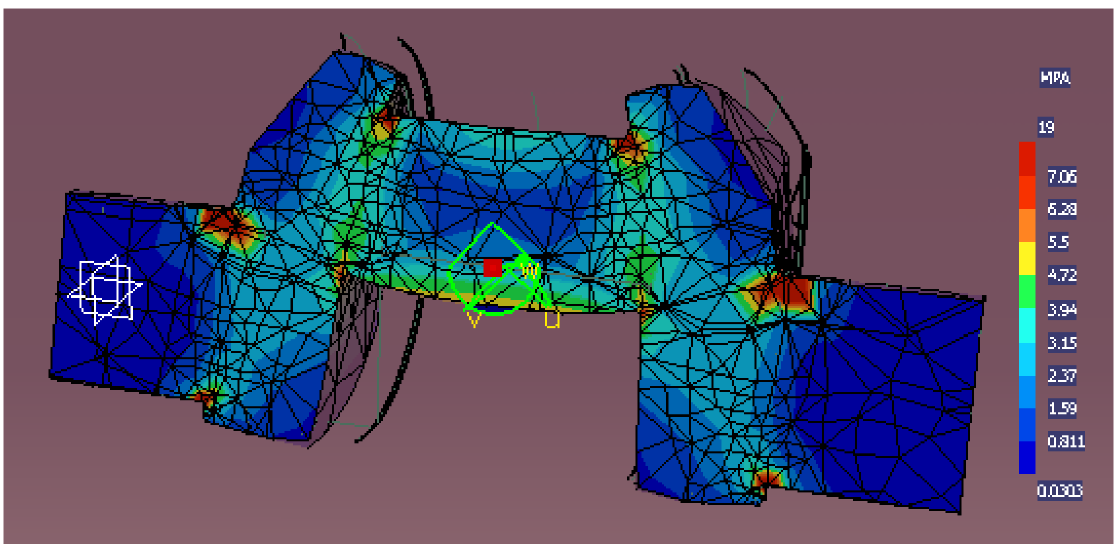
CONSTANȚA MARITIME UNIVERSITY
FACULTY OF MARINE ENGINEERING

STUDIES AND RESEARCH IN THE DYNAMIC BEHAVIOUR OF MARINE STRUCTURES



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The present was born as an incentive to use advanced engineering methods in the study, design and building of marine structures.

Chapter 1 presents introductory aspects on the tackled theme, the necessity and opportunity of modelling phenomena by modern behavioural study methods with a view to achieve more refined design solutions, for a better understanding and modelling of solutions and an efficient outcome even before the prototype is manufactured.

The phenomena analyzed with finite elements have been assumed as being independent with the time.

In dynamic analyses, the variations of movement in time take place so fast that by ignoring the initial effects can no longer be considered as viable pre-requisites for the analysis and building of FEA model of the system.

The effects of the damping forces which are associated to the structure’s velocities (first rank derivates of displacement) will have to be included in the field equations as well.

The dynamic analysis can be done both on the time and frequency domain; the latter can only be applied to structures with a linear behaviour or to line up a fluctuation over a certain equilibrium state, the frequency analysis being the natural analysis mode of free vibrations.

The essence of static or quasi-static structural analysis is the desire of the analyst to derive the influence of applied forces over the structure, and thus the structure displacements will consequently be taken into consideration as the primary variable in the analysis, since they can be easily determined even for intricate structures.

In the dynamic analysis the displacements become even more important due to the fact that velocities and accelerations are in the end derivates of displacements.

To approach the structures’ analysis only by considering forces is a less reliable approach and leads to useless Mathematical accomplices as long as the dynamics in its simplest definition speaks about the structure’s *displacement*.

Chapter 2 focuses on the actual state of research and outcomes in this field.

FEM goal in the dynamic analysis is the following: given a geometrically well defined solid together with its charges and border conditions, we pursue to determine its dynamic behaviour with a certain imposed accuracy. For the static analyses, the accuracy is obtained by using the common methods of successive refining of the network of finite elements for method-h and the increase of polynomial rank to method - p.

FEM programmes offer the user complex objectives to create the Mathematical model.

At the same time we must take into consideration the following issues:

- The behavioural type which has to be represented in the model must use the appropriate type of finite elements

- The capabilities of the finite elements to represent distortion gradients must be taken into consideration, and in the areas where high gradients are expected the number of elements must be increased

Generally FEM programmes include a number of options to generate weight matrices:

- elements with concentrated weight
- distributed weight associated to continuous elements
- possibilities to directly insert the weight matrix

FEM programmes include a number of options to describe damping:

- Discrete damping elements
- Structural damping
- Modal damping
- Ratio damping

FEM programmes enable to define a set of loadings dependent on space-time. The analysis with finite elements is meant to stimulate physical systems by means of models.

Chapter 3, the most extensive of the thesis, is intended to the analysis of the theoretical fundament of both phenomena determining the dynamic behaviour of marine components and the importance of applying numerical methods in their design by computational methods.

Any software by FEM analysis presupposes the following steps:

1. Breaking down the continuous physical system in a network of discrete finite elements, which, in its turn, has to:

a. disconnect at first the finite elements one from the other and attach to each finite element a proper local system of co-ordinate axes,

b. locate each element- meaning that we consider to compute each finite element taken isolated in its local co-ordinates,

c. compute the rigidity matrix $(K)^e$ for each finite element.

2. Putting together or reconnecting the finite elements among them, a stage which supposes:

a. Globalizing by which the rigidity matrices (equations) of each finite element are reported from the local co-ordinate system to the global system of the physical system. Through globalization matrix relationships are established which connect the displacements and nodal forces of the finite elements taken in global co-ordinates.

b. Connecting all rigidity matrices (equations) of finite elements taken in the global co-ordinate system. The result is the Master matrix or global rigidity matrix of the physical system to which each finite element takes part with its proper rigidity matrix. Physically, the connecting process can be interpreted as a reconnection of all finite elements to "build up" the initial continuous structure of the discrete physical system. In this respect, for each common node of two or more finite elements the compatibility condition will be obeyed for displacements of finite elements with common nodes and the equilibrium of local forces which sets the condition that the sum of nodal forces should be nil in order to preserve the structural equilibrium.

3. Solving the equation systems - a two steps process:

a. Applying border conditions through which the global rigidity matrix is modified in order to insert the effects of conditioned displacements and/or the forces imposed at borders by backings and loadings. The global equations thus modified are solved by different equation systems solvers (Frontal Solvers, Sparse Direct Solvers, Jacobi Conjugate Gradient Solvers, Preconditioned Conjugate Gradient Solvers, Automatic Iterative Fast Solvers, etc.), to recover the displacements of each node of the finite element.

b. Post-processing by which derived quantities are derived such as internal forces, tensions and reactions in backings, all derived from the solutions of nodal displacements.

FEM programmes cover these stages simultaneously, for example applying the border conditions before connecting the local rigidity matrices, explicitly removing the nodal displacements known from the equation systems while these are connected. Other programmes perform the simultaneous connection and solving with the help of frontal solving.

The aim of **Chapter 4** is to present the research works and experimental results referring to the analysis of the entire crankshaft, between extreme planes. The experimental results are then compared to the ones obtained by using other calculus methods. The differences of about 10% for the comparison with non-determined static methods are tolerable and differences of 10÷80%,therefore very high, in reference to the static determined one; in fact the range of allowable values for the coefficient of fatigue safety is 0.9÷1.3.

Comparing the two category methods was focused on the specific data of a Diesel generator engine type AT25 manufactured by New Sulzer Co. Here we have presented the sub-structural model of the crankshaft together with the results obtained from the calculus of influence coefficients received from a proper model of the above four cycle engine with four cylinders in line.

When comparing an essential illustrated fact clearly results, namely that determining the fatigue safety coefficients of the crankshaft is a much more sensitive matter than checking the bearings; we may also ascertain that the static determined methods lead to a rather inconclusive prediction of the crankshaft loading amplitudes and fatigue endurance and does not highlight on the critical engine operating regimes.

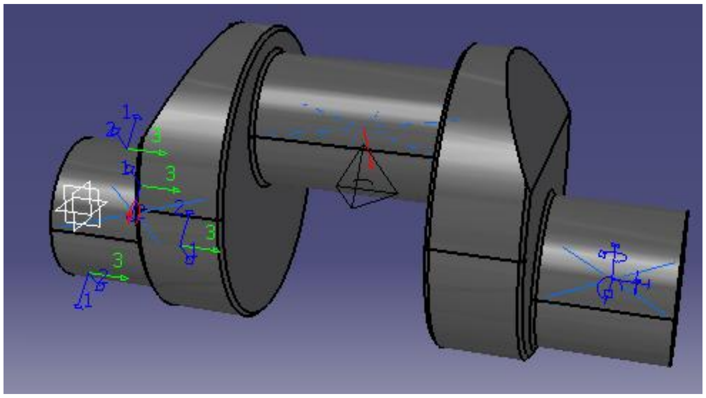


Fig. 1.1- Scheme of the loadings applied to the crankshaft.

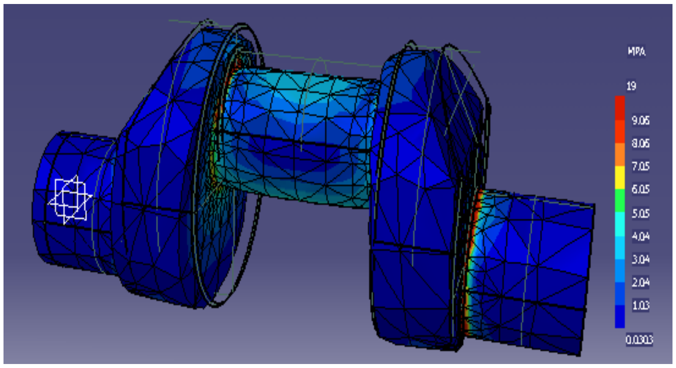


Fig. 1.2- The spatial tension state at the crankshaft surface

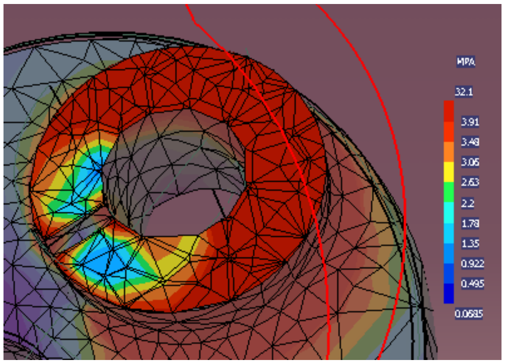


Fig. 1.3 – Section through the crankpin in the greasing area

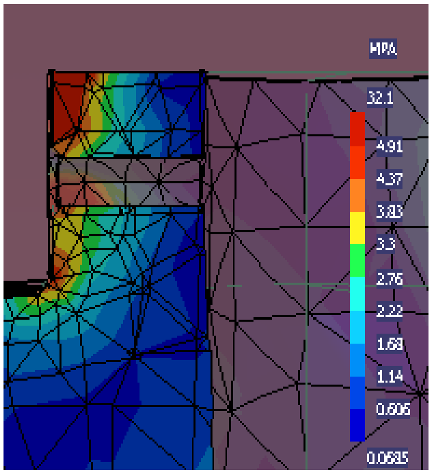


Fig. 1.4- The greasing area in the crankpin

In **Chapter 5** the model validation is done by a comparative analysis of the results from the calculus of the proper values (proper vibration frequencies), for Sulzer 6RND90 engine and an engine type MAN K6SZ 52/105, installed on board of a container ship of 12500 tdw in the Romanian merchant fleet, of 4700 kW for 140 rev/min, with the same crankshaft layout as the Sulzer type and pulling along a four blade propeller of 4.2 m in diameter (the propeller pulled by the Sulzer type is 6.4 m in diameter). The comparison is made between the proper frequencies of the individual vibration types and the coupled ones.

- The values of the proper pulsations of the engine K6SZ 52/105 Cle are higher than those corresponding to engine 6RND90; this can be explained by the fact that the weights and the inertia momentums are lower for the first engine with respect to the second one;
- The proper torsion pulsation of 1st degree is close to the axial one of 2nd degree , a phenomenon much more pronounced for the 6RND90 engine; this suggests the possible occurrence of a coupling phenomenon between the mode 1 of torsion vibration and the axial one of 2nd degree in the sense of the latter being excited by the former one;
- The proper pulsations of the coupled vibrations are lower for both engine types then the individual torsion ones, but higher than the individual axial and bending ones, which shows the coupling phenomena between all individual vibration types;
- The proper pulsations of the individual vibration modes have the smallest values, the influence of this vibration type on the coupling phenomena being by far reduced than that of the individual torsion and axial types;
- The proper pulsations of the coupled vibrations of K6SZ 52/105 CLe engine are very close to the ones of the axial individual vibration modes; this leads us to the suggestion of a powerful influence of the axial vibrations within the coupling phenomena; for the same engine the proper pulsation of 2nd degree of the coupled vibrations is close enough to the one of the first degree of the individual torsion vibrations a fact that

further confirms the pronounced phenomenon of coupling between the respective vibration modes.

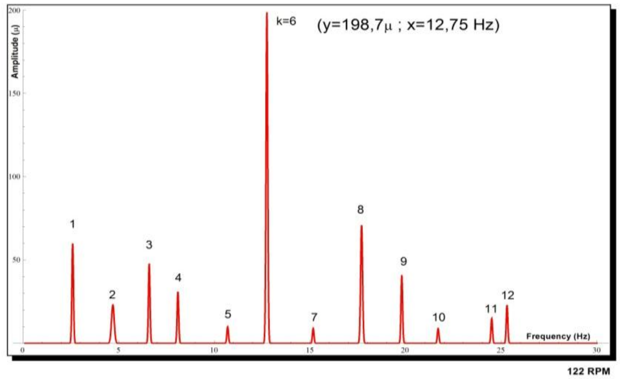


Fig.1.5.The analysis in frequency of the axial vibration amplitude of the engine Sulzer 6RND90 free end

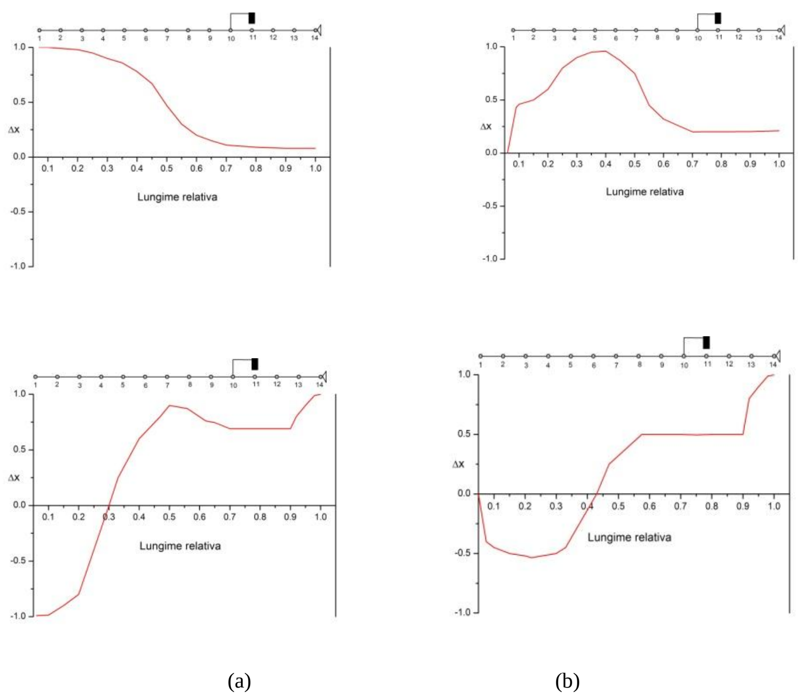
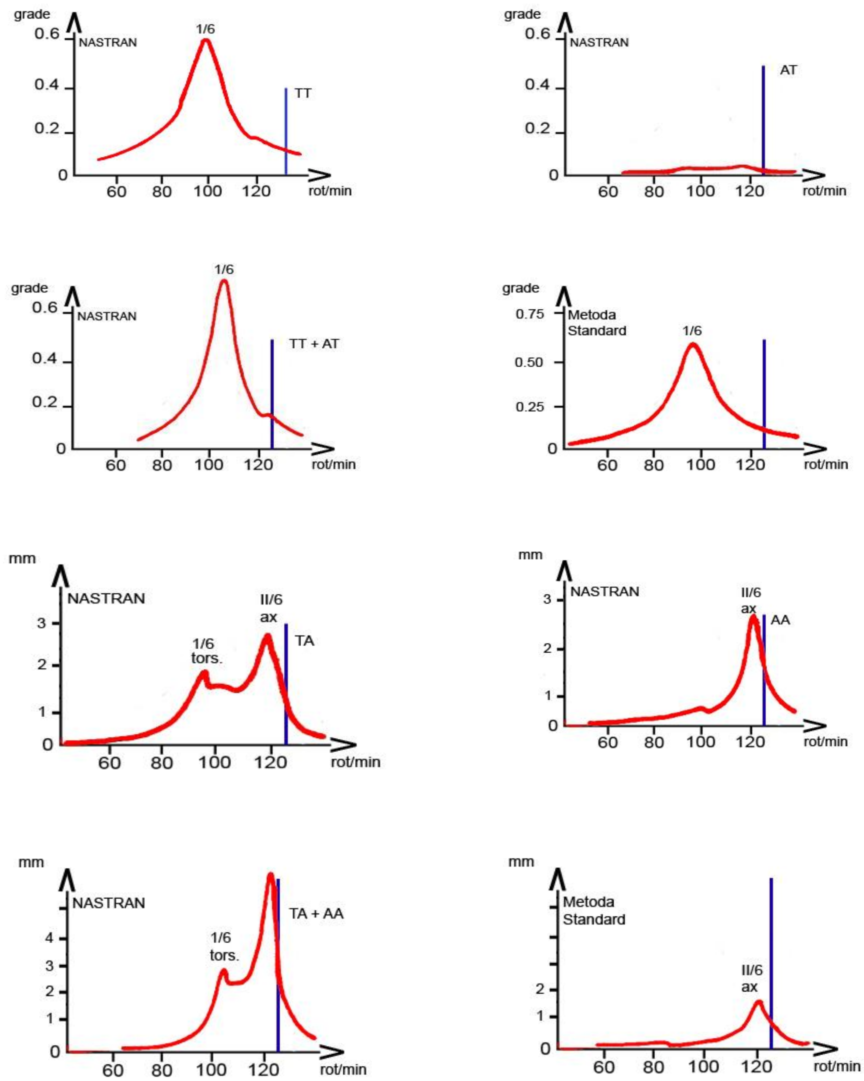
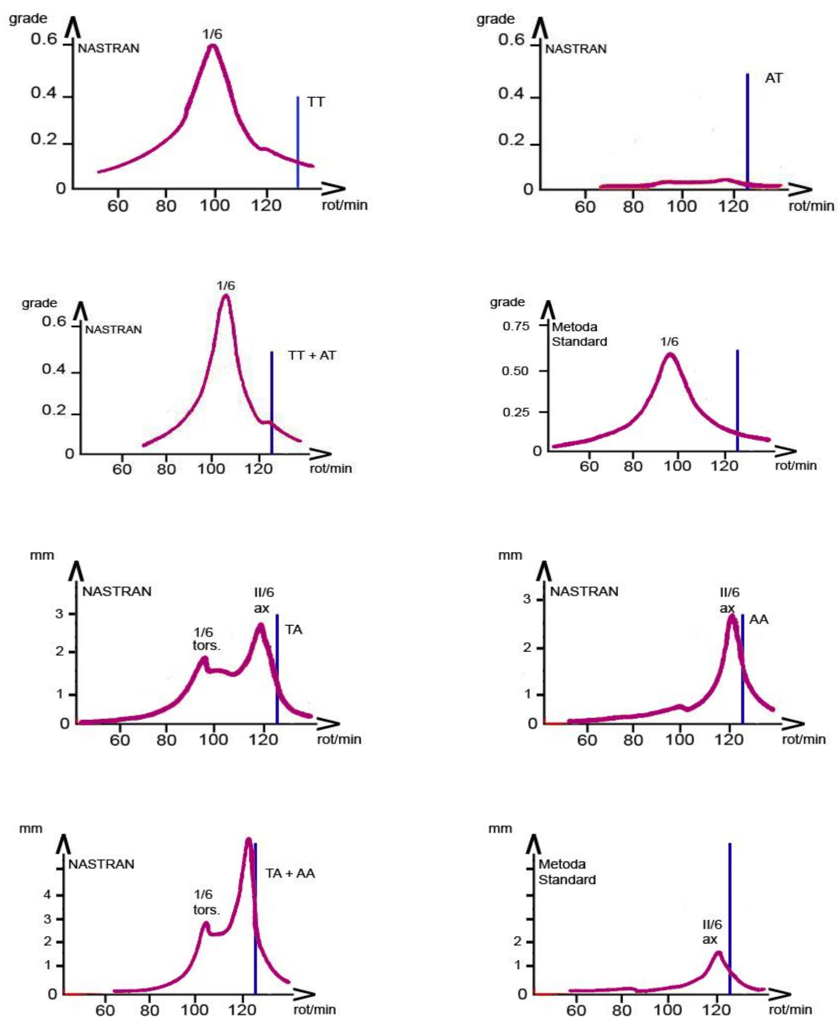


Fig. 1.6. Proper individual axial vibration modes of Sulzer 7RTA62 engine:
a- without damper; b-with damper for axial vibration.

Torsion amplitude | Axial amplitude



Torsion amplitude   Axial amplitude



At present the analysis with finite elements is the most performing analysis instrument. This is used for mechanic and thermal strains and tensions in the complex structures specific for mechanic buildings.

Due to high performances which enable to optimize products even in the design phase this instrument is largely used currently because its use leads to a consistent cut in the assimilation costs since the number of tests is very much reduced.