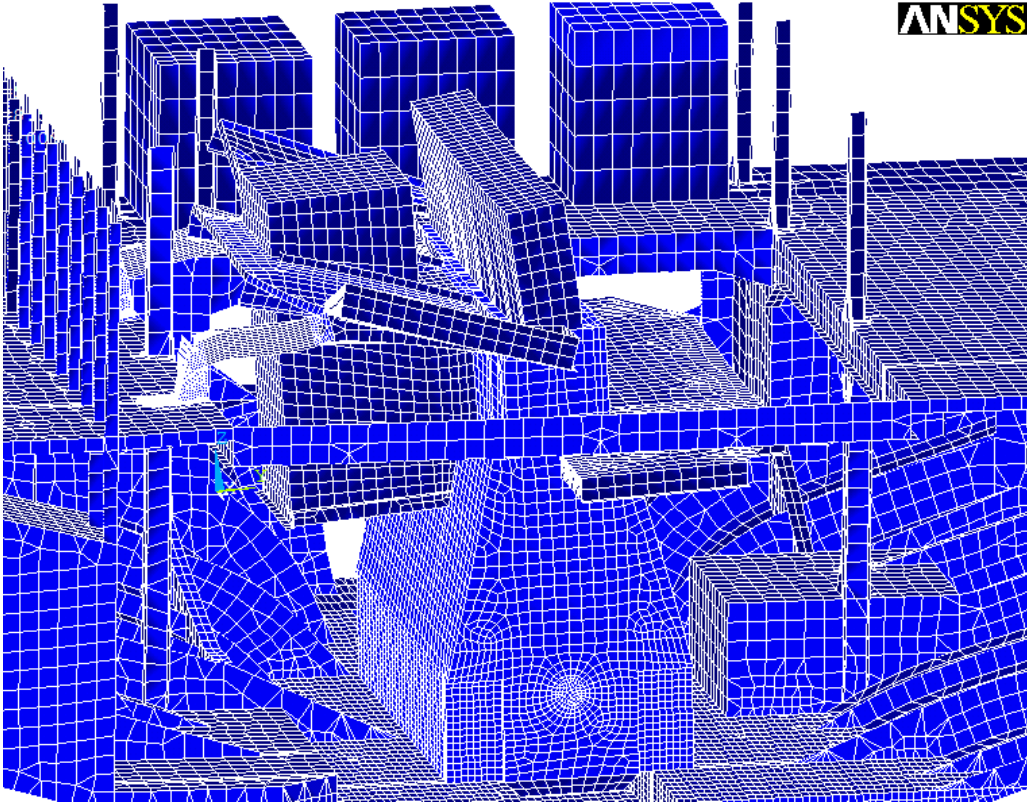


Figure 7.7 H-type mode. Natural frequency is 8.51 Hz

8 CONCLUSION

Present thesis research is devoted to investigation of vibration problem on board of a bulk carrier m/s Miedwie.

After performing of several linear forced vibration simulations in ANSYS APDL it became clear that the main reason of high level vibration in engine room and fatigue cracks is incorrect regulation of the friction side stays. Such regulation shifts H-type natural frequencies of the engine structure to the region of operating speed (80-107 rpm) and, as result, resonance occurs at 8.51 Hz with velocity amplitude equals to 85 m/s (corresponds to the crew reports). Nature of this process is that stays add only stiffness but not damping.

The first way of solution is an accurate installation of stays that dissipation forces start to work. To perform a calculation in that case, a nonlinear analysis is required. Meanwhile, such step was done and that solved vibration problem on the board of m/s Miedwie. It proves the decision which was done about source of the problem.

Possible solution may be an installation of hydraulic stays. Here is dissipation force in view of viscous force in the cylinder actuator. Engine structure with that type of stays is possible to simulate by mean of linear harmonic analysis in ANSYS APDL. Unfortunately, there is also a problem. In contrast to a pair of friction stay, four hydraulic stays are required (see Fig.8.1). Present ship does not have supporting hull element on the level of lower engine platform and therefore it is impossible to adjust two new stays. Nevertheless vibration behavior of the engine system with new type stays may be investigated with modified hull design. This topic may be studied in future researches.

Another interesting question for future research is to model engine foundation more realistically. Epoxy resin gaskets may be installed between the engine and the doublebottom. Such resins have large dissipation potential so, probably, using such complex foundation it is possible to solve vibration problem without side stays.

Moreover the significant meaning of the accurate stiffness modeling for engine structure was determined. Absolutely rigid engine gives positive shift up to 2-2.5 Hz comparing to realistic design. That leads to the overestimation and value of the numerical simulation is utterly reduced.

Conversely, absence of the superstructure slightly impacts natural frequency of the engine (the difference is around 0.2 Hz). That fact allows us to model superstructure very simplified (only masses) or do not include at all.

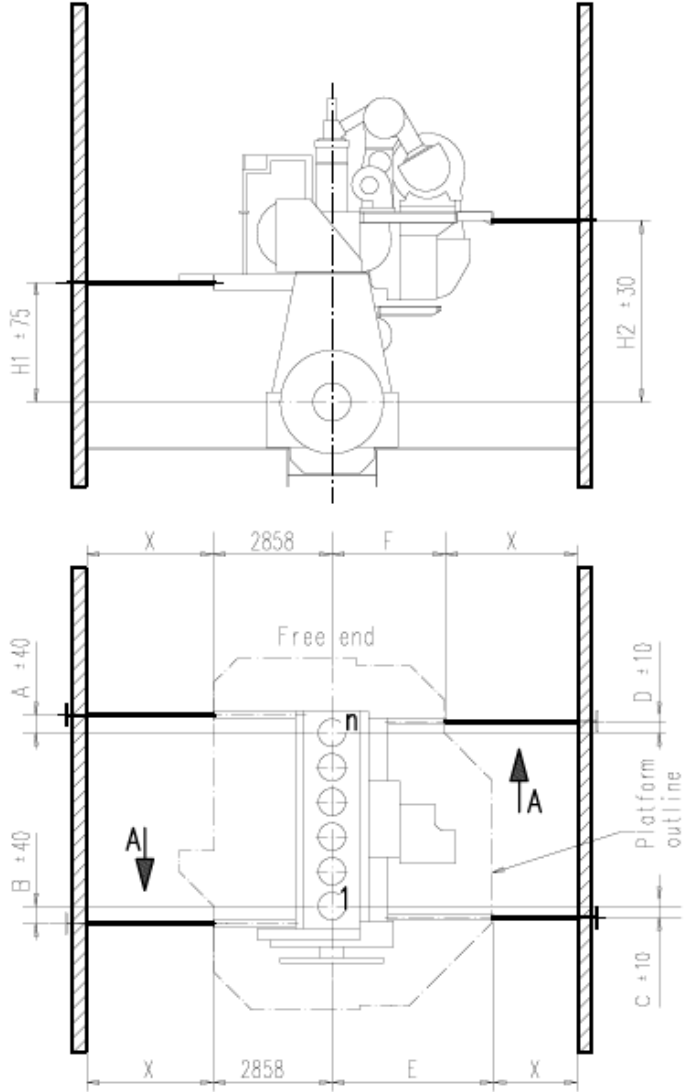


Figure 8.1 Hydraulic type stays

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APPENDIX 1

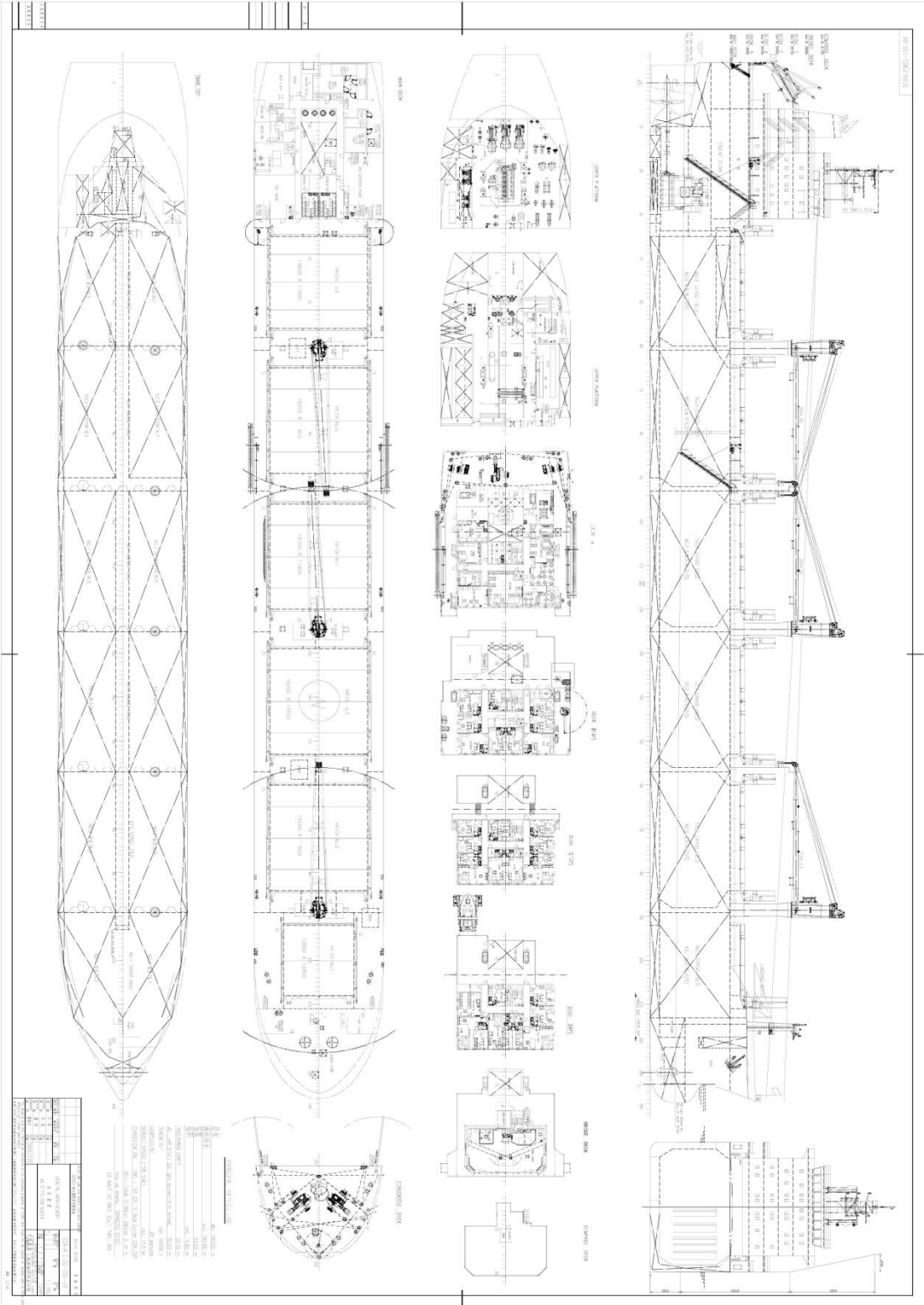


Figure A1.13 General arrangement

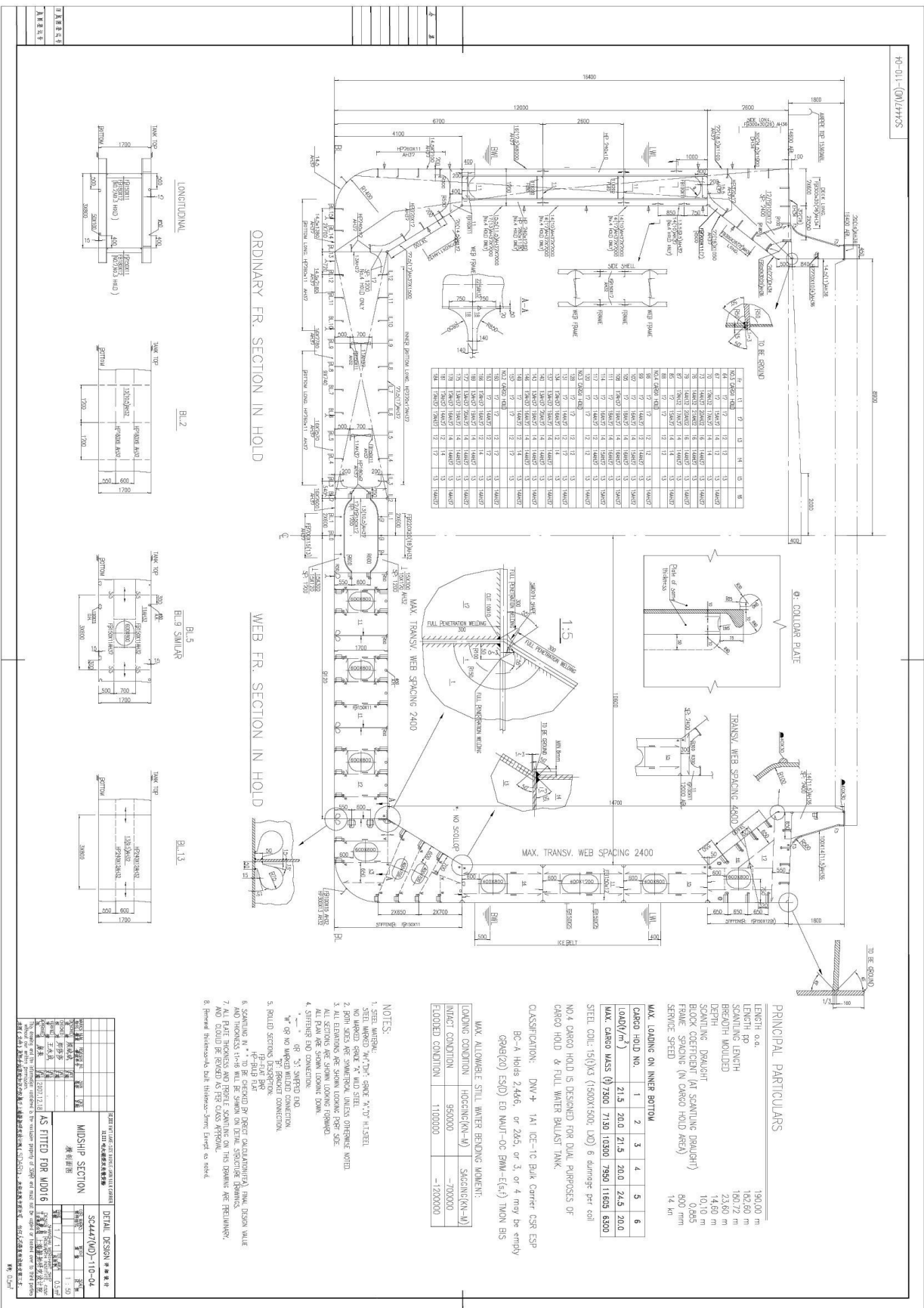


Figure A1.14 Midship section

APPENDIX 2

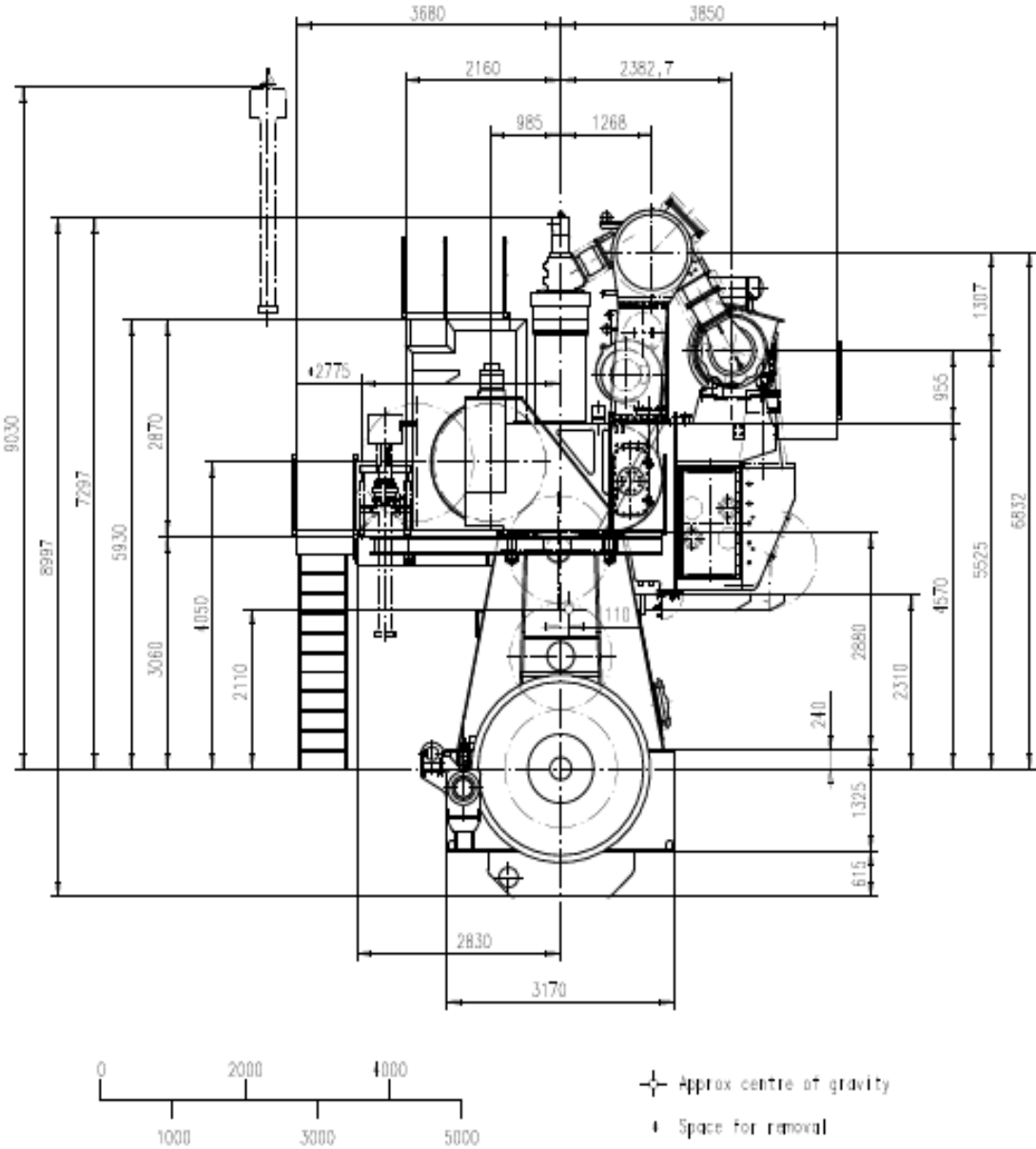


Figure A2.15 Outlines of Wartsila RTA48T-B engine

Approximate position of centre of gravity is shown on Fig. A2.1 and Fig.A2.2.

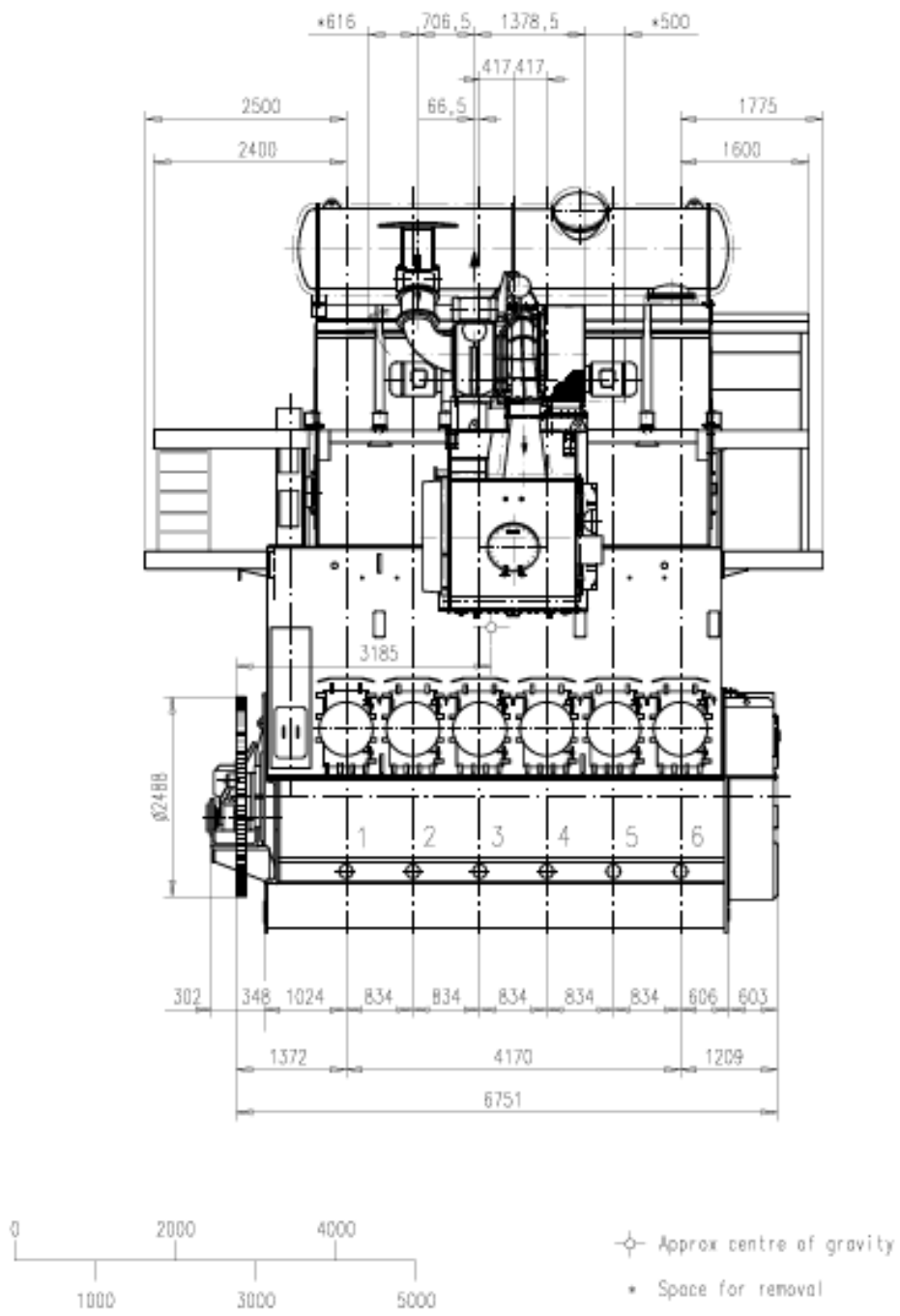


Figure A2.16 Side view of Wartsila RTA48T-B engine

APPENDIX 3



Figure A3.17 Upper platform and side stay

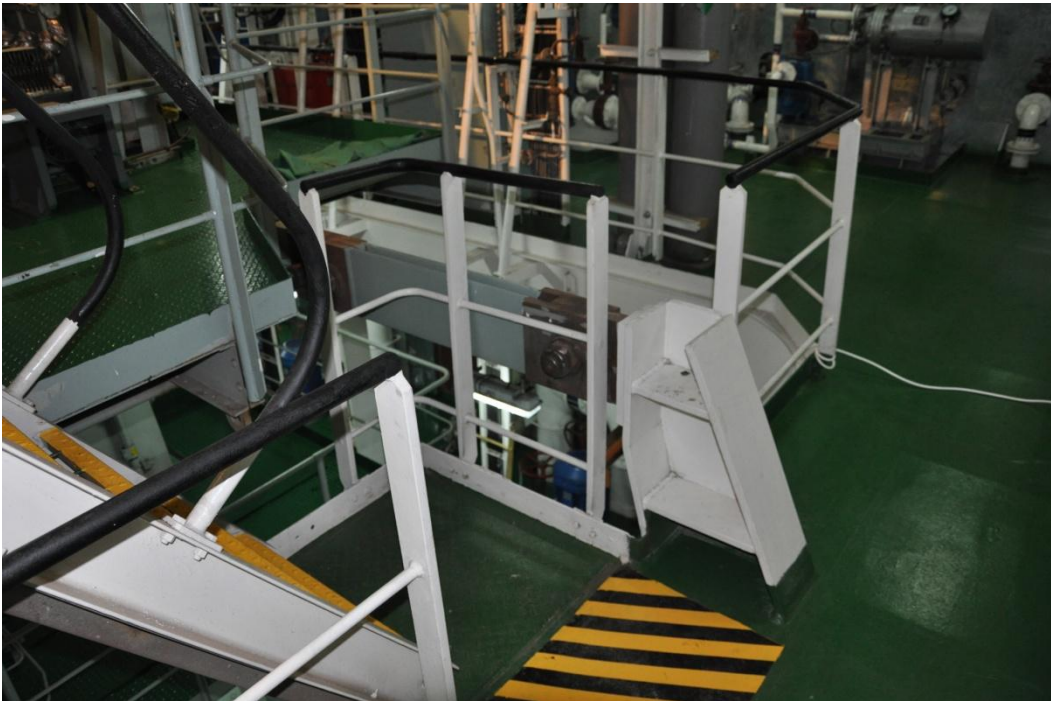


Figure A3.18 Upper platform and side stay



Figure A3.19 Top view of upper platform



Figure A3.20 View on upper platform (driving end side)

APPENDIX 4

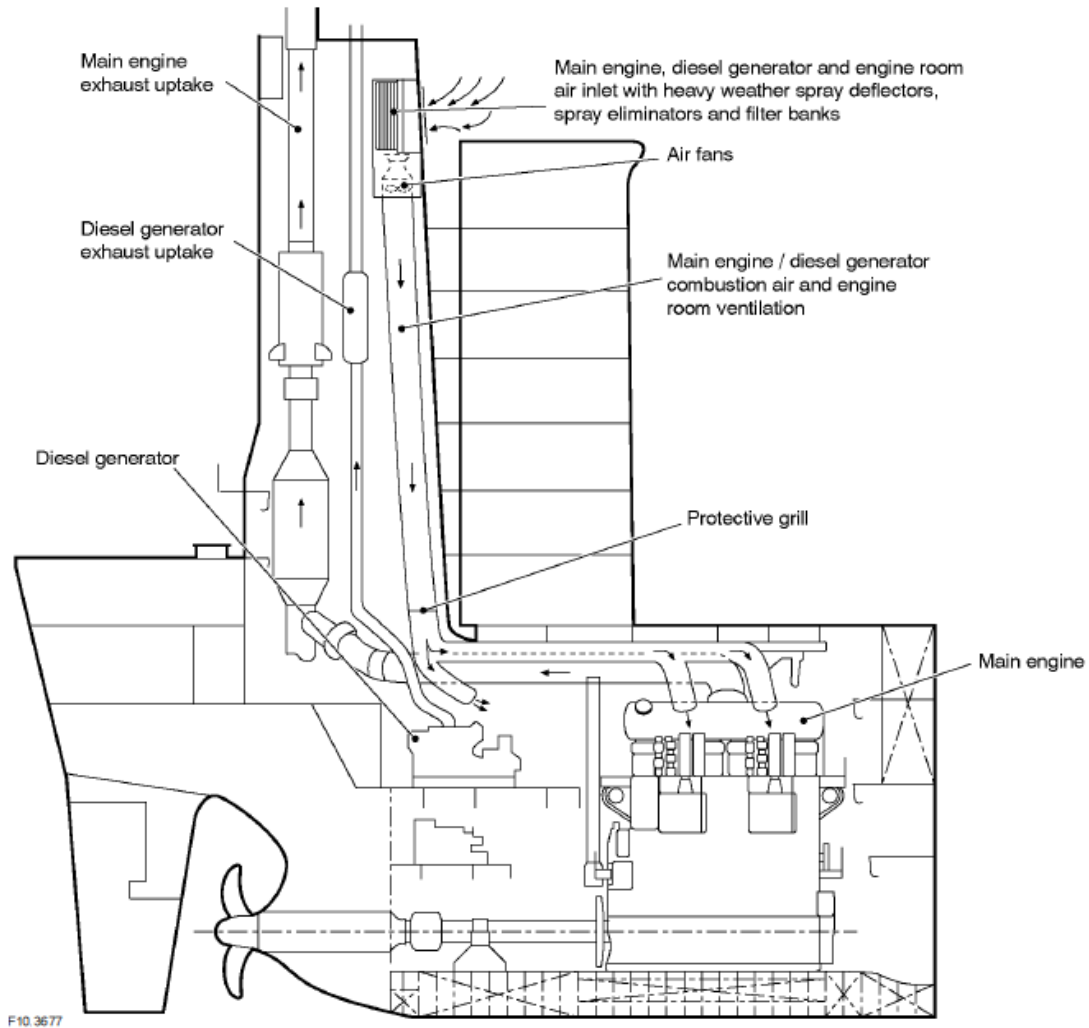


Figure A4.1 Direct suction of combustion air

Except structural failures of the foundation or support elements which can be caused by the main engine vibration there is another problem linked with high level of the oscillation. Figure A4.1 shows that a vast and complex exhaust system is attached to the top of the engine. High level of its vibration can break working capacity of whole tract and this is dangerous due to a risk of high smoke pollution inside the deckhouse and engine room.

Actually exhaust tract adds additional stiffness to the engine structure but its influence is considered as relatively small.

Also it is worth to notice that lateral vibration of the engine is transmitted through the funnel to the superstructure.