THE SOCIETY OF NAVAL ARCHITECTS AND MARINE ENGINEERS
One World Trade Center, Suite 1369, New York, N.Y. 10048
Paper to be presented at the Ship Vibration Symposium
Arlington, Va., October 16-17, 1978

Dynamic Analysis of the Afterbody of a Ship—Toward: a Successful Correlation Between Analytical and Experimental Results

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ABSTRACT

Calculations for predicting propeller-induced ship vibrations at the design stage involve the determination of four essential factors: rigidity, mass, damping and excitation, which the paper reviews separately. It is concluded that, although complex structural models yield a good approximation of the rigidity as well as the structural mass of the ship, there is on the other hand a lack of fundamental knowledge of added mass, damping and excitation forces, and that this deficiency is being filled by simplified methods which are often questionable. However, where it is possible to determine the added masses with reasonable approximation, a mass-rigidity model of the structure will permit the reliable determination of the ship's natural frequencies and modes.

Also, difficulties inherent to any full-scale measurement are pointed out, and serve to partly explain poor correlations between calculations and measurements as have been recently reported in the literature.

It is concluded that prediction of the actual vibration behaviour of a ship under service conditions is not possible at present. However, the response of the structure to a single vertical force of given intensity applied at the proper location, for arbitrarily-chosen viscous damping, can simulate, within the approximations introduced, the results of a full-scale exciter test, and may allow potentially dangerous frequencies to be recognised so as to permit a reliable choice of the number of propeller blades.

The actual case of a 122,000 cubic meters LNG carrier is studied in detail, and serves to highlight some of the problems encountered at the computational as well as the experimental level.

INTRODUCTION

Vibration of ships is an ever-present problem, which has in the recent past received more and more attention. The increase in the flexibility of ship structures has been accompanied by an increase in the propulsion power, result in high vibrational levels with various adverse effects both upon the structure and equipment of the ship and upon the comfort of passengers and crew. Althoug much research has gone into the developm of mathematical models and the use of sophisticated computer programs for dyna analysis of these models, there are stil grey areas, the neglect of which can lea to disastrous errors in prediction.

It is the purpose of this paper to review the various assumptions introduce in analytical models used for the study of propeller-induced vibrations, and to show their influence on the final results of the calculation, but also to stress some of the difficulties inherent to any full-scale experimental verification. Imprecision in the measured data cannot be avoided due to various factors, and result in hardly convincing correlations, even for the most sophisticated mathemati models and the most carefully planned and recorded experiments.

Progress in this domain is dependent on a better knowledge of all the assumpti involved both for the analytical model an the experimental apparatus. The case of a 122,000 cubic meters LNG carrier, for which both extensive calculations and measurements have been conducted, serves to illuminate the above remarks.

THE MATHEMATICAL MODEL

The dynamic analysis of any linear elastic structure, be it represented by a highly sophisticated finite element model or a simple spring-mass system, generally consists of two steps.

-First, the eigenvalues and corresponding eigenmodes of the system are calculated from the sole knowledge of the stiffness and inertia (mass) characteristics of the system, since it is common (and justified) practice to neglect the effect of damping on the free vibration behaviour within the assumption of linearity.

-The response of the system to any given excitation is then computed using the mode superposition procedure, requiring a knowledge of the exciting forces as well as of the damping characteristics of the system.

Only in case one follows the above steps, which involve a knowledge of the stiffness, inertia, damping characteristics of the system as well as of the exciting forces applied to it, can one attach a value to the maximum displacement or acceleration of a given point of the structure under service conditions.

On the other hand, knowledge of the stiffness and inertia characteristics of the system is sufficient to obtain the eigenfrequencies and eigenmodes of the structure and to reasonably simulate the results of an exciter test covering any given frequency range, provided that reasonable values for the damping factor in each mode have been assumed, and taking into account the fact that the magnitude and direction of the exciting force is in such a case known with a good precision.

We will show in what follows that the latter approach is the only one to be reasonably undertaken considering our present knowledge of the four factors: stiffness, inertia, damping and exciting forces, which we will now study separately.

1. Stiffness

During the recent past, the finite element method of analysis has rapidly become a very popular technique for the computer solution of complex problems in structural mechanics. Basically, the method can be understood as an extension of earlier established analysis techniques, in which a structure is represented as an assemblage of discrete truss and beam elements. The same matrix algebra procedures are used, but instead of truss and beam members, finite elements are employed to represent regions of plane stress, plane strain, axisymmetric, three-dimensional, plate or shell behaviour.

A three-dimensional finite element model of the complete ship structure usually consists of membrane elements, and involves a considerable extent of structural lumping to account for the extreme complexity of the actual ship structure. Examples of models used in static analysis are presented by Roeren (1), together with an extensive literature survey of pioneering papers up to 1969. More recently, models used in dynamic analysis may be found in the papers by Volcy et al. (2), Johannesen et al. (3), Risse (4), Kavlie and Aasjord (5), among others, some of these authors presenting extremely elaborate models involving thousands of degrees of freedom. A common feature to all the models used is that the aft part of the ship is represented using a fine mesh, which gets progressively coarser, the fore part being represented either by a simple beam or a crude three-dimensional prismatic model.

Increasing use of data generation procedures linked to sophisticated graphical devices has permitted to considerably reduce the actual time involved in building the model as well as to diminish the risk of errors, leading to ever precise models. It is therefore only fair to state that, at the present stage, correctly evaluating the stiffness of a ship structure is only a matter of computational effort, and that most actual models yield quite satisfactory knowledge of the stiffness factor.

In the case of a 122,000 cubic meters LNG carrier, the complete ship structure has been idealised using a three-dimensional finite element model of the afterbody (aft peak, engine room and superstructures), which is progressively connected to a two-dimensional model of the fore part through a transition part; the fore part is assumed to behave as a two-dimensional structure by reason of its slenderness.

The complete structure is made up of six sub-structures. Fig. 1 depicts models of the aft peak and of the engine room so as to give an idea of the fineness of the mesh used.

As we are solely interested in symmetrical vibrations, neglecting transversal and torsional modes, only one half of the ship has been considered, and appropriate boundary conditions prescribed in the centerline and for the model of the fore part to take the condition of symmetry into account.

The complete model consists of:
2193 elements (beams and membrane elements)
608 nodes
1717 degrees of freedom.

For the solution of the eigenvalue problem, the model has been reduced to 353 dynamic unknowns by static condensation techniques.

2. Mass

The inertia characteristics of the system consist of

i/the structural mass, which is automatically computed at the element level and then distributed among the nodes once the density and the cross-sectional area (beams) or equivalent thickness (stiffened or unstiffened plates) have been given. It is easy to check that the structural mass distribution alongship obtained in this manner coincides with that given by the hull mass curve,

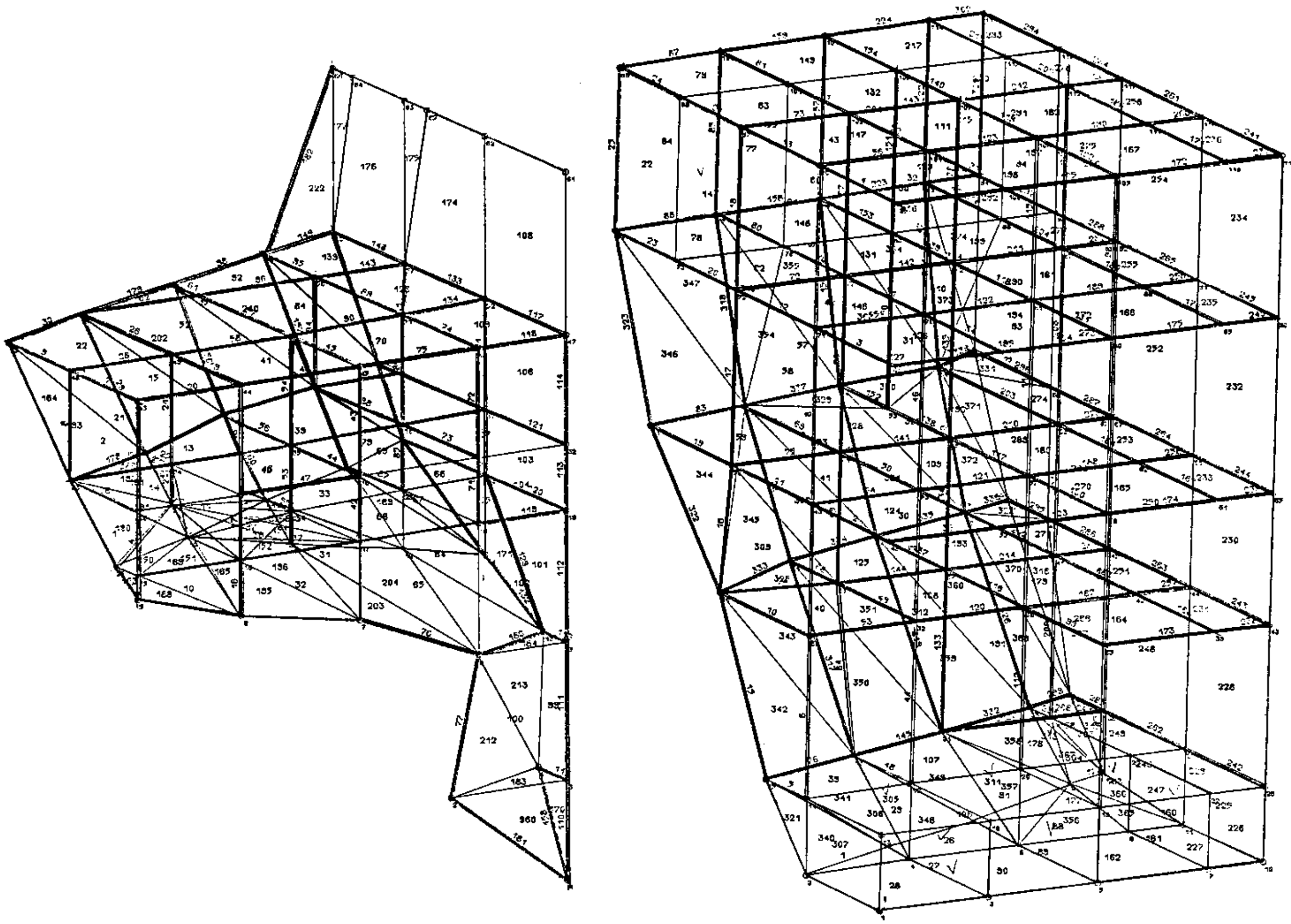


Fig. 1: Beam and plate model of the aft peak and of the engine room used for the dynamic analysis of a 122,000 cubic meters LNG carrier.

familiar to every naval architect.

ii/the mass of equipment and machinery, which is entered into the model in the form of point masses at the same location.

iii/the deadweight, consisting of cargo, fuel, water, stores, ballast and so on, also entered in the form of point (lumped) masses distributed among the nodes located on the boundaries of the compartments to which these items are located.

iv/the added (hydrodynamic) mass: the influence of the hydrodynamic pressure on the submerged surface of the vibrating hull can be taken into account through an additional mass term in the usual equations of motion of the structure, as was first shown by Lewis (6) and Lockwood Taylor (7). This mass term, which takes into account the effect of immersion, is known as the added (virtual) mass of the entrained water. Its importance should not be underestimated, as is too often the case.

The classical approach consists in dividing the ship hull into sections, each with uniform cross-sectional area, around which the flow of water is assumed to be two-dimensional. Corrections allowing for three-dimensional effects are then introduced through the so-called reduction factor J, defined as the ratio between the kinetic energy of the exact three-dimensional flow and the kinetic energy of the approximate two-dimensional flow, known only approximately and subjected to empirical corrections.

It has now become obvious (8) that such an approach, valid for the first few modes of vibration of the hull girder, for which the cross-section may safely be assumed to remain undeformed, is not consistent with the refined three-dimensional finite element structural models which are presently used to obtain the higher frequencies of vibrational modes of interest, in the range of the propeller blades frequency, in which deformation of the ship cross-section plays a paramount role.

A three-dimensional finite element discretization of the fluid domain as well appears to be necessary in order to obtain accurate values of the frequencies of vibration in the range of interest. By using fluid elements with curved boundaries so as to allow a perfect fit with the immersed part of the hull, such as those first described by Zienkiewicz and Newton (9), the connection between water and the most complex structural geometries is rather easy to describe. Since only one degree of freedom per node (the dynamic pressure) has to be considered, the computer time needed for the effective calculation of the added mass matrix is negligible in comparison with the time required for the complete structural analysis (10).

Fig. 2 shows the discretized model of the water surrounding the hull of the LNG carrier already considered. Twenty-node isoparametric finite elements have been used.

The complete model of the liquid domain includes 6 sub-domains and is entirely described using
372 twenty-node isoparametric fluid elements
2229 nodes with one degree of freedom per node
283 nodes in contact with the hull.

3. Damping

Contrarily to what has been said above concerning rigidity and mass, the present knowledge we have of damping phenomena arising in complex structures cannot be satisfactorily modelled, as there appears to be a nearly complete lack of knowledge concerning this topic.

In the case of a structure as complex as a ship, distinction should be made between

i/ the structural damping, which is commonly treated as hysteretic. Hysteresis of the structural joints is much higher than that of the ship hull material. The main sources of joint damping are working, slipping and fraying of overlapping connected elements. An exhaustive review of hull damping of internal origin may be found in a paper by Betts et *al.* (11).

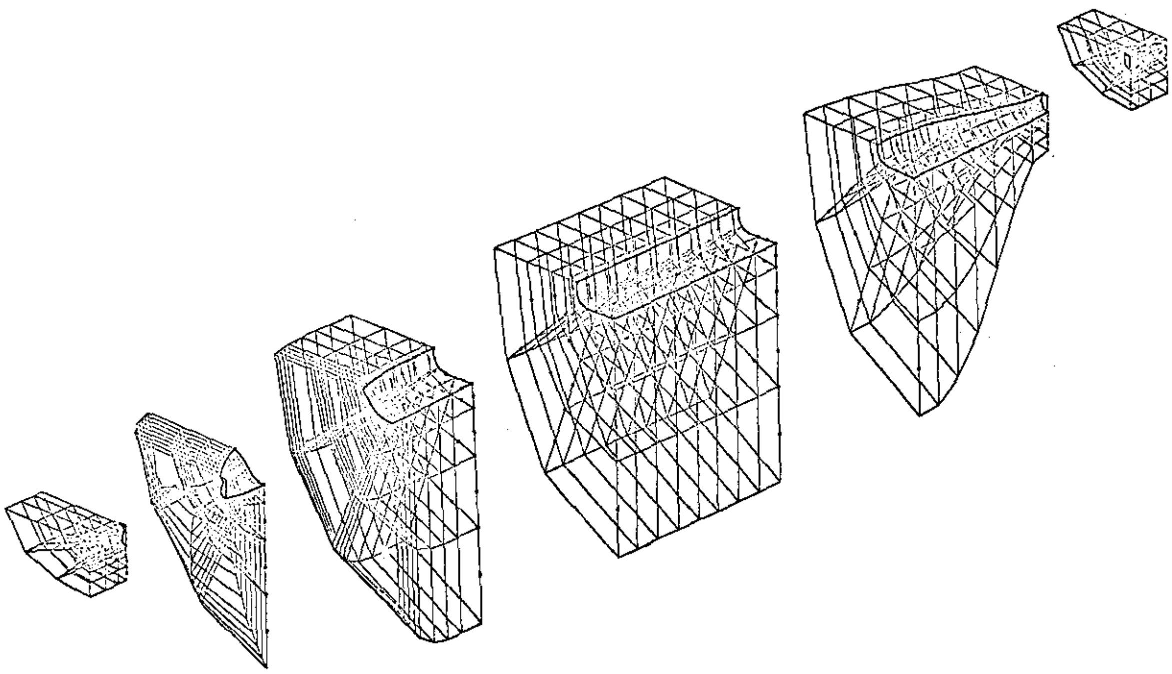


Fig. 2: Finite element discretization of the water surrounding a vibrating LNG carrier.

ii/the cargo damping, of which even less is known. The nature of the cargo has a major influence on the damping. Damping of solid cargo may be treated as hysteretic, whereas damping of liquid cargo is of viscous type.

iii/the hydrodynamic, or external, damping, which may in turn be conveniently divided into four components: water friction, resistance, radiation and generation of surface waves. All four components are considered to be of the viscous type. It is to be noted that frequency has a major influence on the last three components.

Quantitative introduction of damping in a model is therefore extremely difficult and whatever solution is finally retained can never pretend to be all that satisfying. The above remark serves as a justification of the current practice, now universally accepted.

Only global damping is considered, and assumed to be of linear viscous type. It is introduced in the model through the so-called damping matrix.

The complete set of equations of motion of the complete structure may now be written in matrix form as

$$[M]\ddot{q} + [C]\dot{q} + [K]q = Q(t) \qquad (1)$$

equation in which

$[M]$ is the mass matrix
$[C]$ is the damping matrix
$[K]$ is the stiffness matrix
q is the vector of (unknown) nodal displacements
$Q(t)$ is the vector of (known) applied nodal forces

Moreover, damping is assumed to be proportional, an assumption which results in decoupling the set of equations of motion (1) when the so-called normal coordinates r are introduced. System (1) may then be rewritten under the form of n independent equations, n being the total number of degrees of freedom of the structure,

$$\ddot{r}_i + 2\,\zeta_i\,\omega_i\,\dot{r}_i + \omega_i^2\,r_i = R_i(t)$$

$$i = 1, 2, \ldots n \qquad (2)$$

ζ_i is the damping coefficient for the i-th mode.

It can be shown that this assumption is valid if the damping matrix is a linear combination of the stiffness and mass matrix:

$$C = \alpha\,K + \beta\,M \qquad (3)$$

Hylarides (12) assumes a viscous damping which is proportional to the stiffness, i.e. lets β equal to zero in Eq. (3).

In usual practice, damping is characterized as a percentage of critical damping for each mode. This is obviously not a very good assumption, since it leads to a global value of damping which is the same everywhere in the structure. Moreover, most computations until now have considered the same value for this coefficient in each mode, when it should be obvious that modal damping cannot have the same value in vibrations of the hull girder or in local vibrations of the superstructure, for instance.

The simplifying assumptions described above may seem quite arbitrary. However, it should be pointed out that, under our present state of knowledge of damping phenomena, a more elaborate introduction of damping, e.g. via a full damping matrix obtained from experimental tests on similar ships, would seriously increase the computational effort without real comparable increase in the precision of the solution.

Such assumptions are the reflect of a compromise between a realistic model and a model which can be apprehended on a computational level, a situation arising in various modern disciplines. This philosophy may be endlessly discussed. We will only conclude that the current approach cannot entirely satisfy the engineer who is above all concerned about reliable results. This engineer will wish that more research of both fundamental and applied nature be undertaken so as to confirm the validity of the linear viscous model. In case it is proven to be a good assumption, it will then be the task of the researcher to characterize values of the damping coefficient in various modes for ships of a comparable nature, a task which has yet to be undertaken.

4. Excitation

Since the determination of the response of the structural system to a given excitation requires knowledge of both damping and exciting forces, it is apparent by now that a prediction of the behaviour of a ship under service conditions or even a simple exciter test, i.e. the computation of displacements and accelerations at specified points of the ship, is simply unrealistic, no matter to what degree of precision are the exciting forces known. On the other hand, it might also be said that comparison of the results of calculations and measurements yield by identification, for a given excitation, useful values of the modal damping coefficients which could be stored after they have been sorted according to ship type and mode characteristics, therefore constituting some sort of a catalog of damping coefficients which could be used in future calculations of similar ship types. An exciter test provides the ideal setting for such correlation. On the other hand, the situation is not quite so simple when actual forces encountered during service conditions are to be considered.

For large, modern ships, the propeller is the source of the main excitation forces. Since the propeller works in an uneven wakefield, thrust and torque variations are transmitted to shaft and bearings. At the same time, propeller blade cavitation causes large pressure fluctuations on the hull. Although of the same origin, the fluctuating loading and the fluctuating hull surface pressures are of quite different character and request different treatments for their computation as well as for their introduction into the model.

i/Dynamic loading on the propeller: its calculation implies that of the pressure distribution on the propeller blade during one full revolution for a given propeller geometry and a given hull wake field.

The traditional approach using the so-called lifting line theory has been modified into so-called lifting surface methods featuring a sounder mathematical basis, but still making extensive use of correction formulae and tables based on systematic model tests.

Although some of the assumptions introduced are quite restrictive, such methods give reasonably accurate values of the dynamic loading on the propeller. Integration of those yields the resulting forces and moments acting on the shafting system, which are introduced into the model at the nodes coinciding with the bearings.

ii/Propeller-induced hull surface pressure: the loading on the propeller is partly transmitted to the hull through the surrounding water. The main problem is that of finding the propeller-induced hull surface pressure under the restrictive assumptions made when calculating the loading on the propeller, since it had been assumed that the pressure distribution on a blade was not affected by the presence of hull and water surface. The introduction of solid boundaries and a free surface will therefore change the pressure field in a way to be precised in a quantitative manner.

Various analytical approaches have been proposed. Without going into details, it suffices here to reference the works of Breslin and Eng(13) and Vorus (14), both offering an elaborate mathematical treatment of the problem. However, a numerical solution of the complete problem, in view of the extreme complexity in shape of the aft part of a ship as well as of the nature of the mixed boundary conditions, appears to be very tedious but also very unreliable. This explains why the empirical approach via the so-called solid boundary factor S is still very popular with naval architects.

Although the amplitude of the pressure pulses decreases rapidly with the distance from the source, the pressure forces from the propeller may still be important even relatively far from the propeller plane. It should also be remembered that the pressure pulses propagate with the velocity of sound in water, and therefore that pressure pulses at points away from the propeller exhibit a phase shift with respect to the reference signal at the origin.

It may therefore be concluded that the present knowledge of propeller-induced hull pressures is far from being satisfactory. The precision cannot be compared with that obtained for the dynamic loading on the propeller.

Even if the dynamic pressure were known at each point of the surface of the hull, practical input of all this information at each node of the hull as well as processing the data to obtain the response of the structure would represent a formidable, if not prohibitive, task. The only way to obtain the value of nodal forces is to integrate the pressure over portions of a complex surface. The resulting dynamic loading will exhibit different phases in different points, and practically the only way to obtain the response of the structure is by surimposing the individual response to a single harmonic excitation applied successively to each degree of freedom of the hull, leading to fantastic computation times.

By contrast, the response to the propeller forces is easy to obtain. However, it has not been proven that the response to the propeller-induced hull surface pressure is negligible in comparison with the response to the dynamic loading on the propeller itself, and both contributions should be considered.

EXPERIMENTAL INVESTIGATIONS

The dynamic behaviour of a linear structure is completely described by the transfer function or response amplitude operator concept.

Measurement of the transfer function is well known practice. However, in the case of a ship, it involves particular aspects due to the large size and complexity of the structure considered.

These are:
-choice of the exciter and of the excitation type
-choice of exciter location
-choice of an experimental procedure insuring the scientific quality of measurements. Among various quality factors, attention has been mainly paid to experimental results reproductibility which has to be checked through both time and space variations.

Before any correlation study between theoretical and experimental data is undertaken, the experimental procedure should allow for satisfactory checking of basic theoretical assumptions.

Experimental procedure

i/Exciter and excitation

The large mass and stiffness of the structure under investigation requires a powerful exciter, the amplitude of the exciting force being preferably larger than 20 kN (roughly 2 tons) in the frequency range of interest.

Two different types of exciter have been used up to now by the shipyard in charge of measurement:

-hydraulic exciter: a large mass (5 tons) driven by a hydraulic jack up generates inertia forces. The motion of the mass is controlled by electronic devices through a servo valve.

Advantages of this system are numerous:
-easy control of the excitation using low frequency signal generator
-possibility of generating high excitation forces at very low frequency by increasing the amplitude of motion
-the time needed for a complete experiment can be shortened by using random, white noise type, excitation and by obtaining the results through statistical analysis by digital signal processing.

However, the sophistication of the equipment designed for laboratory work is not adapted to sea trials environment; furthermore, when this type of exciter is used at sea, ship motion, and mainly roll, can lead to difficulties as far as the guiding system of the mass is concerned.

-unbalanced-mass exciter: a constant-speed electric motor drives two unbalanced masses through an electromagnetic clutch. This thyristor controlled clutch is suitable for electronic remote control of rotation speed of the masses and therefore of excitation frequency.

The excitation force is basically sinusoidal, but a special remote control device allows slow frequency sweep over the whole frequency range.

The generated force is proportional to the square of the frequency, and the structure of the exciter is designed for a maximum load of 196 kN (roughly 20 tons). The maximum frequency at which a force of this magnitude is generated is function of the unbalanced masses. Two different sets of masses may be used, leading to two different maximum frequencies, that is 14 Hz for large masses, 25 Hz for small masses.

This heavy duty type of exciter was selected for the experimental analysis presented here though it is not suitable for very low frequency excitation due to the weakness (below 3 Hz) of the exciting force leading to a poor signal to noise ratio.

This type of exciter has been developed by Bureau Veritas; model E 20,000 was used for this experiment.

ii/Exciter location

In order to excite most of the calculated natural modes, it was decided to locate the exciter at the extreme aft end of poop deck, the excitation forces acting vertically. Besides, this location is adapted to a rough simulation of actual propeller-induced surface forces, and the experimental analysis is then expected to supply actual critical frequencies of the structure as far as propeller-excited vibrations are concerned.

In order to insure proper transmission of exciting forces to the structure, the exciter is welded on the deck. Local reinforcements were also decided in order to increase the local stiffness of the supporting structure and eliminate local vibrations of a parasitic nature.

iii/Pick-ups location

Above 3 Hz, most of the mode shapes may be reasonably identified by measuring the vertical deformation of engine room double bottom, of poop and upper deck, and the fore and aft deformation of deckhouse front bulkhead. Fig. 3 shows the location of the 14 selected pick-ups.

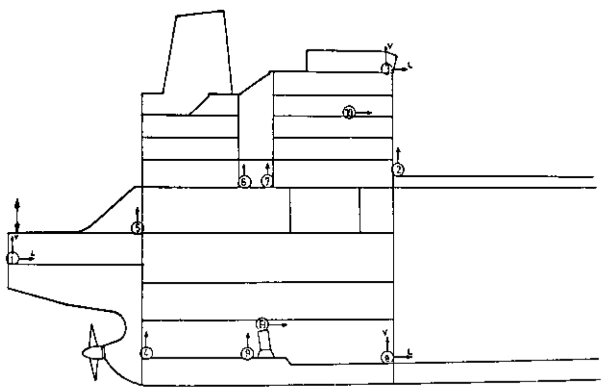


Fig. 3: Transducers location

iv/ Measuring equipment

Vibrations are detected by servo-accelerometers delivering a low impedance signal with a sensitivity of 100 mV/g (g = gravity acceleration 9.81 $m.s^{-2}$). The signals go through a multichannel amplifier that multiplies sensitivity by a factor 10 and their through low-pass filters with cut off frequency of 32 Hz in order to eliminate high frequency noise. A multi-channel magnetic-tape recorder stores resulting experimental data.

In order to get a phase and frequency reference of excitation force, a photo-cell delivers an electric pulse every time unbalanced masses of the exciter pass through a known position, the angular position of which is refered to excitation force maximum amplitude (= 0° phase reference).

While measuring, this rpm pulse signal is fed into a digital frequency meter that gives a precise measurement of excitation frequency when pure sinusoïdal testing, with step by step increase of frequency, is carried out.

v/ Measurement program

An extensive measurement program was decided in order to check theoretical calculation assumptions and reproductibility of experimental data through variations of excitation mode and experimental conditions.

First, in order to measure a proper influence of water added mass, all the experiments are to be carried out in open sea with a depth of water below keel larger than 50m according to theoretical calculation conditions.

Previous similar experimental investigations performed on the same ship in harbor alongside the quay and in open sea had already shown drastic change in transfer functions pattern.

- Checking of theoretical assumptions :

The main theoretical assumption is the linearity assumption which is equivalent to the assumption of viscous damping.

Though actual damping is presumably not linear, small valve of steel structure damping should confirm the rightness of this assumption.

In order to check structural linearity, the experimental analysis was repeated with two different sets of unbalanced mass, the ratio of exciting forces at the same frequency being 0.32. With the set of large mass used throughout the experimental investigation, exciting force reaches 20 tons at 14 Hz. With small mass, force is only 6.3 tons at 14 Hz.

For the linearity assumption to be acceptable the transfer function should be similar in both cases.

The hydroelasticity problem is formulated in such a way that there is no radiation condition at infinity. In other words, surface wave generation is neglected and fluid dynamics is only described by a single scalar pressure term.

As a consequence of this assumption, consistent with high frequency vibration compared to seakeeping vibration, there should be no effect of ship forward speed on added mass and so on structural transfer function. Experiments are then to be repeated at maximum speed just after having cut off shaft power, during the ship speeding down and once more with zero forward speed.

- Reproductibility of experimental data

The way experimental data reproductibility can be guaranted is twofold :

* Time variations : measuring process is to be repeated in similar environmental conditions

* Space variations : measurements are to be carried out on two sisterships which are refered to as ship 1 and ship 2 in the text.

Loading conditions of both ships are of premium importance for the results to be comparable.

For both ships, displacement and trim are similar:

Aft draft :10.5 m

Fore draft : 7.5 m

So that there is no change of water added mass distribution from ship 1 to ship 2.

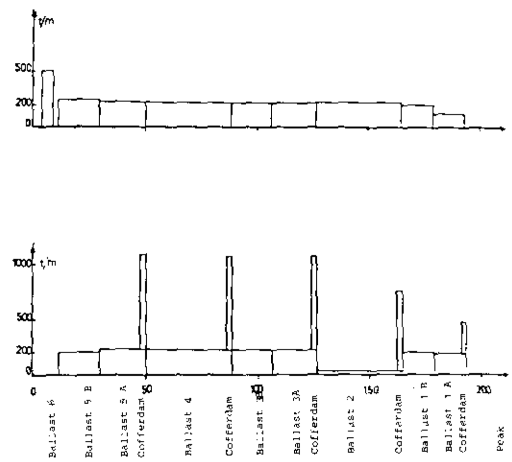


Fig. 4 : Water ballast weight distribution

But, as shown Fig. 4, the water ballast weight distribution are not identical :

In ship 1 all the cofferdams are empty but the aft ballast is partly filled. In ship 2 the aft ballast is empty but cofferdams are full. Nevertheless, the weight distribution aftward of machinery room front bulkhead is the same for both ships.

These differences should lead to small modifications of the results. However, a fundamental similarity is to be observed for theoretical prediction of ship vibration to make sense.

Ship 1 weight distribution corresponds exactly to theoretical calculation data.

- Variation of experimental procedure

Experimental investigationswere carried out during official sea trials and the time required by experiment was to be shortened as much as possible. This is the reason why sine sweep excitation mode was adopted. The automatic sweeping rate of the exciter is about 1.4 Hz per minute so that a complete speeding up or down of the exciter in the 3-14 Hz range would last about 8 minutes while a pure sinusoïdal excitation mode with step by step change in frequency requires more than an hour. However, such a step by step procedure was performed once in order to check that the type of excitation had no influence on the resulting transfer functions. As a matter of fact, due to presumably low damping coefficients (of the order of 1% of critical damping) it was felt that sweeping rate could be too fast for low frequency resonance vibration to build up at maximum stationary amplitude.

As the energy spread over the structure is not constant in the frequency range of interest, a possible influence of sweeping mode, decreasing or increasing of excitation frequency and force, was also to be checked.

Not to lengthen the sea trials program, the opportunity of carrying out experimental investigations during mooring trials with ship at anchor was taken. The calculated natural mode shapes showing small deformation in the fore part of the ship, this procedure was considered acceptable if compared to similar results obtained with ship drifting along.

vi/ Data processing

It was performed by a digital processing system programmed for random vibration analysis by Fourier transform.

The absolute value of transfer functions was obtained by dividing the amplitude of the signal at a considered frequency by the known amplitude of excitation force at this frequency. For sine sweep excitation the frequency resolution was that of the data processor i.e. 0.16 Hz.

For pure sinusoïdal excitation the resolution was selected aboard the ship at 0.1 Hz.

Between measuring period, the step by step change in frequency was carefully controlled thanks to the digital frequency meter.

The phase of transfer functions was obtained in two steps :

First the absolute phase of a reference signal with respect to excitation force was measured by analog means.

The vertical vibration at transom, close to the exciter,(pick up 1V),was selected as the reference point.

Then, the phase differences between all measuring points and the reference signal were automatically obtained by the data processor through cross spectrum calculations. The combination of both steps leads to the absolute phase of all points.

Rather than providing the reader with columns of numbers, it seemed better to select the significant results and to present them in graphical form.

Complete transfer functions are only given for two characteristic points : vertical vibration at transom (1V) and longitudinal vibration at wheelhouse deck (3L).

The behavior of other points is described at resonance by means of mode shape type representation.

Experimental data analysis

i/ Verification of theoretical assumption and influence of experimental conditions

The linearity of structural response is shown Fig. 5 which represents absolute value of ship 2 reference point transfer functions with the two different sets of unbalanced mass. Both curves are almost identical except in the 8.5 - 9 Hz region where slight discrepancies are noticeable. An explanation for these discrepancies will be proposed later.

As far as effect of forward speed on added mass distribution is concerned, the observed differences between experimental results obtained with an average speed of 12 knots and with the ship at rest are within experimental accuracy. In the propeller induced vibration frequency range, free surface influence on added mass distribution is then presumably negligible.

So is the influence of mooring lines when the ship is anchored.

Regarding the influence of excitation mode, sine sweep or pure sinusoïdal,on the results, it seems that the slight observed discrepancies can be explained by the difference in frequency resolution. For instance, the finer resolution of pure sinusoïdal excitation mode can lead to the separation of what was observed as a single resonance peak with sine sweep mode, into two adjacent peaks.

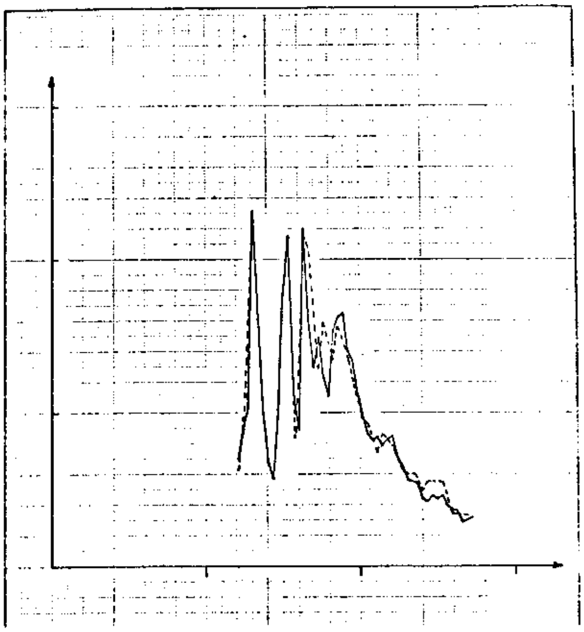


Fig. 5. Structural Linearity Check (Ship 2)

——— Large unbalanced mass

----- Small unbalanced mass

ii/ Transfer functions analysis

Fig. 6 and 7 show reference point (vertical vibration at transom) and wheelhouse (longitudinal vibration) transfer functions. Continuous lines correspond to ship 1 while dotted lines describe ship 2 dynamic behaviour.

As a general comment, it can be observed that the dynamic response of both ships are indeed comparable in amplitude and phase.

Before analysing thse curves in detail, the covered frequency range can be decomposed into four regions.

a- Below 4.25 Hz : due to the type of the exciter, a low frequency excitation was poor but yet a typical resonant figure is observed.

b- Between 4 .25 Hz and 6.25 Hz : both curves show no significant dynamic amplification.

c- Between 6 .25 Hz and 10. 25 Hz.: a rather high density of resonant peaks comes out.

d- Over 10. 25 Hz : the vertical stern

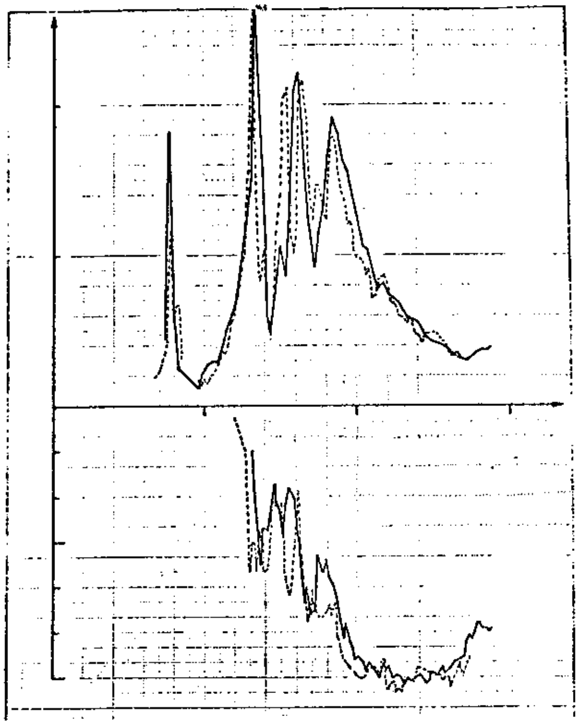


Fig. 6. Transfer Function
Vertical Vibration at Transom (point 1V)

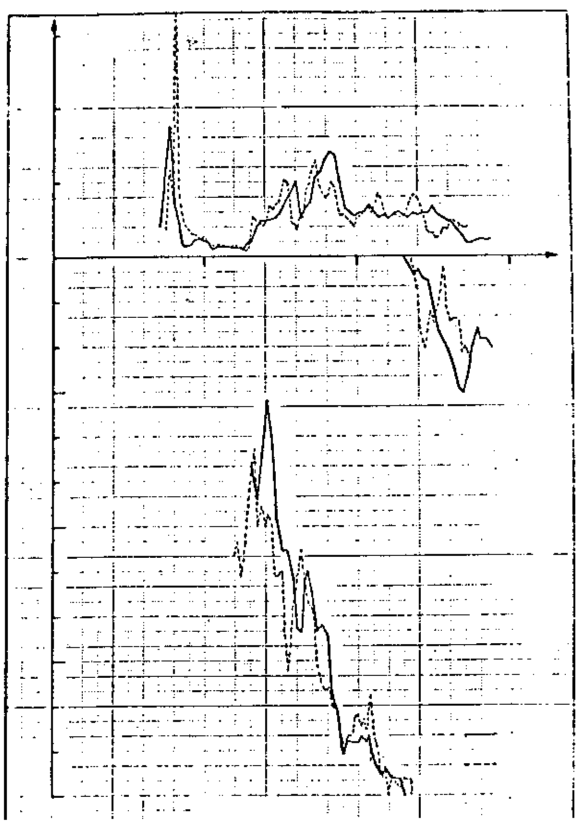


Fig. 7. Transfer Function
Fore & Aft Vibration at Wheelhouse Deck (Point 3L)

response decreases steadily while the phase tends to 180°. The wheelhouse response still shows some small variations but the mean phase keeps on decreasing regularly.

Ship 1 shows a resonant peak at 3.85 Hz. The corresponding dynamic deformation of the whole aft structure is given Fig. 8 where the upper sketch corresponds to the cosine part and the bottom one to the sine part. The vertical deformation of the hull is typical of a high order hull girder type mode with a mode located somewhere in the rear part of main deckhouse. The longitudinal motion of deckhouse is a geometrical consequence of hull deformation. The non negligible out of phase deformation shown on sine representation may be explained by a combination of two adjacent natural modes. This is confirmed by the analysis of ship 2 transfer function. As a matter of fact, ship 1 curves correspond to sine sweep excitation mode but ship 2 was investigated by means of pure sinusoidal excitation and the corresponding finer frequency resolution shows two separated peaks at 3.9 Hz and 4.1 Hz. As far as mode shapes are concerned, the former peak is similar to ship 1 deformation while the latter peak shows an important participation of deckhouse longitudinal motion (see Fig.9).

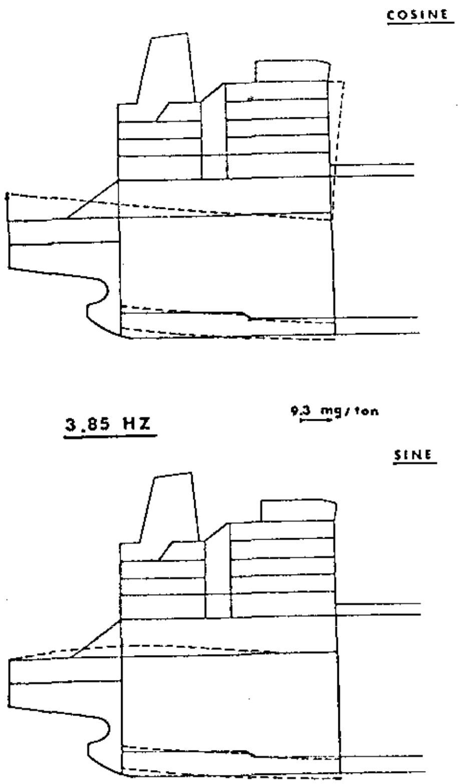


Fig. 8. Ship 1 - Structural Deformations

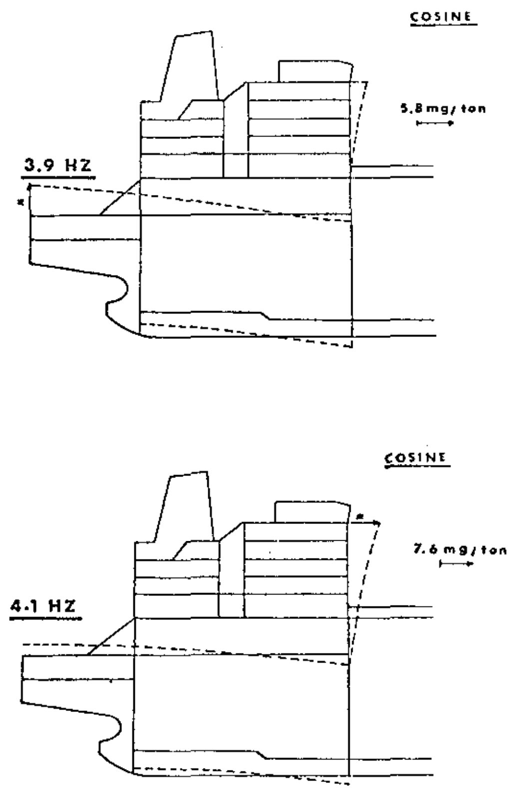


Fig. 9. Ship 2 - Structural Deformations

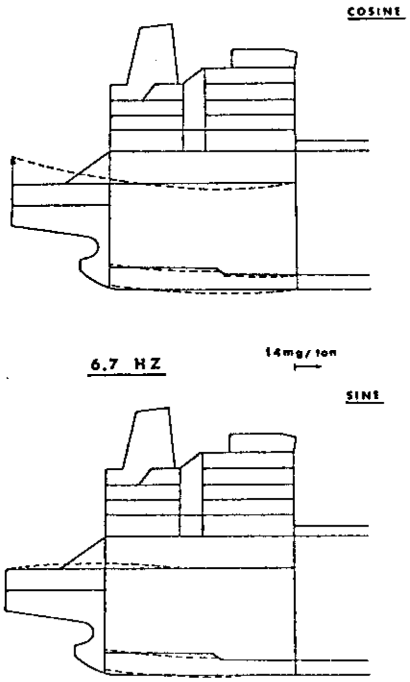
The difference in water ballast weight distribution between ship 1 and ship 2 have very little effect on the natural frequencies but the mode shapes show some variations. So do maximum amplitudes at resonance but the influence of excitation type on such a low frequency mode may be significant as said earlier in the text.

In the middle frequency region (6.25 - 10.25 Hz) several resonant peaks are shown on the graphs :

Ship 1	Ship 2
6.7 Hz	6.6 Hz
7.5 Hz	7.75 Hz
8.2 Hz	8.3 Hz
	8.7 Hz
9.3 Hz	9.2 Hz

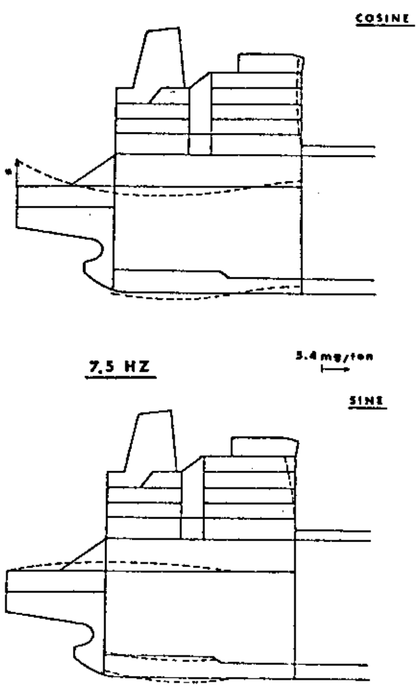
The modal deformation corresponding to ship 1 resonances are given Fig. 10 through 13.

The first three modes show hull girder type vertical deformation with two nodes moving aft as frequency increases.



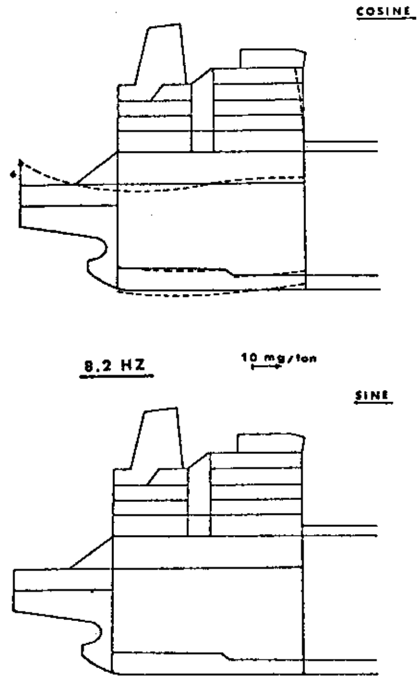
SHIP 1 Structural Déformations

Fig. 10



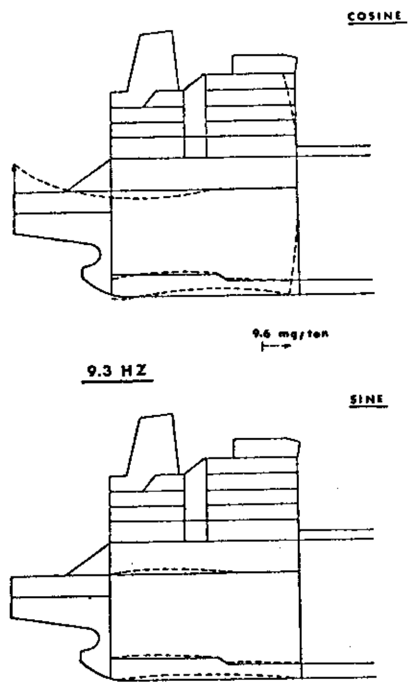
SHIP 1 Structural Deformations

Fig. 11



SHIP 1 Structural Deformations

Fig. 12



SHIP 1 Structural Déformations

Fig. 13

Ship 2 modal representations are very similar though maximum amplitudes may be different (8.2 Hz) and though natural frequencies are slightly shifted. An explanation for these differences is to be found in weight distribution variation from ship 1 to ship 2.

Ship 1 9.3 Hz resonance peak is rather wide and may well integrate several individual natural modes such as 8.7 and 9.2 Hz peaks observed on ship 2.

Fig. 11, 9.3 Hz modal representation shows a more intricate dynamic behavior with opposite phase of main deck and engine room double bottom deformation. Ship 2 shows similar behavior at 9.2 Hz while at 8.7 Hz the upperdeck deformation shows four nodes from deckhouse front bulkhead to transom but the phases between measuring points are far from the ideal 0°-180° figure of pure mode shapes.

The fast decrease of stern dynamic response after 9.3 Hz is translated in phase representation by a rather steady variation toward opposite phase between response and excitation. Yet the longitudinal weelhouse response still shows some peaks after 9.3 Hz. Such a dynamic behavior is typical of a local resonance of exciter supporting structure. The resonance presumably takes place around 8.8 Hz. As a matter of fact, it induces longer vertical motion of the exciter, modifying local damping characteristics in such a way that overall dynamic behavior may no longer be linear about 8.8 Hz. It must be recalled that such non linearities were observed on Fig. 3. The main consequence of this unexpected local resonance is to modify natural frequencies through dynamic coupling effect but mainly to drastically change the intensity of exciting force which is actually transmitted to the main structure. Because of the obviously small damping characteristics of this local resonance, it induces presumably an increase of actual excitation force in the 7 - 9 Hz frequency range but a fast decrease over 9 Hz.

As a consequence, experimental data must be analysed with carefulness in the high frequency range. We can simply state that there exist several natural modes over 9 Hz. Among these slightly noticeable modes, one can mainly distinguish :

Ship 1	Ship 2
10.4 Hz	10.7 Hz
12.5 Hz	11.9 Hz

Corresponding mode shapes clearly show shear deformation of deckhouse, while in the lower frequency range superstructure behaviour was mainly of the rigid body motion type.

Conclusions of experimental investigations

First, experimental data analysis has led to some theoretical assumptions verification.

The second aim of this important experimental campaign was to establish the basis of a correlation study with forced vibration calculations.

Because of local exciter supporting structure resonance which cannot be predicted by calculations, because of a not fine enough finite element representation of this local structure, the correlation will have to be studied within unexpected limits. Nevertheless, for the correlation to be satisfactory, the following precise characteristics will have to come out of theoretical analysis :

a - satisfactory correlation with 3.9 Hz dynamic amplification in both frequency and amplitude with possible influence of two adjacent modes at 3.85 and 4 Hz.

b - clear recognition of a 4.25 - 6 Hz frequency range with no definite dynamic amplification

c - in the 6.5 - 9.5 Hz range the correlation will be mainly focused on free vibration analysis (natural frequencies and mode shapes). Because of the above mentioned local resonance, calculated forced vibration response amplitude should be smaller than measured ones.

d - In the 10.4 - 13 Hz frequency range, free vibration analysis should give several natural modes with high deckhouse dynamic participation.

COMPARISON BETWEEN CALCULATIONS AND MEASUREMENTS

Using the mathematical model described in the first part, a free vibration analysis has been carried out by solving the conventional eigenvalue problem : 45 natural modes have been found between zero and 20 Hz and all those within the range of interest for the afterbody vibrations (6 - 15 Hz) are given in Table 1 with their description.

We can note that most of the modes are located in the afterbody with a more or less important participation of the different parts : aft peak, engine room and superstructures.

At this stage of the analysis (free vibrations), it is difficult to predict the most dangerous modes of vibration for the structure but we can compare the natural modes by using the concept of generalised mass ($m_i = \phi_i^T M \phi_i$), associated with mode ϕ_i and proportional to the kinetic energy of vibration. Table 1 shows that the mode at 10.253 Hz may be the most important after the (local) mode at 7.014 Hz.

From the set of natural modes, it is easy to obtain the frequency response of the structure by modal superposition procedure

TABLE 1 : Natural modes

Mode number	Frequency (Hz)	Description	Generalised mass (ton)
12	7.014	vertical local (ER)	164.
13	7.304	vertical local (TP) + vertical (FP)	8256.
14	7.404	vertical (AP+ER+FP)	5962.
15	8.280	vertical (AP+ER+FP)	3777.
16	8.616	vertical and longitudinal (AP+ER+S)	3696.
17	9.089	vertical (AP+ER+FP)	5306.
18	9.796	longitudinal (AP+ER+S)	3035.
19	10.029	vertical local (FP)	1539.
20	10.253	vertical (AP+ER)+ longitudinal (S)	511.
21	10.602	longitudinal (AP+ER+S)	797.
22	11.404	longitudinal (AP+ER+S)	1486.
23	11.798	vertical (AP+ER)	4110.
24	12.925	longitudinal (AP+ER+FP)	3550.
25	13.373	vertical local (ER)	2758.
26	13.472	vertical (FP)	2584.
27	13.628	longitudinal (S) + vertical (AP+ER)	1571.
28	14.190	" + "	326.
29	14.749	" + "	1478.

A.P. = Aft peak
E.R. = Engine room
S. = Superstructures
F.P. = Fore part
T.P. = Transition part

in order to simulate measurements with harmonic exciter. We have to introduce damping in the dynamic equations of motion : here, damping is assumed to be a fraction (1%) of the modal critical value. Such a relatively low value of damping has been chosen because of the ballast condition of the ship.

Fig. 14 - 15 show the frequency response at transom (in vertical direction) and at wheelhouse deck (in longitudinal direction).

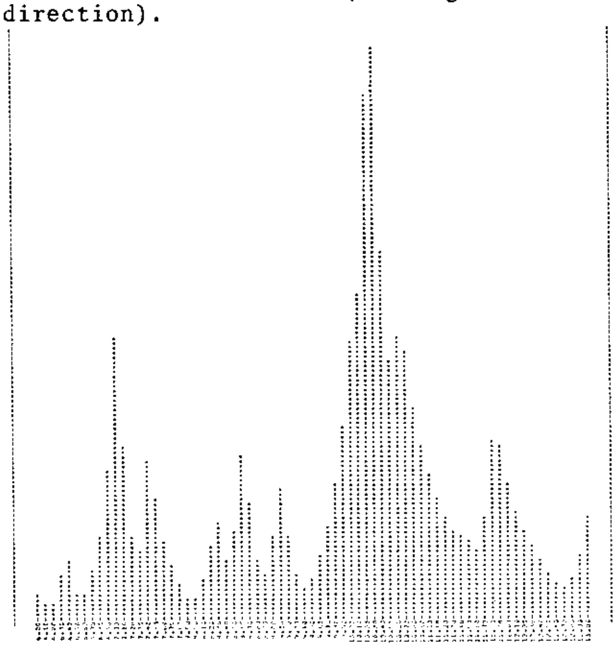


Fig. 14 - Vertical acceleration at transom

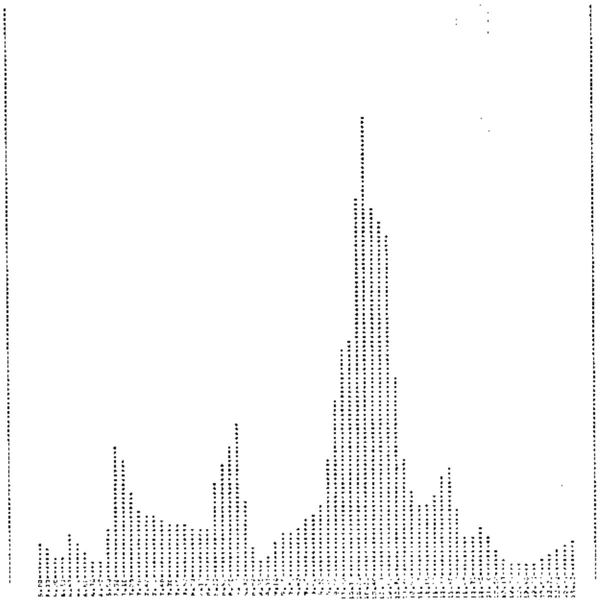


Fig. 15 - Longitudinal acceleration at wheelhouse deck

By correlation with measurements, the 3 principal resonances can be immediately distinguished and are listed in Table 2.

MEASUREMENTS	CALCULATIONS	
Frequency f_m (Hz)	Frequency f_c (Hz)	$(f_c-f_m)/f_m$ (%)
6.72	7.014	4.4 %
8.08	8.616	6.6 %
9.36	10.253	9.5 %

TABLE 2 - Frequency of the principal peaks, obtained by experiments and calculations.

With the exception of these 3 modes, it is quite difficult to identify the calculated modes with measurements; this difficulty can be explained by the fact that the measured response is a combination of modes and therefore is no longer a natural mode of vibration. So, we will restrict the correlation to the only 3 principal modes of the afterbody.

The difference between measured acceleration and calculated amplitude of response appears to be significant due to the lack of knowledge concerning damping.

However, we consider that it is now, and will be for a long time, quite impossible to predict the exact amplitude of the response but it is first of major importance to be able to detect the principal modes of vibration which may be eventually the most dangerous for the vibratory level of the ship.

CONCLUSIONS

The fluid finite element discretization used enables us to deal, more accurately than the conventional approach, with the dynamic fluid-structure interaction which plays a significant role in ship vibrations.

It is now possible to determine with a suitable accuracy the set of natural modes of a ship and particularly of the afterbody in a frequency range (5 - 15 Hz) which was not accessible by calculation using conventional added mass.

Introduction of arbitrary but realistic damping and unitary excitation in the dynamic equation of motion, permits to compare the relative influence of the natural modes on the frequency response and thus to detect the potentially dangerous modes of vibrations of the afterbody.

The first step (free vibrations) of a ship dynamic analysis appears to be numerically apprehendable in a satisfactory manner now. The next step (analysis of the vibratory level in service) still needs numerous correlation studies between calculations and measurements by structural identification in order to better characterize parameters such as damping : a better knowledge of damping should be the first step in this direction. In this respect, extensive measurements campaigns have been carried out at IRCN (Paradis (15), Orsero (16)) and at BSRA (Catley and Norris (17)), among other institutions.

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ACKNOWLEDGEMENTS

Research described in this paper is part of a project on Reduction of Propeller-Induced Vibrations directed by the Institut de Recherches de la Construction Navale and partially financed by the French Secrêtariat Général de la Marine Marchande. Cooperation from the Division Constructions Navales of Alsthom-Atlantique (the former Chantiers de l'Atlantique) for the experimental part during the sea trials is gratefully acknowledged.