INVESTIGATION OF THE DYNAMIC OF SHIP STRUCTURES USING CLASSICAL AND OPERATIONAL MODAL TESTING

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Abstract
The prediction of dynamic behaviour of ship superstructures is important to avoid high vibration levels under working conditions. For model updating purposes and forced vibration calculations experimental determined resonant frequencies and damping values are required. Several studies at the Department of Technical Mechanics have shown that the application of classical modal analysis for the investigation of big ship structures on a ship yard is possible. Under real working conditions the use of classical modal analysis is not practicable for several reasons. To assess the potential of operational modal analysis for a further use under working conditions, both modal testing techniques were applied during experimental investigations on a container vessel at Aker MTW yard in Wismar. Modal testing procedures and obtained results are presented and compared.

1 Introduction
To maximize the storage capacity of modern middle size container vessels, superstructures are usually placed at the back part of the ship, are tall and have a small depth. These requirements of the market are leading to constructions with a reduced stiffness compared to constructions based on the theory of ship design. The main excitation sources, the propulsion system and the main engine, are also placed at the back part of the ship. For these configuration the risk of vibration problems is very high.

To fulfil the demands of classification societies and ship owners according to vibration levels, a precise prediction of the dynamic behaviour of ship structures during the stage of development is necessary. The quality of the prediction model can be evaluated carrying out experimental investigations on the real ship structure. To calculate forced vibrations, the implementation of damping values into the calculation models is required. These damping values also have to be identified by experimental investigations. Usually experimental modal analysis is carried out at a shipyard. But it is known that on ships under working condition, changed boundary conditions affected for instance by the influence of deep water, storage and ballast conditions, are leading to changed dynamic parameters. Thus the identification of resonant frequencies, corresponding mode shapes and damping parameters of ship structures under working conditions is most important.

The presented investigations were done during the preparation of a future research project. In this project classical and operational modal analysis will be applied to ships both at the ship yard and under working conditions. In the presented study the different modal testing procedures were applied to different ships, so individual results cannot be compared. The main interest was focused on the applicability of different modal analysis techniques and on the quality of the results obtained.
2 Numerical Prediction

To get an impression of the range of resonant frequencies and their corresponding mode shapes which have to be investigated by experiment, finite element calculations were carried out. Since the main excitation frequencies of a ship structure under working conditions can be found in a range up to 20Hz, only this frequency range was inspected. The finite element model of the complete ship structure was provided by the Aker MTW ship yard. For the calculation the finite element program MSC Nastran was used. Selected results are shown in the figures 1 to 4. It can be seen, that the vibration of the superstructure is strongly coupled with the vibration of the whole ship body. In the lower frequency range up to 15 Hz mode shapes can be described as different kinds of bending and torsion modes. Several modes with a bending and torsion deflection of the superstructure only differ in the deflection shape of the ship body.

Because of the limited time available for the experimental investigations, measurements with a reduced expense were necessary. So only three points at the top of the superstructure were chosen as measurement points. These points are sensitive to both mode shapes but they are not suitable to distinguish between the different torsion and bending mode shapes. Increasing the number of measurement points at the superstructure only, the results would not be improved for the desired frequency range. For further investigations additional measurement points at selected locations of the ship body have to be chosen to obtain orthogonal mode shapes.

Because of the complexity of the model the number of eigenvalues calculated is very high. To select these mode shapes of the free vibration calculation which are relevant for the behaviour of the superstructure a forced vibration calculation was carried out exciting an outer point at the top of the superstructure. Relevant frequencies and corresponding mode shapes of the free vibration calculation were selected by comparison with the resonant peaks of calculated response. Selected
mode shapes are shown in figure 1 to 4. The calculated response and ranges of characteristic mode shapes are shown in figure 5.

Figure 5. Forced vibration calculation, excitation and structure response at the outer side on the top of the superstructure (point 1, figure 6)

The main excitation frequency of the ship under working conditions can be calculated by the speed of the main engine multiplied with the number of propeller blades and can be found in a frequency range between 7Hz and 9Hz. Taking into account the secondary order of main excitation, attention has to be paid also to the frequency range above 9Hz.

3 Application of Classical Experimental Modal Analysis (Single Input-Multiple Output, SIMO)

In a recent project the dynamic behaviour of a main engine was identified using experimental modal analysis [1]. The structure was excited by a pendulum with a mass of 600 kg. The same excitation technique was applied investigating the dynamic behaviour of superstructures. The excitation mass either supported by a crane (figure 7) or mounted on a rail (figure 8) was applied at the top of the superstructure (figure 9). The excitation force was measured by a piezoelectric force transducer, the acceleration response of the structure by piezoelectric accelerometers. The excitation forces realized were approximately 25kN, resulting in a maximum acceleration of the structure of approximately 0.2m/s². The locations of the transducers (in direction of the length axis of the ship) are shown in figure 6. The data were recorded with a HP VXI system. For data processing and identification the software I-DEAS Test was used. Experimental modal analysis.
was applied onto different types of ships slightly differing in total length, superstructure construction and main engine mounting.

Figure 7. Exciter, supported by a crane swinging against a stiff point of the superstructure

Figure 8. Exciter, completed by wheels and mounted on a rail

Figure 9. Superstructure with point of excitation

Figure 10. Frequency response function of the driving point (point 1, figure 6) of ship type A

Figure 11. Frequency response function of the driving point (point 1, figure 6) of ship type B

Using the spectra of the known excitation force and the resulting accelerations frequency response functions were estimated. Experimentally determined frequency response functions are shown in
figures 10 and 11. Using the complex exponential identification technique resonant frequencies and damping values of selected modes were identified. Because of the high modal coupling of several modes and measuring only at three references a parameter identification of all modes was not possible. The shapes of deflection were determined by using the move response technique inspecting the frequency range of interest. The characteristics of deflection shapes obtained are corresponding nearly with the mode shapes calculated and are shown in figure 5. Between 1Hz and 7Hz a first order bending (see also figures 1 and 2) and between 7Hz and 16Hz a first order torsion of the superstructure (see also figures 3 and 4) were identified. The frequency response functions of ships type A (figure 10) and B (figure 11) differ in the appearance of a significant mode between 4Hz and 5Hz. This phenomenon may be caused by the different length of the ships. The differences in length could result in changed locations of vibration nodes leading to different vibration amplitudes of the superstructure.

4 Application of Operational Modal Analysis (OMA)

The application of classical modal analysis on superstructures using a big excitation mass is restricted to the use on a shipyard only. To obtain modal parameters of ships under working conditions (added cargo, higher vibration amplitudes, deep water) the application of operational modal analysis might be a promising approach. To investigate the capabilities of this testing technique operational modal analysis was applied to two container vessels of type B. During the first measurement the machines were not running and the ship was excited only by quite strong wind (10m/s) and waves. This loading should be random in time and space with small correlation length compared to the size of the ship and a multiple input loading can be supposed [2]. To investigate the influence of additional harmonic excitation resulting from running machines, a second measurement was carried out when a power plant (8 cylinder four stroke diesel engine, generator, rotational speed 916rpm) was running.

To measure the vibration piezoelectric accelerometers with a sensitivity of 10V/g were used. The signals were recorded over a period of 50 minutes, sampled with a frequency of 100 Hz. To process the data Artemis Extractor was used. Spectral densities were calculated and compared with the spectra obtained by classical modal analysis. For the identification of resonant frequencies, corresponding mode shapes and damping values the enhanced FDD technique was applied.

![Figure 12. Singular values of the spectral density matrices and selected modes using Artemis Extractor, investigations on ship type B, only ambient excitation of wind and waves](image)
The results obtained by operational modal analysis using ambient excitation only are showing a good agreement compared with the results of classical modal analysis. Especially the identification of modes in the lower frequency range is improved. Difficulties appear investigating coupled modes. Because of the restricted number of measurement points only two essential mode shapes exist in the frequency range inspected. Since in the FDD modes are selected by MAC value, the identification is difficult.

![Frequency Domain Decomposition - Peak Picking](image)

Figure 13. Singular values of the spectral density matrices and selected modes using *Artemis Extractor*, investigations on ship type B, ambient excitation of wind and waves and additional harmonic excitation of running machines

Inspecting the singular values obtained from the measurement with additional harmonic excitation, the resonant peaks are showing low significance because of the wide dynamic range of the spectra. Nevertheless, inspecting several restricted frequency ranges, the modes become more significant. An identification of well spaced modes is possible. Coupled modes are also difficult to identify as already mentioned before.

Frequency lines of harmonic excitation can be identified clearly, because of their high magnitude and low damping values. The four stroke diesel engine was running with a rotational speed of 916rpm, resulting in a main excitation frequency of 15.27Hz. Because of the four stroke process also the half order can be identified. Due to the fact that on ships under real working conditions an amount of aggregates are running and the main propeller excitation is acting, the identification is expected to become more difficult because of the large number of harmonic excitation frequencies and the high dynamic range of the signals measured.

5 Conclusion

Operational modal analysis is easier to use because of the lower organizational expense. Carrying out classical modal analysis, the transportation of the big exciter and the arrangement of crane assistance is very time consuming. The time expense of the measurement itself is similar for both techniques.

Using only a restricted measurement model the results obtained by operational modal analysis are comparable with these obtained by classical modal analysis. A comprehensive comparison of both techniques will be possible only when more orthogonal mode shapes can be identified using a more detailed measurement model with additional references at selected locations of the ship.
Despite the high excitation mass used, the input of energy into the system is limited. Although the excitation of a restricted frequency range by the choice of a well selected exciter tip is achievable, the excitation of the lowest frequencies (corresponding with first global mode shapes) is not possible. Applying operational modal analysis these lowest frequencies can be identified.

A comparison between measured and calculated responses that can be done easily is an advantage of classical modal analysis. To obtain scaled mode shapes using operational modal analysis, experimental investigations at different loading conditions of the ship might be a possible way.

Attention has to be paid applying operational modal analysis during real working conditions of ships when lots of harmonics are present. High amplitudes in the frequency lines of harmonic responses require a high dynamic range of transducers and sampling device. Due to the fact that classical modal analysis is not applicable under working conditions, there is no alternative to the use of operational modal analysis.

First investigations have shown that the application of operational modal analysis in the field of ship building is promising.

6 References
