THE INSTITUTE OF MARINE ENGINEERS

76 Mark Lane, London EC3R 7JN   Telephone: 01-4893   Telex: 886841

TRANSACTIONS (TM)

EUROPEAN-BUILT SEA BARGE CARRIERS
The Design, Machinery/Hull Interaction and Investigations into Vibratory Behaviour

G.C. Volcy, MSc, CEng, FIMarE, M. Baudin, C. Andreau, A. Manner and S. Seppala

read at 17.30 on Tuesday, 27 November, 1979

ISSN 0309–3948
Trans I Mar E (TM)
Vol 92 1980 Paper 5

European-Built Sea Barge Carriers; their Design, into Vibratory Behaviour

G.C. Volcy MSc CEng FIMarE

Chief Surveyor, Head of Applied Study and Research, Department of Bureau Veritas Marine Departments

M. Baudin and C. Andreau;

Engineers in the same Department

SYNOPSIS

The concept of the transport of lighters or barges on board ships—its operational feasibility confirmed on U.S.A.-built vessels—has been also adopted in Europe, and Valmet Shipyard has obtained the order for building two 37,850 tdw ships of this type for the U.S.S.R. Some considerations connected with the design and building of these ships and their machinery, which is the biggest in the world for this class are presented. In order to ensure trouble-free operation, extensive studies concerning static and vibratory interaction of machinery and hull have been undertaken, involving hydrodynamic excitations as well as treatment from static and dynamic point of view simultaneously, of twin-screw propulsive plant and hull steel-work. This called for FEM technique. Some results of the experimental researches and theoretical calculations are presented as well as their correlation with the measurements obtained on the first ship delivered, which proved vibration-free.

1. INTRODUCTION

The *Yulius Fuchik*, the first of two 37,850 tdw barge carriers, representing the second generation of *Seabee* vessels, was handed over to the Soviet Danube Shipping Co. at Valmet's Helsinki Shipyard on October 20, 1978. This was an important event in that it signalled the beginning of a new era in the development of barge carriers, a craft whose value has often been questioned. Fig. 1 shows the ship concerned.

The emergence of various barge carrying systems, some ten years ago, attracted special attention in Soviet shipping circles.

The planned opening of the Danube–Rhine waterway provided the impetus for a new Soviet-designed barge, the Danube-Sea (D-M) barge.

Before the Soviet officials' decision to award the nearly \$200M contract to Valmet Oy, the company had made a careful study of the various design alternatives available, including its own concepts based on the float-on/float-off method. The company and the customer then agreed that the American Lykes *Seabee* concept was the one best applicable to the project at hand. Due to the customer's specifications, however, the project produced an entirely new kind of ship, sharing with the three existing *Seabee* vessels only one essential feature, the cargo-handling method.

In spite of access to the experience of Lykes, Valmet had to spend nearly two years in hard design work for this exceptional type of vessel. Many other organizations, including Classification Societies, have been involved in the project. Model tests were made by the Technical Research Centre in Finland and the SSPA in Sweden; strength calculations in Norway; and vibration studies in Norway, Germany and France. Nevertheless, since the domestic content of the vessel is about 90 per cent, the *Yulius Fuchik* can be considered a genuine Finnish product.[1]

2. TECHNICAL PARTICULARS

The ship is a diesel-powered, twin-screw vessel of conventional welded steel construction. Its main dimensions are as follows:

Length overall	266.44 m
Length between perpendiculars	222.81 m
Breadth moulded	35.00 m
Depth moulded	22.95 m
Draught on construction waterline	9.00 m
Draught maximum	11.00 m
International tonnage	35,877.50 grt
Deadweight in salt water on maximum draught	35,850 tonnes
Cargo hold capacity (bales)	51,300 m^3
Free deck height, barge supports included/excluded	5.55/6.10 m
Heavy fuel oil tanks	4948 m^3
Diesel oil tanks	654 m^3
Lubrication oil tanks	472 m^3
Water ballast tanks	26,167 m^3
Fresh water tanks	247 m^3

The vessel is built in accordance with the requirements of the U.S.S.R. Register of Shipping for the notation KM☆ L3, A2 and it meets the load line, SOLAS and other international requirements.

The average service speed is 9.8 m/s (19 knots) with the main engines developing 16,500 kW. The main engine consumption of heavy fuel oil is 130 tonnes/day. The range is 12,000 nautical miles with stores sufficient for 40 days' consumption.

In the stern a cantilever hull extension has been added on each side of the vessel to support the hoisting machinery of the 2700 tonne capacity loading platform and to protect barge handling from heavy seas. Fig. 2 shows this arrangement and the lifting platform loaded with two barges.

The main engines and other machinery are remote-controlled from a central control room. The propulsion machinery is also remote-controlled from the bridge.

The propulsion machinery consists of four Finnish-made SEMT-Pielstick 16 PC 2.5 V medium-speed diesel engines (6,620 kW at 504 rev/min) in two pairs, coupled

Machinery/Hull Interaction: and Investigations

A. Manner and S. Seppala

Engineers from Valmet Oy Shipyard, Helsinki, Finland

via Renk reduction gears (520/138 rev/min) to two four-bladed KaMeWa controllable pitch propellers with diameters of 5500 mm. Due to the considerable distance between the propellers and the engine room there are six bearings on each shaft.

The vessel is equipped with a 900 kW electrically powered bow thruster, a controlled passive stabilizing fin, and a separate internal anti-rolling tank system.

Steam is generated by one oil-fired boiler with a capacity of 6.5 tonne/hour and by two exhaust gas boilers with a capacity of 2.5 tonne/hour each.

Electricity is generated by four Valmet B & W 6 S 28 Strömberg diesel alternators, each generating 1250 kVA. The power plant has been dimensioned to meet the needs of the barge hoisting and transporting system.

3. BARGE HANDLING AND HOISTING

The ship can accommodate a total of 26 D-M barges, each weighing a maximum of 1300 tonnes. The total loading capacity is thus 33,800 tonnes.

Containers can be carried in two different ways: in barge holds and on top of barge hatch covers, as well as on special container adapters.

The maximum number of container adapters on board being 82, the ship's theoretical container loading capacity is 1312 × 6.1 m (20 ft) containers plus 240 empty one

The barge handling system is designed to work at a minimum ambient temperature of −20°C; relative humidity of 0 – 100 per cent; maximum wave length in the loading area of 1.0 m; and maximum water flow velocity of 2.5 m/s.

The speed of the lifting platform is 0.7 m/min loaded and 1.4 m/min unloaded. The corresponding speeds of the transporter can be varied between 5.7 and 12 m/min.

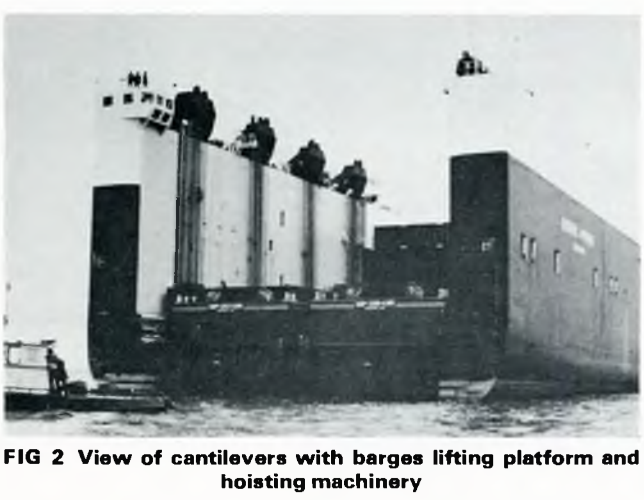


FIG 2 View of cantilevers with barges lifting platform and hoisting machinery

FIG 1 Sea-barge carrier m/s *Yulius Fuchik*

Based on these, the theoretical loading time is 13 hours.

A 450 kW (600 hp) pusher tug is needed to handle the barge during loading and unloading.

The cargo handling system (Fig. 2) was supplied by Kone Oy, and consists of a lifting platform; hoisting machinery; railed ramps connecting the decks to the lifting platform; a positioning system and two separate barge transporters which move along rails on the decks.

Barges can be loaded into or unloaded from the carrier one or two at a time. A pusher tug moves the barges between the hull cantilevers when the lifting platform has been submerged. After barge positioning, the operator lifts the platform until the barges are resting on the supports.

When rising further, the platform approaches the previously determined cargo deck, and mechanical limit switches stop it about 100 mm below the deck level. Height indicators are lowered and check the position of the platform in relation to the deck. The platform is then centred automatically by hydraulic cylinders and railed ramps are lowered for transport into the hold.

From this moment on, control is switched to the transporter operator on the deck. He has either a fixed control station or a removable radio control panel. He takes the rail transporters from the hold under the barges and limit switches stop the driving motors when positioning is correct.

The barge is raised from the elastic supports by means of jacks and is moved on to the actual deck. The height controls continuously indicate the platform inclinations due to moving of cargo and rope elasticity. They also regulate the hoisting machinery to keep the platform and crossing beam inclinations within permitted limits.

The lifting platform is locked for the sea voyage on special supports between the cantilevers, about 200 mm above the upper deck.

4. VIBRATIONS AND MACHINERY/ HULL INTERACTION

With the exception of the preliminaries, all vibration calculations were performed by the Applied Study and Research Department of Bureau Veritas. The Society's involvement started in the middle of 1976 when decisions concerning the number and rev/min of the shafts and the number of propeller blades had to be taken.

The delivery times offered by the manufacturers concerned being rather tight, and since only some of the drawings of the ship structure were then available, it was necessary to adopt the special, unconventional approach to vibrations and interaction problems established by Bureau Veritas[2,3].

This consists in the research and detuning of what has been called forced vibration resonators. Where these appear they considerably increase structural vibrations in their vicinity because of following dynamic amplification of their response (which can be considered as secondary excitation). They even affect an important part of the hull itself.

This approach has been adopted, tested[4,7–9] and has proved very efficient, especially for sophisticated, unconventional and not previously tested types of ships.

In these studies, it was also essential to take account of interaction between machinery and hull, as had been done in past studies by Bureau Veritas[5–10], in order to determine correct contact conditions between them.

5. RESEARCH AND DETUNING OF RESONATORS

5.1 Free vibrations of cantilevers

The freely overhanging cantilevers, together with platform and hoisting machinery, form an elastic system which, if excited at a resonant frequency and with dynamic amplification of its response, could greatly increase the vibration level of the aft part, and even of the whole ship.

For this reason the 3-D FEM model of the steel structure of the aft hold was used to simulate the cantilever boundary conditions and to obtain the most faithful spectrum of natural frequencies. Fig. 3 shows the corresponding elastodynamic model.

The ship being considered symmetrical, only the port side of the relevant steelwork has been modelled, in the form of 1765 elements, connected in 453 nodes, having 1290 dynamic degrees of freedom. Using this full model, the calculations of free vibrations gave a spectrum of 71 natural frequencies and corresponding symmetric and asymmetric mode-forms.

The greater part of these vibratory modes represents those of the aft part of the hull girder, but local vibratory modes were also found. Especially interesting in this context were the symmetric and asymmetric modes of cantilever vibration.

At the beginning, the normal rotational speed contemplated by the shipyard for the propeller shafting was 134 rev/min. The corresponding excitation frequencies were 8.93 Hz and 17.8 Hz for four-bladed propellers and 11.17 Hz and 22.34 Hz for five-bladed propellers.

Comparing these excitations with the spectrum of natural frequencies, resonances could be expected for four-bladed propellers at 8.66 Hz and 10.34 Hz; and for five-bladed propellers at 10.34 Hz and 11.85 Hz (symmetric) and 11.69 Hz (asymmetric mode).

In view of these results and in order to keep down vibration of the aft part and cantilevers, a four-bladed propeller was recommended, for the following reasons.

In general, the components of propeller forces and moments (acting in lateral direction and causing bearing forces) are much smaller for four- than for five-bladed propellers. Hence the response, even in resonance, of the cantilevers will be smaller.

Since the free vibrations at 8.66 Hz are the transverse vibrations of the lower part of the cantilevers, it would be also possible to detune this frequency by filling the tanks; the surrounding water would also exert a favourable damping effect. Fig. 3 also shows the mode forms corresponding to the frequency 8.66 Hz.

Forced vibration calculations gave a more precise response on the subject of vibratory level.

5.2 Rational alignment of shafting

Once the decision on the subject of propeller blades had been taken, it was necessary to investigate the possibility of the appearance of active resonators which may be constituted by propellers, propulsion shaftings and their bearings. These might act in lateral and longitudinal directions.

But it is obvious that, before vibratory phenomena can be investigated, it is of the utmost importance to determine correctly the contact conditions between the shafts and their bearings and the elastic steelwork on which the latter are mounted; as well as to understand the static properties of these elastic systems.

Hence, before studying the lateral vibrations of shafting it was necessary to investigate its interaction with its supports and arrive at what is known as "rational alignment". This means the judicious distribution of bearing reactions, taking account of the static, quasi-static and dynamic loadings in different operating conditions of the propulsion plant[3,5,6,10,11] which can cause interaction problems between the shafts and their bearings.

For the given ships, considering the long and flexible shafting (supported by ten supports or bearings) as well as the structure of the double bottom which can be considered rigid, it has been possible to dispense with detailed calculations of steelwork deformations due to loading conditions.

On the contrary, it has been necessary to study in detail the stiffness of the aft bush supporting struts. This was necessary, not only to determine the matrices of elasticity of aft the bush struts and white metal (necessary for the calculations of static contact conditions between tail shaft journal and white metal) but also to obtain realistic values for the calculations of lateral vibrations of line shafting (see 5.3 below).

In Fig. 4, sketch A shows the line shafting from tail to gearbox.

Sketch B indicates the results of calculations (calling for use of influence coefficients) of deformations and distribution of static reactions of the line shafting for cold and hot main gearing.

Sketches C to G represent the different practical steps of this alignment technique which uses the "sag-and-gap" coupling method. These drawings are self-explanatory.

The lower left part of sketch H shows the results of the final calculations. This is called elastic alignment because it takes into account, besides the flexibility of the shafting, also that of the bearing white metal[10,11].

These calculations make it possible, after preliminary parametric studies, to determine a convenient slope of white metal bore to ensure contact between the deformed journal and the white metal over the whole length of the bush.

From sketch E it can be deduced that maximal pressures do not exceed 80 bars, the maximum deformation of white metal being 15×10^{-6} m. Thanks to the assured contact over the whole length of the bush, the corresponding breadth of contact, for a 650 mm diameter shaft, is about 80 mm.

These contact conditions and results of alignment calculations (reaction values) have constituted the input data for the calculations of dynamic behaviour in the lateral direction of the tailshaft.

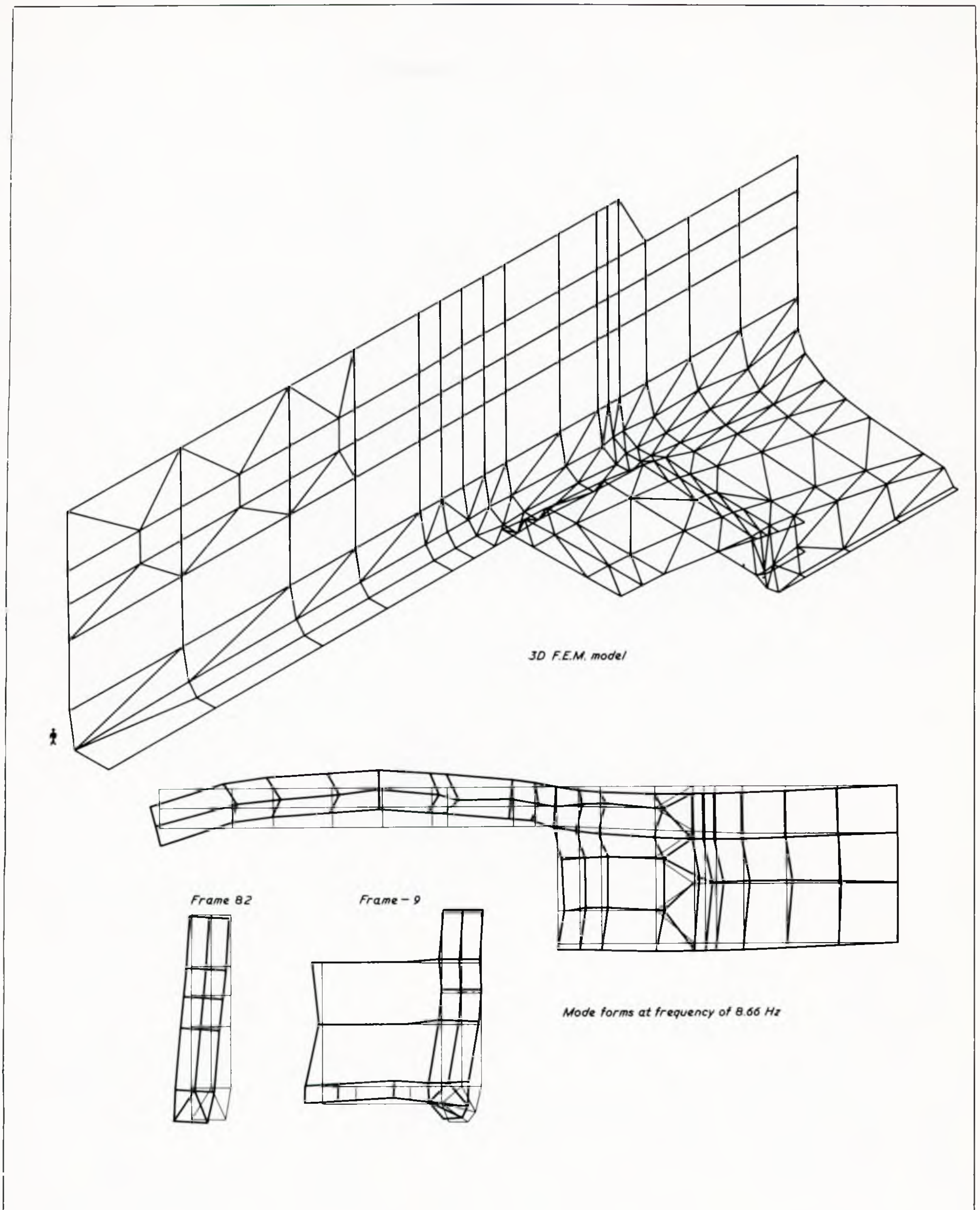


FIG 3 Free vibrations of partial model of cantilever

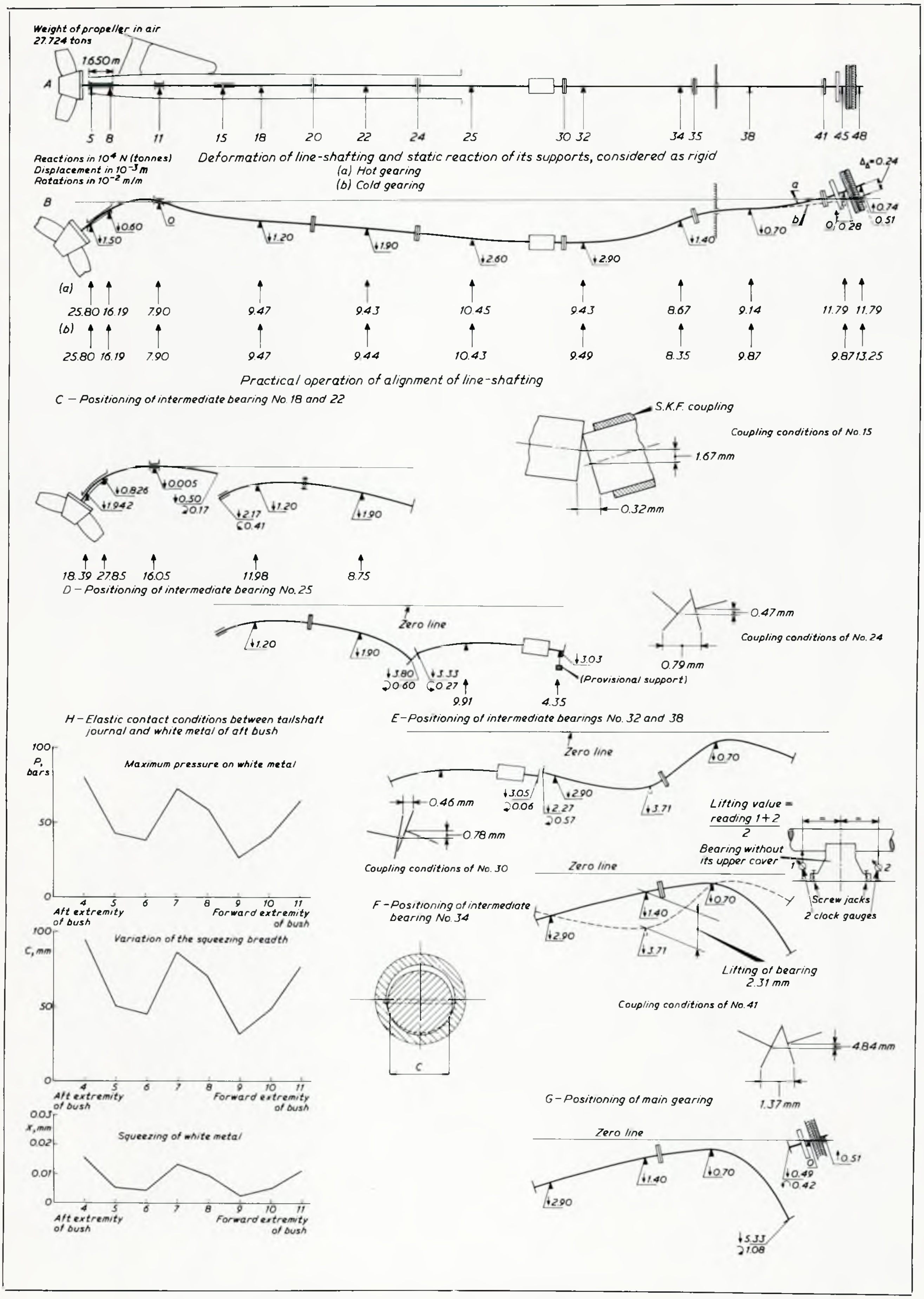
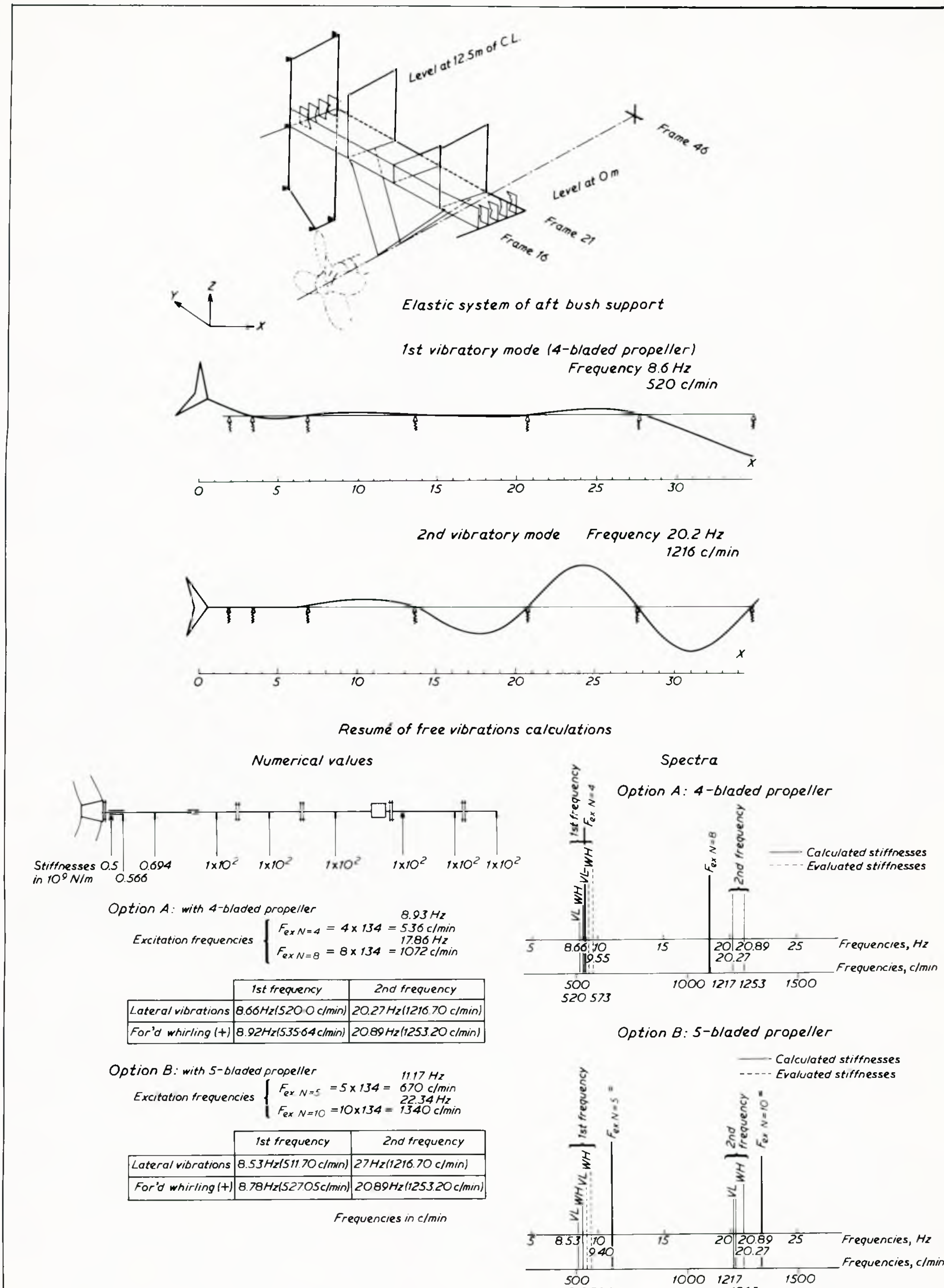


FIG 4 Rational and elastic alignment of line-shafting



Option A: with 4-bladed propeller

	1st frequency	2nd frequency
Lateral vibrations	8.66Hz(520.0 c/min)	20.27Hz(1216.70 c/min)
For'd whirling (+)	8.92Hz(535.64c/min)	20.89Hz(1253.20 c/min)

Option B: with 5-bladed propeller

	1st frequency	2nd frequency
Lateral vibrations	8.53Hz(511.70 c/min)	27Hz(1216.70 c/min)
For'd whirling (+)	8.78Hz(527.05c/min)	20.89Hz(1253.20 c/min)

FIG 5 Lateral and precessional (whirling) vibrations of line-shafting

5.3 Lateral and precessional (whirling) vibrations of shafting

In parallel with the studies mentioned in 5.1 and 5.2 and in order to investigate the possibility of a dynamic amplification of bearing forces due to resonant lateral vibrations of the tailshaft, in response to the excitations from the four-bladed propeller, a parametric study was made of the lateral and precessional (whirling) vibrations of the shafting. As the contact and alignment conditions were determined (in 6.2 above) the only parameters to be determined were the stiffness of the aft and forward extremities of the aft bush; and of the forward bush of the sterntube.

For the former calculations the values used were derived from Bureau Veritas experience are also indicated in Fig. 5. The corresponding natural frequencies (in Hz) of the two lowest modes were lateral vibrations 9.28 (1st) and 20.29 (2nd mode); and forward whirling 9.55 (1st) and 20.89 (2nd mode).

The first natural frequency, being relatively close to the excitation frequency of the four bladed propeller turning at 134 rev/min, it was necessary to calculate the stiffness of the tail shaft supports. For this purpose a 3-D FEM model was built, which is shown schematically in Fig. 5 (top).

The stiffness values of the supports, deduced from this model and also indicated on Fig. 5, were very similar to the calculated ones, and therefore the corresponding frequencies also agreed well. They are indicated on Fig. 5 for two shafts equipped with four- and five-bladed propellers.

From the analysis of these spectra it could be deduced that, from point of view of lateral and whirling natural frequencies, the five-bladed propeller was superior to the four-bladed one, chosen on the basis of the natural frequencies of the cantilevers (see 5.1).

Therefore it was of the utmost importance to investigate the possibilities of detuning the encountered resonance as well as to study thoroughly the influence on the shaft vibrations of external factors such as the four components of propeller forces and moments; and the corresponding characteristics of oil film.

During these detailed investigations the following parameters were varied: diameters of tail and intermediate shafts; successive shifting in forward and aft directions of the plummer bearing and even its elimination; modifications of contact and alignment conditions of line shafting.

Unfortunately, the limited practical solutions which could be adopted in this rather advanced stage of the construction of different components did not permit detuning of the resonant frequency to a very significant extent.

But during these investigations it has been found (thanks to the simultaneous oil film calculations for a skewed misaligned tailshaft[12]), that propeller forces and moments modify not only static contact conditions between journals and bushes, but also the matrix of elasticity of the corresponding oil film.

Also shown is the spectrum of these vibrations and the comparison with the excitation frequencies from two types of propellers, as well as the corresponding two lowest mode forms.

After mutual consultation between shipyard, engine builder, propeller and line shafting manufacturer and Bureau Veritas, it was decided to modify the diameter of tail shaft propeller and to reduce the rev/min of the line shafting to 125; in which case the fundamental blade frequency became 8.33 Hz.

The consecutive calculations of vibrations of line shafting have given the following results:

Frequencies in Hz	1st mode	2nd mode
Lateral vibrations	8.97	20.31
Forward whirling	9.23	20.98

Considering the relative vicinity between the 1st natural frequency and the one due to blade excitations, the importance of correct alignment calculations was obvious. Their realization, as described in 5.2, has avoided any supplementary increases of bearing forces due to the absence of whipping phenomena of the tail shaft.

5.4 Longitudinal vibrations of line shafting

The elastic system constituted by the propeller, line shafting, thrust bearing and its foundations, together with the steelwork of double bottom and machinery, may constitute a forced vibration resonator: hence the need for its investigation.

This was even more important due to the choice of a propeller with an even number of blades, which always causes bigger variations of torque and thrust than an odd number.

These investigations, like the previous ones, were in two steps: first, parametric studies, being of the stiffness of the assembly of the thrust block foundations as well as the equivalent mass of engine room, double bottom and machinery; then, detailed studies, with modifications and precalculated values of different parameters, especially related to the stiffness of thrust bearing foundations (always very difficult to obtain).

In the parametric studies the following variable parameters have been used:

Stiffness of steel-work of double bottom: $k_{DB} = 2$, 3 and 4×10^9 N/m

Equivalent mass of double bottom and machinery: $M_{EDB} = 5$, 10 and 15×10^4 Kg.

The definitive stiffness of the thrust bearing assembly and main-wheel, delivered by the Manufacturer, were:

$$K_{T.B.} = 7 \times 10^9 \text{ N/m}$$

$$K_{M.W.} = 74 \times 10^9 \text{ N/m}$$

The elasto-dynamic model used comprised 40 beam elements.

The results of parametric studies, showed the following values of frequencies:

Frequencies in Hz	1st mode	2nd mode
with four-bladed propeller	between 10.63 and 12.54	26.55
with five-bladed propeller	between 10.61 and 12.51	26.54

In view of these results, it is interesting to note that, from the point of view of resonance phenomena, and contrary to its effect on lateral vibrations, the four-bladed propeller was more convenient than the five-bladed: the first and second harmonics of blade excitations at 134 rev/min of the former, being 8.93 Hz and 17.86 Hz respectively, could not originate a resonant response. On the contrary, the first harmonic due to the five-bladed propeller, being 11.17 Hz, was in the resonant region of the fundamental natural frequency of the shafting's longitudinal vibrations.

This is a good example of how studies of free vibratory phenomena may give conflicting results and the final decision must often be a compromise, between the levels of excitations and response in forced vibrations.

The longitudinal vibrations effects of the finally adopted gearing and thrust bearing were calculated, starting with the stiffness of the thrust bearing foundations, their

Table I Calculated values of six components of propeller forces and moments

Phase angles according to: $F_x = \sum F_{xq} \cos(qZ\varphi - \alpha)$			Amplitudes kN or kNm	Phase angle α (degrees)	Dimensionless amplitudes
Thrust F_x: (kN)	Steady: (T)	q = 0	978.0	0	
	Unsteady:	= 1	7.6	57	F_{xq}/T = 0.8 %
		= 2	2.8	−157	0.3 %
Torque M_x: (kNm)	Steady: (Q)	q = 0	879.0	0	
	Unsteady:	= 1	4.5	71	M_{xq}/Q = 0.5 %
		= 2	2.3	162	0.3 %
Horizontal force F_y: (kN)	Steady:	q = 0	20.8	180	
	Unsteady:	= 1	2.1	−172	F_{yq}/T = 0.2 %
		= 2	1.2	− 26	0.1 %
Vertical force F_z: (kN)	Steady:	q = 0	33.1	0	
	Unsteady:	= 1	3.9	86	F_{zq}/T = 0.4 %
		= 2	0.9	−124	0.09 %
Horizontal bending M_y: (kNm)	Steady:	q = 0	114.0	180	
	Unsteady:	= 1	15.5	−175	$M_{yq}/0.3DT$ = 1.0 %
		= 2	4.6	− 10	0.3 %
Vertical bending M_z: (kNm)	Steady:	q = 0	141.0	0	
	Unsteady:	= 1	22.5	77	$M_{zq}/0.3DT$ = 1.4 %
		= 2	4.0	−118	0.2 %

Table II Calculated values of six components of surface forces

Phase angles according to: $F_x = \sum F_{xq} \cos(qZ\varphi - \alpha)$		Max. amplitudes	Mean amplitudes	Phase angle α (degrees)
F_x (kN)	q = 1	161	100	− 160
	= 2	–	56	− 20
F_y (kN)	q = 1	247	153	120
	= 2	–	86	160
F_z (kN)	q = 1	1058	656	120
	= 2	–	370	160
M_x (kNm)	q = 1	1054	653	120
	= 2	–	369	160
M_y (kNm)	q = 1	4452	2760	120
	= 2	–	1558	160
M_z (kNm)	q = 1	2992	1855	− 160
	= 2	–	1047	− 20

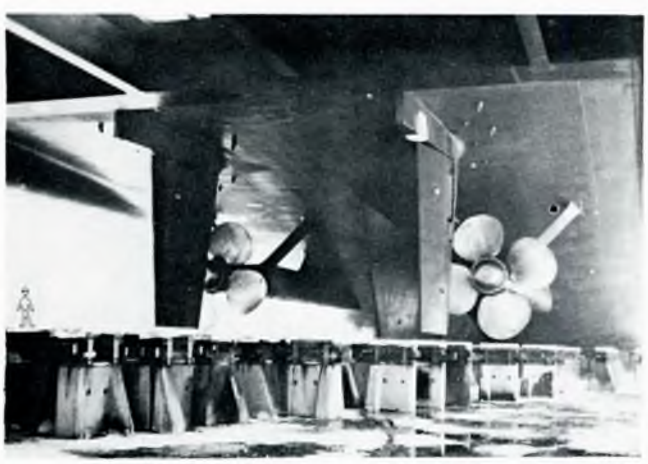


FIG 6 Arrangement of twin c.p. propellers, bossings and rudders

scantlings being meantime considerably increased, obtained from the 3-D FEM model of the double-bottom in the engine room and its environment (outside shell, webs, etc.).

This calculated value, with increased scantlings, was equal to $k_{DB} = 7.63 \times 10^9$ N/m.

The results of final calculations of free longitudinal vibrations of the modified assembly were:

1st mode: 13.18 Hz; 2nd mode: 29.88 Hz

When comparing them with the two lowest harmonics of blade frequencies of a four-bladed propeller at 125 rev/min which, owing to lateral vibrations of the shaft and cantilevers, are at 8.33 Hz and 16.66 Hz, it can be seen that no resonant response, and no subsequent dynamic amplification of longitudinal vibrations can be expected.

6. HYDRODYNAMIC EXCITATIONS

The Shipyard has carefully studied the hydrodynamic problems related to propulsion performance and excitations.

The shape of the ship was studied on model tests, performed in Swedish S.S.P.A. Laboratory in Gothenburg, and at the Ship Hydrodynamic Laboratory of the Technical Research Centre of Finland in Helsinki. Two alternatives (with, and without, a bulbous bow) have been studied. The bulbous variation resulted in a speed almost 0.5 knots higher.

A test series was also carried out in order to find the best solution for propeller bosses. The unsheltered shaft alternative was rejected at the outset. Based on the results of propulsion performance, wake and streamline tests, a 1.9 m diameter tube with strut supports from the hull, was selected.

Fig. 6 shows the afterbody of the ship in drydock, together with selected four-bladed propellers, their bossings with struts, and rudders.

Wakefield

In order to obtain the steady and unsteady components of propeller forces, necessary for vibration calculations, axial and tangential wakes were measured behind the fully loaded model (without fins) in the towing basin.

Propeller forces and moments

These calculations, done by S.S.P.A. programs, in principle, assume a quasi-steady state. Corrections based on two-dimensional unsteady wing theory were, however, made on both amplitudes and phase angles. The results of the calculations performed on an inward turning righthanded propeller (P/D = 1.116) in the portside wakefield are shown in Table I. The unsteady amplitudes are very small as expected from the harmonics of axial wakefield.

Hull surface forces and moments

As the exciting surface forces are distributed over the whole afterbody of the hull, the propeller pressure impacts were measured in the large cavitation tunnel of the S.S.P.A. at 15 points and the resulting forces and moments have to be obtained by integration, using a S.S.P.A. computer program.

The measured pressure impacts without fin were acceptable, presumably because of limited cavitation. These impacts, however, increased radically when a fin was added in both cavitating and non-cavitating conditions; hence, no fin was installed.

The results of the integration of pressure fluctuations correspond to hull surface forces and moments, used as input (together with propeller forces and moments) in the calculations of forced vibrations of machinery and hull, are indicated in Table II.

7. OVERALL CALCULATIONS OF MACHINERY AND HULL VIBRATIONS

7.13 3-D FEM model

The dynamic behaviour of the assembled ship and machinery was the subject of overall calculations. The first step consisted of building a faithful elasto-dynamic model. Since the machinery and ship are considered as symmetrical about the longitudinal plane, only the port half of the ship was modelled.

Almost the whole steelwork was represented by membrane and bar elements. The bossings and related outside shell were represented by triangular bending elements in order to simulate correctly the transmission of rotations.

The shafting up to the main gearing was represented by beam elements, connected to the steelwork in way of the supports by equivalent beam elements, and taking account of oil-film stiffness.

Main and auxiliary engines were represented by membranes, the corresponding moments of inertia of the engines being supplied by the manufacturer. The thrust block was considered as a beam grillage with infinite stiffness.

The virtual mass of water associated with the cantilevers and outside shell were taken into account as of directional masses, connected to the nodes of outside shell. The same principle has also been adopted for the masses of the main and auxiliary engines and rudders.

The ship as a whole was considered as freely supported beam.

Along the longitudinal plane of symmetry, two types of boundary conditions were introduced, allowing calculation of symmetric and asymmetric modes. The latter was necessary to assess the transverse vibrations of the superstructures and cantilevers.

This 3-D FEM model consisted of 4963 elements connected in 1490 nodes, having 3773 dynamic degrees of freedom. Its size can be appreciated from Figs. 7 and 8, which also show particular vibratory modes.

7.2 Free vibrations and correlation of results

Calculations of free vibrations were performed on a CDC 7600 computer, by the general finite element program ASKA, by the technique of "modal synthesis" recently developed by Bureau Veritas. This permits use of the results of the previously executed partial calculations (see Section 5).

Altogether 116 vibratory modes were calculated for symmetrical and asymmetrical boundary conditions. They have been determined in order to cover the range of first and second harmonics of propeller blade excitations, being considered predominant for the type of machinery in question.

The spectrum of natural frequencies starts at $f_1 = 0.5$ Hz (30 c/min) and ends at $f_{116} = 16.90$ Hz (1014 c/min) as shown in Fig. 9.

It is interesting to note that only the five lowest (three vertical and two transverse) mode forms of the spectrum are pure hull girder modes. The modes then change rapidly into coupled modes of substructures such as cantilevers, line shafting and bosses, superstructures, decks, etc.

Due to the limited length of this paper, attention will be centred only on the families of vibratory modes including cantilevers, and line shafting (lateral and longitudinal directions), previously treated by partial calculations.

Cantilevers

A great number of vibratory modes are found in the vertical, transverse and/or longitudinal vibrations of the cantilevers. Attention here will be centred only on the natural frequency spectrum within ±15 per cent of the excitation due to first and second harmonics or propeller blades.

For the fundamental excitating frequency, the natural frequencies are mainly between 7.68 Hz and 8.68 Hz. The two lowest modes are local ones, so their detuning was very simple. The other modes, on the contrary, arise from the assembly of the cantilevers, mainly transversely.

Fig. 7 shows the asymmetric mode of the overall free vibration at 8.68 Hz, with the transverse vibrations of cantilevers.

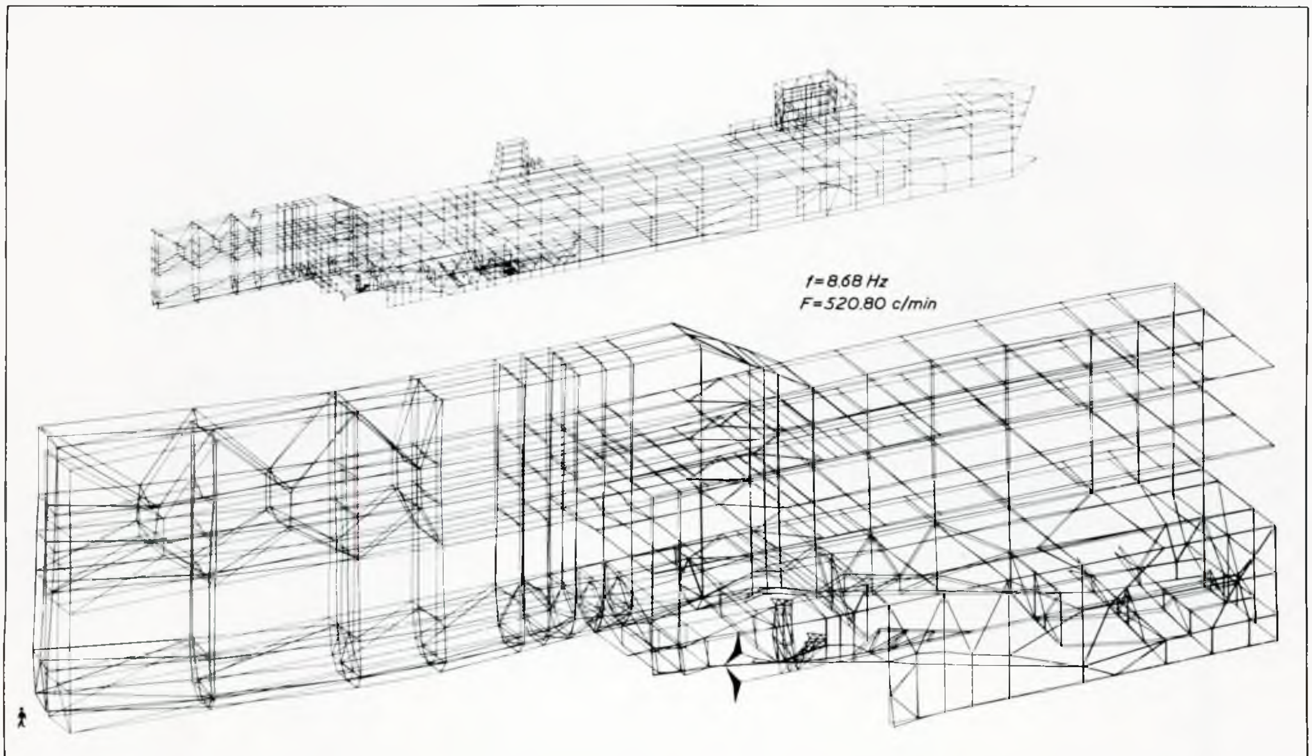


FIG 7 Asymmetric No. 65 mode form of global and free vibrations with important transverse vibrations of cantilever

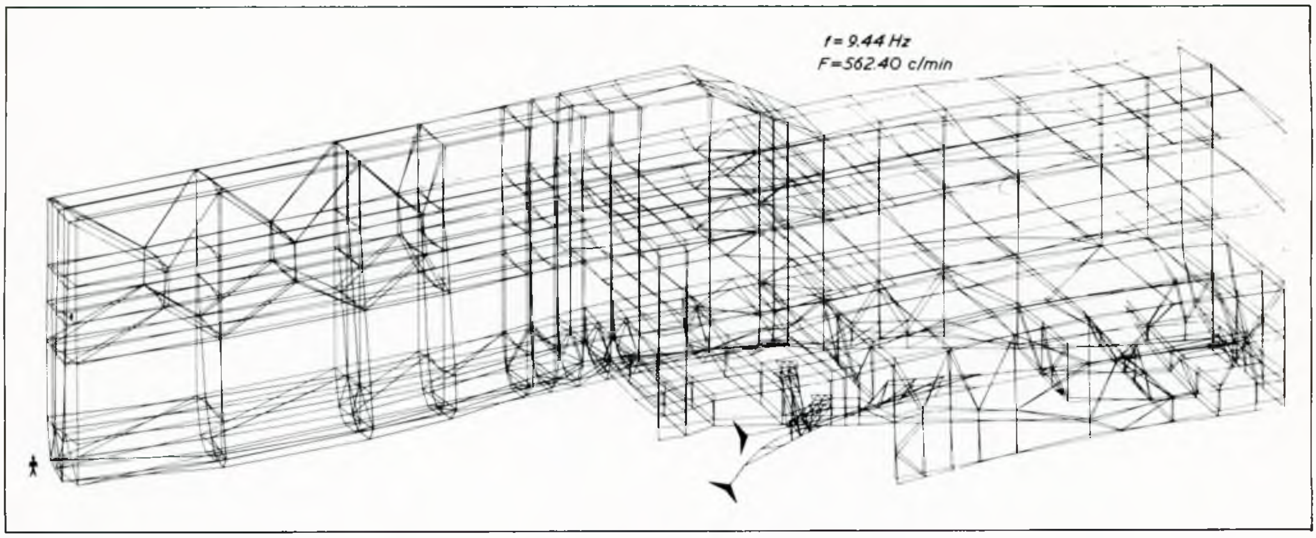


FIG 8 Symmetric No. 72 form of global free vibrations with important vertical vibrations of propeller shaft

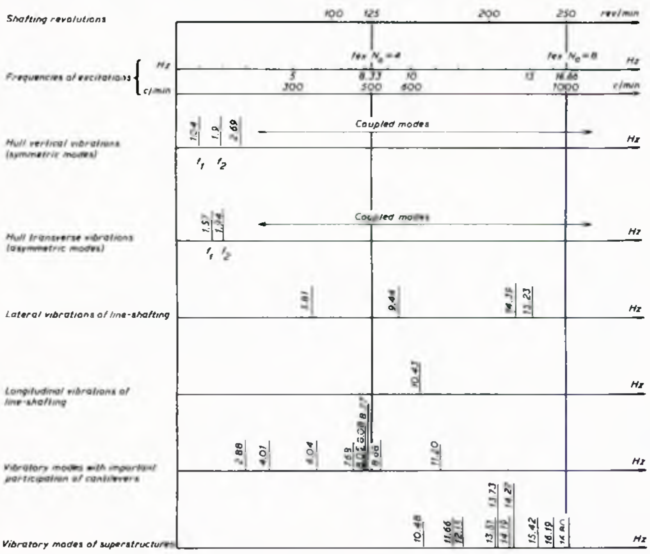


FIG 9 Spectrum of natural frequencies of global calculation of free vibrations of machinery

Table III Comparison with previous calculations

Vibrations frequencies (in Hz)		Partial	Overall
Vertical	1st mode	8.97	9.44
	2nd mode	20.31	
Longitudinal	1st mode	13.18	10.43
	2nd mode	29.88	

As these modes are transverse, where no important source of excitation is present, the danger of important resonance is not evident as the water also exerts a damping effect.

The detuning may consist of filling various watertight compartments of the cantilevers with water.

The comparison between results of partial and overall calculations of the lowest modes is shown below. A good degree of correlation can be seen, indicating the correct assessment of boundary conditions in the partial calculations.

Partial	7.91	8.18	8.66 Hz
Overall	8.05	8.27	8.68 Hz

Line shafting

Vertical or longitudinal modes are generated by coupling effects between line shafting and steelwork of the double bottom or hull girder. From the spectrum of frequencies shown in Fig. 9, the values at which they occur can be seen. Their comparison with previously executed partial calculations is shown in Table III.

A rather good correlation was noted for the frequencies of vertical vibrations, increasing the safety margin in respect of fundamental propeller blade excitations at 8.33 Hz.

The first natural frequency of longitudinal vibration, as deduced from the overall vibration mode, is lower than the one obtained from partial calculation. This is logical as the partial elastic system considered, which introduced unavoidable fully embedded conditions at engine-room bulkheads, had given excessive values for the thrust bearing foundations.

The value of the first harmonic of blade excitations being much lower, there is no danger of dynamic amplification due to thrust variations. The usefulness of the increased scantlings of the foundation was proved by this calculation.

7.3 Forced vibrations

Knowing the hydrodynamic excitations (those due to medium-speed engines could be neglected) and eigenvalues and vectors of the assembled ship, it was possible to calculate the forced vibrations. This was done by the same computer program as was used for free vibration calculations; with the same elasto-dynamic model on all degrees of freedom; using the modal response method without damping (to obtain the highest vibratory level and because no resonance of important assemblies had been encountered).

The dynamic loading conditions used were five components of propeller efforts (three forces and two moments, transverse and vertical) applied on the nodal point at the centre of gravity of the propeller; six components of hull surface efforts introduced on two nodes of the outside shell above the propeller. For all excitation efforts, the first and second harmonics were taken into account.

From the results, vibratory amplitudes of 0.01 to 0.03×10^{-3} m; and accelerations of 0.1 to 94×10^{-3} g were deduced. In fact, the calculated accelerations affecting superstructures and crew accommodation were in zone C of the Bureau Veritas criteria[3] which do not cause any discomfort for the crew.

The highest calculated vibrations were found at the aft extremity of the cantilevers. Even these did not fall into the Bureau Veritas region of unacceptable vibrations, at least in the vertical direction. And, since they do not affect crew accommodation, this calculated level was not considered as objectionable.

The calculated results for forced vibrations showed a good general dynamic behaviour of the ship assembly, the response decreasing as a function of distance from the principle source of excitation: the propeller. The reasons for this are the adoption of practical dispositions for detuning the main forced vibration resonators (active and passive), eliminating possible dynamic amplifications, and the relatively low excitations of hydrodynamic origin.

8. CORRELATION OF MEASUREMENTS WITH THE THEORY

The measurement of vibrations was carried out at the outfitting quay (vibrations exciter), and during sea trials when different types of excitations were operative, such as waves, anchor dropping, exciter and propulsion plant.

Hull girder

The lowest hull resonances were found during sea trials, excited either by anchor dropping or by the sea and the wind. These were recorded on several occasions and for different loading conditions. The closest to the calculated conditions were $T_{aft}=8.6$ m at sea depth 78 m; and $T_{for'd}=6.1$ m with main engines at rest, during anchor tests. Table IV gives the comparison.

In spite of some discrepancies which can be attributed to differences in loading conditions, the correlations were rather good.

Cantilevers

The natural frequency of the cantilevers has been experimentally investigated with the exciter operating, also during the sea trials. In the transverse direction it was 8.58 Hz.

The comparison between the partial and complete calculation results and the measured frequencies in Hz is shown below and indicates rather good correlation.

	Calculated from		Measured
	Partial system	Global system	
Lateral vibrations	8.66	8.68	8.58

Lateral and longitudinal vibrations of line shaft

The measurements of shaft vibrations were performed by Wärtsilä AB during sea trials, in the same loading conditions as those of hull girder during anchor tests (see above). A certain number of transducers were installed along the shaft, to determine the amplitudes and corresponding frequencies.

Natural frequencies were measured on several occasions, over the speed range of variable rev/min shafts. The most relevant results obtained are compared in Table V with the results of calculations.

In spite of unavoidable discrepancies there was a rather satisfactory degree of correlation.

It is worth noting that the measured values were obtained with one shaft engaged. The results of measurements with two shafts engaged showed, for both types of vibrations, a tendency to decrease the natural frequencies. We have insufficient reliable information to provide a satisfactory explanation of this phenomenon.

Table IV Measured results compared with calculated natural frequencies (Hz)

Vertical		Transverse	
Measurement	Calculation	Measurement	Calculation
1.05	1.04	1.52	1.57
1.85	1.90	2.65	2.46
2.67	2.69	4.07	4.01
3.62	3.59	5.12	5.16
4.52	4.17	6.27	6.04
5.42	5.49		
6.10	5.86		
7.02	6.89		

Table V Natural frequencies and calculated values

	Calculated from		
Fundamental natural frequencies in Hz	Partial system	Global system	Measured
Vertical	8.97	9.44	9.50
Longitudinal	13.18	10.43	12.00

Overall vibratory level

During sea trials, and in parallel with the measurement reported above of the free vibrations of machinery and hull, the response in terms of forced vibrations of the same assembly was also measured by Wärtsilä AB to verify the ship's conformity to specification.

The overall vibration level was found very satisfactory, being compared by the owner's representative to one of "a passenger ship". As expected, since damping effects were neglected in the calculations, the vibration level recorded on the ship and on machinery was lower than the calculated one.

It must be admitted that the correlation between calculated and measured response in forced vibrations was much less satisfactory than those encountered during previous analogue studies by Bureau Veritas[5,7,8]. The level of hydrodynamic excitations was particularly low, hence the inherent possibility of higher discrepancies between low vibratory amplitudes and the acceleration measured and calculated.

Also there were probably bigger differences than expected between actual and calculated hydrodynamic excitations (propeller forces and moments), in the present case than on the previously measured models. This has been mainly the case in Bureau Veritas' previous experience. It is also endorsed by the fact that measurements and calculations of free vibrations done with the same computer program and the same elasto-dynamic model had shown good correlation previously. Consequently, if the correlation in forced vibrations is less satisfactory, this can only be attributed to the new input data introduced into the previously tested and reliable calculations.

9. CONCLUDING REMARKS

It is extremely difficult to present, in a limited paper, extensive considerations and details related to the complex problems of new sophisticated types of ships such as the sea-barge carriers.

In this paper, we have attempted to show, by a concrete example, the rather difficult task facing the shipyard in the building of such ships. There are many conflicting requirements but, by means of rational approach, correct solutions can be found.

The existing know-how, together with today's available computer soft- and hard-ware (modal synthesis included) and experimental technique, and the adoption of a modern integrated treatment of interaction problems from the static and dynamic points of view (simultaneously for machinery and hull) together the investigation (and subsequent detuning) of forced vibration resonators, permits the building of even the most complex and powerful ships. At the same time the comfort of the crew can be ensured. Some of the lessons that can be drawn from this study are presented below.

As it could be seen from conflicting requirements regarding the number of propeller blades, the most convenient time to study vibratory behaviour of cantilevers and vibrations of shafts is at the earliest stage of the project when design options are still open. The parametric, and subsequent detailed studies related to resonators avoid the shipyard and its suppliers later calling for inconvenient compromises with the attendant risks of delayed delivery of sub-assemblies; but allows them to look for the best solution in the first place.

In order to keep vibrations to the lowest level, it is important to detect and detune any possible resonators; but it is also of utmost importance to keep the level of excitations as low as possible. Installation of the fins does not always decrease the pressure fluctuations. Calculation of forced vibrations is very useful where it is impossible, in practice, to detune a resonator, since it permits one to exercise one's judgement on the risk of running near, or at, the resonant frequency.

These studies also show how the different specialists (mechanical, structural, hydro-dynamics, etc.) complement one another; hence the need for a mutual understanding of their problems, in order to arrive at the most reasonable solution.

Alas, as highlighted by these studies, calculations alone cannot solve such problems but good practical experience is needed in this vibratory field, which as yet remains an art as well as a science.

10. REFERENCES

1) V. KOSKIVIRTA, V. HEIKKILAE, M. NIINI H. HARJUVAARA, 1978, "New Finnish Barge carriers for the U.S.S.R." *S.N.A.M.E. New York*
2) G. BOURCEAU and G.C. VOLCY, 1969, "Forced vibration resonators and free vibration of the hull" *Nouveautés Techniques Maritimes, Paris*
3) "Recommendations designed to limit the effects of vibrations on board ships", 1971, *Guidance Note N.I. 138 BM. 3*
4) K. RESTAD, G.C. VOLCY, H. GARNIER and J.C. MASSON, 1973, "Investigation on free and forced vibrations of an LNG tanker with overlapping propeller arrangement" *S.N.A.M.E. New York*
5) G.C. VOLCY, H. GARNIER and J.C. MASSON, 1974, "Chain of static and vibratory calculations of propulsive plants and engine rooms of ships" *A.T.M.A. Paris*
6) "Machinery Hull Interaction—Vibrations", 1976, *BUREAU VERITAS*
7) G.C. VOLCY and M. NAKAYAMA, 1977, "Studies leading to vibration and noise free ships" *V.A.N.C.E.*
8) G.C. VOLCY, 1977, "Vibrations of aft part of high output ships" *P.R.A.D.S.*
9) G.C. VOLCY, M. BAUDIN and P. MOREL, 1978, "Integrated treatment of static and vibratory behaviour of twin screw 553.000 tdw tankers" *R.I.N.A.*
10) G.C. VOLCY, 1978, "Interaction and compatibility between machinery and hull from a static and vibration point of view" *S.N.A.M.E.*
11) G.C. VOLCY, 1978, "Modern treatment of stern gear behaviour in view of difficulties encountered in the past" *I.S.M.E.*
12) G.C. VOLCY and R. VILLE, 1979, "Actual tail shaft behaviour on oil film taking account of propeller forces and moments" I.C.M.E.S. and I.Mar.E. Conference, London, *Trans I Mar E(C) Vol 91*

11. ACKNOWLEDGEMENTS

The authors wish to present their thanks to the Board of Directors of Valmet Oy, Helsinki Shipyard and Bureau Veritas for permission to publish the results of their work on the ships concerned.

They also wish to express their gratitude to representatives of all societies and organizations mentioned in this paper, for their help, understanding and collaboration during both studies and building.

The last words of thanks are addressed to all collaborators with the authors, who have assisted them during their work and in the preparation of the present paper.

Discussion

MR. T. W. BUNYAN, BSc, CEng, FIMarE (Pilgrim Engineering Developments Ltd) opened the discussion by commenting on the details of a most comprehensive research investigation, which had included exciter tests on the ship, complex mathematical models for structural vibration analysis, co-ordinated shafting alignment, etc. The trial results had confirmed that no significant problem was present because the shipyard had gone to great lengths to achieve an excellent design, which included all the best features proven in service to produce minimum vibration excitation of hull and machinery.

To quote some of those features in order of significance they were:

1) twin screw propulsion;
2) very large propeller aperture clearances with no inhibition of propeller design regarding cavitation performance;
3) long lengths of shaft lines;
4) insignificant vibratory torque from the twin-geared, 16-cylinder Pielstik engines.

While one could not quantitatively judge how good the end result would be without the results of the extensive vibration records made during the trials, the abstracts given in the paper had suggested that they had been very good.

As the wake variation at the propeller disc was the most important factor in assessing probable hull vibration performance, he would ask the authors to include the wake plot shown in one of the slides used in the presentation of the paper.

The one unusual feature had been the massive cantilever extension of the aft end of the ship. The shipyard's concern about those areas was understandable as, with the normal levels of excitation from the propellers, severe vibration could be expected under resonant conditions. He wondered if the authors would indicate whether the detuning tanks included in the cantilever construction had ever been used during trials or in service.

If the exciter tests included vibration measurements of the cantilevers made with and without detuning, as was most probably the case, he would ask the authors to add that valuable information.

Finally, he asked if the authors' firm rejection of the naked tailshaft, in preference to the closed-in shaft, had been based on measured results of the fall-off in propeller performance due to the greater wake disturbance of the naked shaft. Quantitative data on the problem was hard to come by, and inclusion of that information was requested.

MR. D. K. MARTYN, CEng, MIMarE (Lloyd's Register of Shipping) felt that the authors had presented a very interesting account of some of the extensive design studies carried out on a unique type of vessel.

It was some seven years since the first SeaBee, *Doctor Lykes,* and the first European-built LASH vessel, *Bilderdyk,* had entered service. Whilst many LASH vessels had subsequently been built, until the appearance of the vessels that had formed the subject of the authors' paper, there had been only the three SeaBees in service.

The barge-carrier was such an unusual type of vessel that it was a pity that the authors had not provided a comparison between the American and European designs. A figure showing the machinery layout of the authors' design would also have been appreciated.

The machinery installations of the American and European barge-carriers were quite different. The *Lykes* SeaBees had steam turbine main propulsion and drove a single five-bladed fixed pitch propeller at 120 rev/min. The European LASH vessels were engined with nine-cylinder, two-stroke diesels but, again, driving a single five-bladed fixed pitch propeller at virtually the same rev/min, i.e. 122 rev/min.

The authors had selected medium-speed diesels developing the same power as that installed in the American SeaBees. That selection of prime mover was quite understandable due to the low headroom available. The selection of twin screws was not so readily understood and had not been dealt with in the paper. It did, however, provide for less heavily-loaded propellers and better clearance between propellers and hull, together with less excitation. A good point to start from in combating vibration.

The selection of the number of propeller blades was inevitably a compromise. It was also, as pointed out by the authors, a decision that had to be made at an early stage in the design. However, a six-bladed propeller, had not appeared to have been considered at all in the paper and he was curious as to the reason.

In making the statement that "in general, four-bladed propellers produce lower bearing forces than five-bladed propellers", the authors had accepted that there would be exceptions. Lloyd's Register had certainly found that to be the case. What had led the authors to believe that the four-bladed propellers for those vessels would fall into the general category?

Fig. 6 of the paper had depicted the sterngear arrangement and had shown the bossings supported by struts which had appeared to subtend an angle of 90°, i.e. the same angle as between adjacent blades of the four-bladed propellers. Had the authors experienced any problems on other vessels where the strut angle had been the same as the blade angle?

Turning to the question of alignment, one effect of the trend towards flexible shafting systems of recent years had been the increase in bearing pressures. It was vital to ensure that plummer bearings had plenty in hand to counteract the effects of hull deformations.

The fear that was always at the back of one's mind was that failure of one plummer could lead to progressive failure throughout the main shaft line.

The authors had not indicated whether the main shaft bearings were white metal plummers or roller bearings. If they were the former, it would be appreciated if they would indicate the order of magnitude of the static bearing pressures.

For slow speed operation, it was important to ensure that hydrodynamic lubrication was maintained in the stern bearing. That was part of the purpose of carrying out an alignment study.

At slow speeds, the effect of thrust was very small and the dynamic alignment would follow closely the static condition. That was not the case of operation at the service speed where the effect of offset thrust could be significant. It was not clear in the authors' paper whether that effect had been taken into account, and their comments in that respect would be welcome.

When considering lateral vibration of the shafting, the authors had opted for a stiffer tailshaft, substantially in excess of the rule requirement he suspected. Had they considered dispensing with the struts and adopting a Grim type stern tube, as first fitted on the container ship *Hamburg Express* and since then on several other vessels, in order to detune the resonance well below the service speed?

On the question of longitudinal vibration of the shafting, one was left with a somewhat unsatisfactory situation.

On the basis of a finite element model of the engine room double bottom, the authors had increased the scantlings to provide greater stiffness and raise the fundamental frequency. Could they indicate how the calculated increase in stiffness and also the increase in frequency had been obtained?

Following that, an extensive analysis showed a drop in the fundamental frequency of some 26%. Whilst it was logical to expect a slight decrease in frequency if the original model was completely constrained at the tank top boundaries—a decrease in magnitude of 26% was somewhat alarming! Had the authors been able to examine more closely the possible reasons for that change in frequency?

In any study undertaken, the cost-effectiveness had to be borne in mind. Major design parameters had to be determined at an early stage—a point brought home in the authors' conclusions. In that case they had been established from a conventional examination of the shafting and local areas of the hull. In fact, design changes were made to the scantlings of both the shafting and double-bottom based on a conventional analysis.

Before a large-scale analysis of the hull and machinery inter-action could be tackled, the design must be almost finalized and might well be in an advanced stage of construction. In that particular case it would be interesting to learn which design parameters could have been changed by the time the overall study had been completed.

MR. C. F. W. EAMES, BSc (Stone Manganese Marine Ltd) said that the authors had given a most informative account of a major investigation to validate the design of unusual ships with novel features. The design was possibly the forerunner of many such ships that would link the Europa channel with major waterways in other parts of the world.

The twin-screw investigation could perhaps be compared with that for the single screw Maersk line "A" Class vessels described in a paper presented to the Institute two years previously by Langenburg and Andersson.*

In that twin-screw case, the wakefield, as would be expected, was a great deal easier. An estimate of $\frac{\Delta\omega}{1-\bar{\omega}}$ at say 0.8 R would, from Reference 1, be about 0.35 for these ships, but about 0.9 for the "A" Class vessels.

In a paper** presented to the North East Coast Institution Mr R. Rutherford had proposed a criterion for the prediction of propeller excited vibration, although it should more properly be termed wake induced vibration. That was based on $\frac{\Delta\omega}{1-\bar{\omega}}$ and a cavitation number σ_n, clearly a wake excitation factor. For those ships $\sigma_n \simeq 1.55$ and for the "A" Class about 1.65. On that basis, Mr Rutherford's criterion would indicate that those ships should be trouble-free, whereas the "A" Class would plot in a region where vibration might be a serious problem.

Furthermore, the wake plot in Reference 1 had shown that the wake gradients for those ships were much less steep near the hull than those for the "A" Class or for single-screw ships in general. Those ships did not, therefore, seem to present the same challenge for propeller/hull interaction effects as the "A" Class.

The unusual features of those ships, and in particular the cantilevers aft, had presented unknown, and possibly undesirable, influences in the vibration characteristics, but the ability to fill selected compartments in the cantilevers, in order to detune any undesirable vibration, must have relieved the situation to a large extent.

For a ship of that form, it would seem unlikely that fins should be needed and so it was not surprising to find that the results were better without fins.

Those responsible for that design were to be congratulated on the successful outcome of a highly sophisticated investigation, although the above remarks might have seemed to suggest that a sledge hammer had been used to crack a walnut. But he was sure that had not been the case, since Mr Volcy had explained very eloquently that it had been some walnut. Would the authors recommend a similar scale of investigation for every new type of large ship?

* Langenburg, H. and Andersson, G. O. 1977 "Design of a High Speed Single Screw Containership" *Trans I Mar E, Vol 89 Series A pp. 163-199*

** Rutherford R. 1979 "Aft End Designs" *North East Coast Institution of Engineers and Shipbuilders*

DR. S. ARCHER, CEng, FIMarE, commented that the paper was clearly another of Mr Volcy's many able contributions to the science and art of vibration engineering. It had reported the investigations, calculations and measurements with the high degree of thoroughness and attention to detail to which one had become accustomed from his team. Some might even suggest that, having regard to the many favourable factors inherent in the particular design of those ships and their machinery, Mr Volcy had quite a lot going for him and that, therefore, the application of a rather less sophisticated, time-consuming and costly approach might well have sufficed. On the other hand, the structural design of those special-purpose vessels had embodied a number of new and untried features and arrangements and, bearing in mind also the reputation of the owners for insistence upon the highest achievable standards, there was some justification for carrying out a more elaborate programme on that prototype ship.

One of the more obviously favourable design features had been the use of c.p. propellers, whereby it was much easier to "dodge" otherwise objectionable critical speeds by slight alterations of pitch and rev/min.

Another had been the more favourable wake variation with twin screws and the reduced effects of propeller forces and moments on hull vibration and shafting alignment, and thus upon the latter's lateral and whirling vibration characteristics. Could the authors confirm that draught variations for those ships in service would be relatively small? If that was so, it would again suggest that, unlike single screw tankers, there would be relatively little change in the propeller forces and moments with draught.

It had been noted that a double-strutted design of bossing had been adopted and clearly that would have an important influence upon the lateral stiffness of the stern tube and its bearings and, hence, upon the natural frequencies of lateral vibration. Could the authors indicate whether that feature was adopted for that reason or even, perhaps, partly so?

The subject of strutted bossings brought to mind some wartime experience with Lloyd's Register of Shipping on the old RMS *Andes,* which had suffered from severe lateral vibration of her twin bossings excited by her three-bladed propellers in a coupled mode with torsional vibration of the shafting. With riveted construction, the riveted joints had progressively loosened causing heavy leakage and reduced bending stiffness of the bossings as cantilevered appendages, eventually causing near-coincidence of bossing flexural and shafting torsional natural frequencies. The fatiguing effects were severe enough to cause cracking of the case steel spectacle frame on one side. The obvious and most effective cure was the fitting of single struts between the bossings and the keel.

The authors were to be congratulated on making available a valuable example of the application of refined vibration analysis and computer technology to a new, and hitherto untested, type of vessel.

MR G. H. SOLE, BSc, CEng, (Lloyd's Register of Shipping) said that the paper had represented a very considerable design and analytical effort on the part of the builders, Bureau Veritas and others, and had resulted in a successful design. He wished to comment particularly on the hydrodynamic excitation forces and the overall calculations of hull vibration and response.

With regard to the selection of the number of propeller blades (para 5.1), it would appear that the only dynamic forces considered with respect to the cantilevers and aft part were the propeller lateral forces and moments. In addition, symmetric and asymmetric forces might be created in a twin-screw ship by oscillating pressures acting on the hull surface, asymmetric when the two propellers were running out-of-phase, i.e. with the blades unsymmetrically disposed with respect to the ship's centre line. Although the peak amplitudes of the resultant asymmetric force (which would tend to excite transverse cantilever motions) would vary with time as the phase relationship between the two propellers continually changed, its magnitude might well be considerable, compared with the propeller lateral forces and moments, as those had appeared small from Table I. What was the opinion of the authors on that aspect?

Incidentally, Table II had shown integrated surface forces which had appeared very large. That would seem to contradict the text, unless there was a possible error in the Table. Also, the possibility of fitting fins had been mentioned several times, which was a fairly unusual consideration at the design stage for a twin-screw ship. What type of fins had been envisaged?

Most of the analytical techniques in the paper had been very similar to those reported in other papers by Mr Volcy, but he was interested in the application of the modal synthesis technique which, he believed, had not appeared in those previous papers. That technique was potentially very attractive and had been used for many years in the aviation industry. However, a great deal of care would have to be exercised to ensure that continuity of deformation would be maintained at the junctions between the sub-structures in order to maintain accurate mode shapes, on which the responses would depend. He would be interested to learn what methods had been used in that respect.

Calculations of overall ship vibration using large finite element models involved a considerable effort, and use of that procedure at Lloyd's Register was, therefore, generally restricted to particular circumstances. In that case it would seem that having established:

i) that the wake pattern was acceptable;
ii) that the propeller forces were low;
iii) that the hull pressures and integrated surface forces were acceptable;
iv) the principle resonances of the aft cantilevers and methods of detuning those if required;
v) the shafting resonances;

then, the additional information gained from constructing a finite element model of the complete ship was relatively small compared with the effort required, especially as it was generally recognized that accurate response calculations were not possible with the current state of the art. However, there might be many considerations, technical and otherwise, leading to that depth of investigation, which had not been apparent in the paper.

In conclusion, he wished to endorse the authors' final remarks that, provided the study of ship vibratory characteristics was commenced early in the design stage, it was possible to carry out sophisticated analytical studies in a time scale which was not inconsistent with the design and production schedule. The findings from that work could then be used to improve the design at a time when design options were still open. However, as had been noted, it must be remembered that those analytical predictions should be supplemented by good practical experience.

Authors' Replies

The authors were grateful for the quality of the contributions they had received from so many outstanding specialists in ship and machinery vibrations. The kind comments on the results given in the paper were particularly welcome, as the work described had been carried out in difficult conditions due to the relatively short period of study undertaken at an advanced stage of the project and construction.

Those circumstances were a partial answer to several contributors who wished to know whether such extensive investigations had been necessary in view of some apparently favourable characteristics of the propulsive plant and the ship, and especially because of the very positive final results obtained.

Afterwards everything seemed very much more simple and easy than beforehand. For that reason, Valmet Shipyard, despite their own preliminary studies, had asked for the assistance of Bureau Veritas in order to dissipate doubts and to reduce the risks when deciding on such crucial parameters as the number of propeller blades and rev/min. Vibration studies often gave very contradictory results and thereby put the shipyard in a perplexing situation which raised doubts as to the vibratory behaviour of future ships.

The problems which had been put to the shipyard were even more severe, as they concerned:

i) a prototype of special purpose vessels embodying a number of new and untried features and arrangements;
ii) very stringent vibratory requirements stipulated by the Shipowner.

That was why the shipyard had tried to minimize the risks by asking for the reliable assistance and opinions of three specialist hydrodynamic organizations concerning the level of vibratory excitations in addition to the Bureau Veritas investigations into the response side of the vibratory phenomena of the ships.

From the volume of details of those rather sophisticated investigations, it might have been judged that "a sledge hammer had been used to crack a walnut", but the contents of the paper and the contradictory information arising from the different partial calculations would indicate that that was not the case, as there were, indeed, "some walnuts".

The above general considerations in reply to several contributions could be summed up as follows:

1) A sophisticated approach and extensive analytical studies were necessary in order to investigate an unknown design and untried features, and to avoid making a "penny economy" approach which could lead to problems requiring difficult and extensive modifications.

2) A more simple approach could be adopted in analytical investigations of structural and machinery arrangements of conventional or known types, for which the practical experience of previous research would be most helpful.

Dr Archer's remarks concerning the small variation of propeller forces and moments due to the anticipated small variations of the draught had been very valid for that type of ship, but that was not necessarily the case during the operation of the vessel concerned.

To avoid any risk from this point of view, the double-strutted design of bossings were adopted as these had a paramount influence on the lateral stiffness of the assembly and were needed for controlling the lateral and whirling vibrations of the shafting. According to the shipyard that was, perhaps, the most important single reason.

Dr Archer's welcome and timely comments on his experience with RMS *Andes* had drawn attention to the possibility of coupling phenomena between torsional and lateral vibrations of line shafting. They had contradicted a tendency which had been appearing in some technical literature to the effect that the possibility of coupling phenomena only occurred between torsional and longitudinal vibrations. Such a restrictive statement was in opposition to the authors' opinion that there was also a possibility of coupling between torsional and lateral vibrations.

The fittings of a single strut on the ship concerned, and consequent cure of severe vibrations was a worthwhile example of the detuning of a forced vibration resonator formed by the propeller and tailshaft as well as the bossings of the ship

Mr Bunyan's comments concerning the unusual features constituted by the massive cantilever extending from the aft part of the ship, had indicated the possible appearance of another forced vibration resonator, which could originate a secondary source of excitation to disturb the vibratory behaviour of the ship concerned. So, despite careful analytical investigation, it had been thought worthwhile to forestall the possibility of detuning by means of detuning tanks which could be easily filled with water. Fortunately, as the calculations (and their excellent correlation with measurements shown in the Tables in Section 8 of the paper) had proved to be correct, it had not been necessary to use those emergency tanks.

Mr Bunyan had requested details of wake variations at the propeller disc. These were shown in Fig. D1.

The installation of a closed-in tailshaft had been requested by the owner before proceeding with the wake measurements. That decision was in line with the author's experiences with the naked tailshafts of a series of powerful container ships, where serious problems with the tailshaft and its aft bush had occurred due to the deficiency of the watertight devices.

The authors agreed in principle with Mr Eames' comments when comparing the ships concerned with the Maersk Line "A"—vessels, and the shipyard's opinion had been that a simple criterion such as that proposed in the paper by Mr Rutherford might give very useful indications if the criterion showed that trouble could be expected. But the opinion of the shipyard was that, if they had relied solely upon such a simple criterion and not ordered the elaborate vibration analysis carried out by Bureau Veritas, it was almost certain that some severe local vibration problems would have been found during trials, at which time the changes recommended and easily applied at the project stage would have been very expensive to execute. The shipyard had felt that such an analysis was indispensable and only regretted that for the lack of necessary data, an analysis of that magnitude

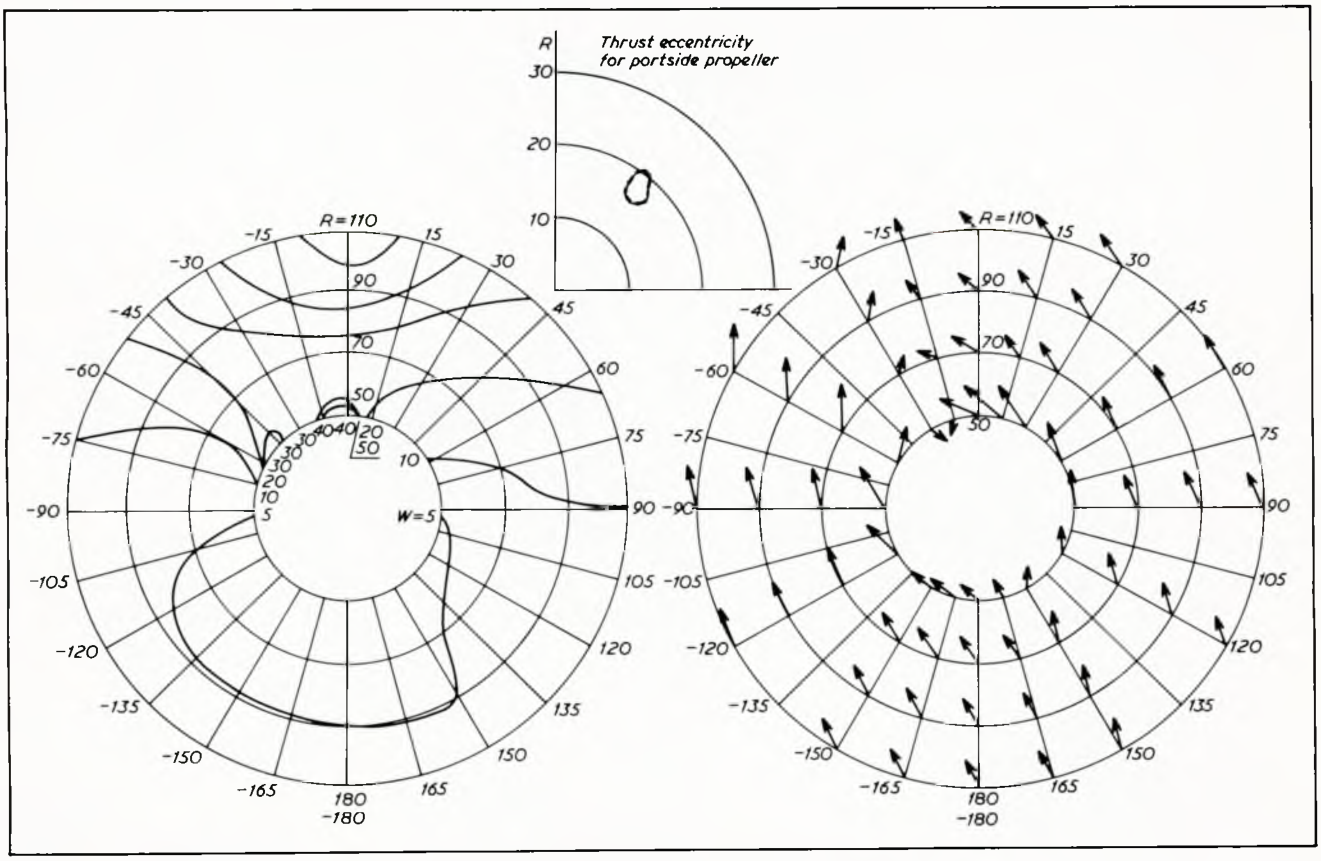


FIG. D1 Wake variations at the propeller disc

could seldom be ordered so early that the full benefit of the final results could be received in time.

As to Mr Eames' remarks and the possibility of forseeing the filling of selected compartments for detuning purposes, that idea was in line with the philosophy conceived (and often applied) by the authors with regard to the research and detuning of any possible forced vibration resonator.

The remark about the fitting of fins would be answered in the reply to Mr Sole, where more information was also given concerning investigations related to hydrodynamic excitations.

Replying to Mr Martyn who had made a request for supplementary Figures: Fig. D2 showed a comparison of the Valmet design with the original SeaBee concept and Fig. D3 showed the machinery lay-out of the ship concerned.

The twin screw arrangement was adopted at the request of the owner. The installation of a six-bladed c.p. propeller was not considered because, as far as the authors were aware, such propellers were not yet available in the size needed for the barge carriers. Consequently, the authors had not wished to suggest such a new and untried c.p.p. installation to KaMeWa.

The authors' opinions concerning the less favourable characteristics of five-bladed propeller forces and moments had been based on the extensive and lengthy experiments conducted by NSMB, the results of which had confirmed the authors' experience gained during past investigations in the field of propeller and line shafting behaviour. They would admit that, sometimes, the five-bladed propeller might give lower forces and moments than the four-bladed one but, in their opinion, that might only occur in extreme cases where a really bad design of four-bladed propeller had been involved and compared with the excellent (from the point of view of hydrodynamic characteristics) five-bladed propeller. But as yet the authors had not encountered such an unusual combination.

It should be emphasized that the angle between the supporting struts was not 90° but 81°. The authors wished to confirm that it had been reported to them that unfavourable vibratory phenomena would occur in the case of similarity between angles encompassed between struts and blades.

The main shaft intermediate bearings were not of roller but white metal type. The mean pressure on the plummer bearing was 4.93 kg/cm^2 and that agreed with the rational line shafting alignment calculations.

The authors wished to confirm, as described in Section 5.3 of the paper, that during calculations of tailshaft and line shafting the effect of variable thrust on lateral forces and moments had been taken into account by using the values of propeller forces and moments as indicated in Section 6.

That had only been possible because of their success in solving the problem of oil film calculations for skewed misalignment in the tailshaft (see References 11 and 12).

To ensure the most homogeneous and least important mean pressure (at very low rev/min) alongside the aft bush, calculations of elastic alignment had been made, taking account of the flexibility of the tailshaft journal and white metal which, after the optimization of the aft bush profile, had led to the slope boring of the white metal. The results of those calculations had been shown in Fig. 4.

The Bureau Veritas approach to the question of tailshaft stiffness emphasized that the line shafting should be conceived in a rational way by creating different

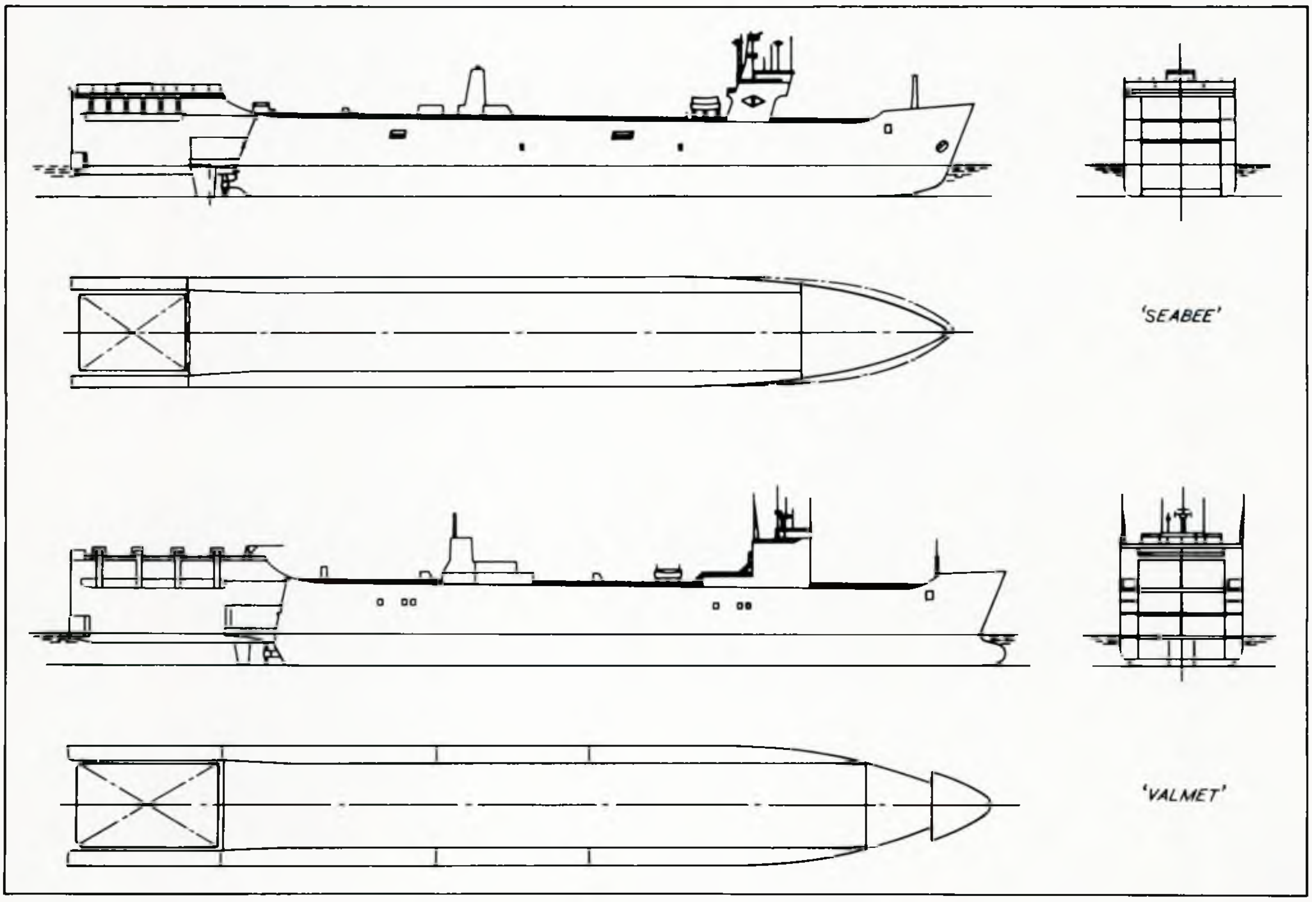


FIG. D2 Comparison of Valmet design with original Seabee

flexibilities for different parts of the shafting. In fact:

a) the tailshaft should be stiff in order to realize high natural frequencies of lateral and whirling vibrations.
b) the intermediate shafts should be as flexible as possible in order to be adaptable to the deformation of the steelwork.

The coexistence of the shafting with the ship hull structure was dependent on the compatibility of their respective stiffness and elasticity.

Realization of the Grim-type stern tube aiming at detuning the resonance below the service speed had not been envisaged for the ship concerned because of the owner's request for double-strutted bossings. However, it was worth noting that dispensing with the struts would have resulted in a more massive stern tube and the total effect on the wakefield would not have been favourable.

The Grim-type stern tubes had been previously studied and adopted by the authors, e.g. for the Danish flag ship *Dana Anglia*. That solution had proved itself very efficient from the point of view of line shaft behaviour and vibratory phenomena.

As to the problem of the increase in stiffness of the thrust bearing foundations, that had been done not only to push the natural frequencies of longitudinal vibrations into a higher region, but also because of the owner's request for an increase in the output of the installation.

The decrease in the natural frequencies of longitudinal vibrations between partial and overall FEM calculations could be logically explained by the fact that, in partial calculations, full embedding conditions of the double bottom in the engine room environmental structures had been presupposed. These had been changed in the overall calculations by the real boundary conditions where they had been realistically calculated. The decrease mentioned by Mr Martyn had seemed to indicate that by partial calculations those boundary conditions had been considered too severe and that the partial model should be extended into outside shelling and bulkhead steelwork.

In answer to Mr Martyn's final question, due to constraints imposed for the number of blades (four) according to the results of free vibrations of lateral vibrations of line shafting and cantilevers, the only practical means of aiming at overcoming the vibratory phenomena which might result in longitudinal vibrations (due to the four-bladed propeller having more important thrust variation than the five-bladed one) could be the increased stiffness of the thrust bearing.

Referring to Mr G. Sole's comments, the problem of symmetric and asymmetric hull surface forces of the twin-screw installation for running the propeller blade in and out-of-phase could be explained as follows: a preliminary vibration study made by Det norske Veritas had indicated that in some load cases the resulting horizontal force components caused by the propellers could be greater when the propellers were running in phase than when running out of phase, if the phase difference was carefully chosen and controlled. For that reason, the phase difference could be changed and automatically

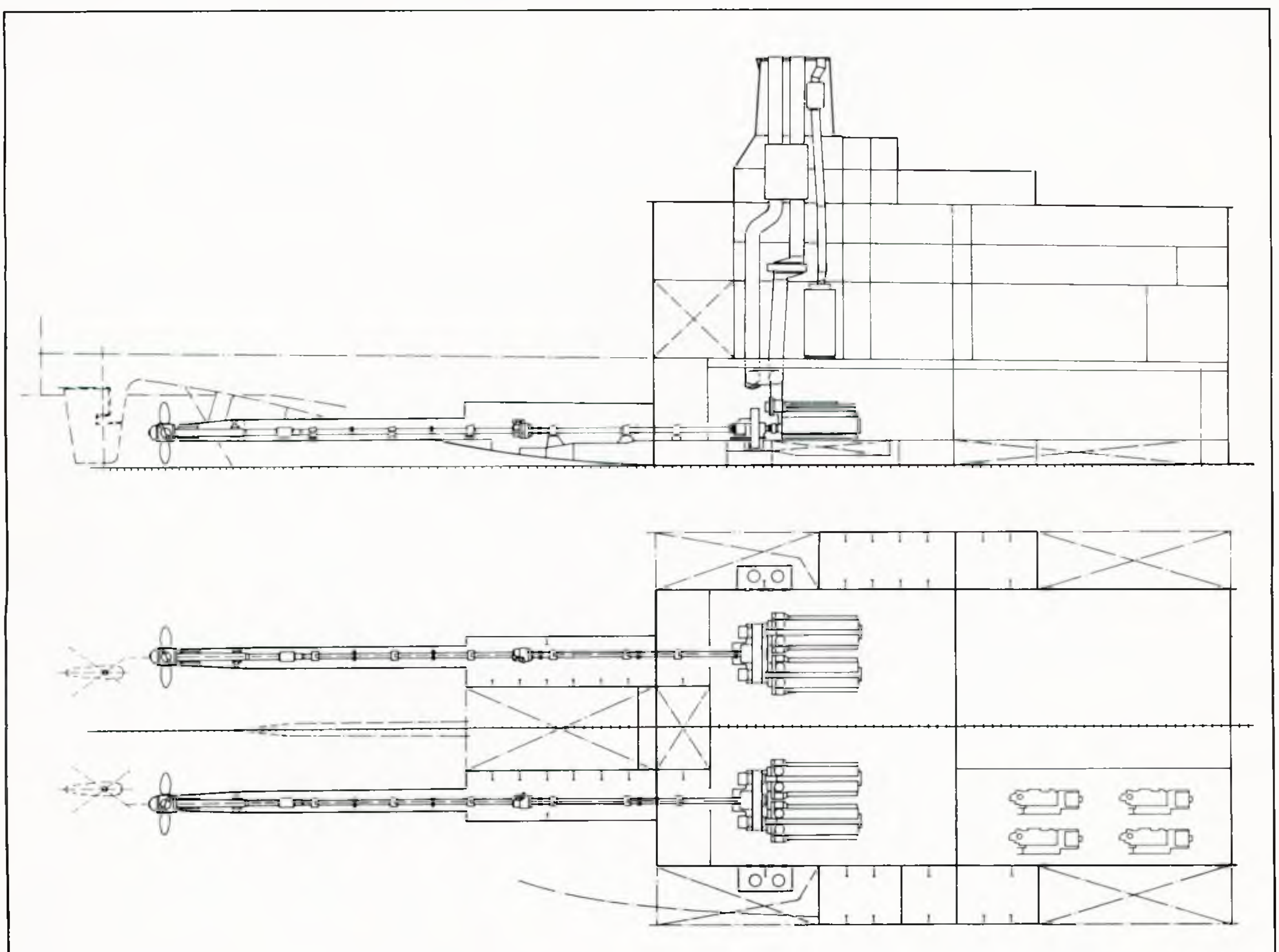


FIG. D3 Machinery arrangement of new design

controlled if necessary. The shipyard did not know, however, if that possibility foreseen on the ship had been used in service, although it had been tested during trials.

Attention must also be drawn to the very flat bottom of the ship, which permitted less consideration of the problems raised by the contributor.

The use of fins was considered because the first pressure measurements made by KaMeWa had given very high maximum pressures. Later measurements made by KaMeWa, and in particular those made by SSPA where a model fitted with fins was also used, showed that the fins were not needed for that ship. On the contrary, the pressures were a little higher with fins than without them because the effect of the decrease of the clearance between hull and propeller blade tip was greater than the effect of the somewhat better wakefield obtained with the fins.

The authors had noted the interest in the Bureau Veritas introduction into shipbuilding of the modal synthesis technique which had actually been commonly used by them for about three years. Further details followed.

Modal synthesis technique

The method used in their calculations had been the modal synthesis technique with "embedded substructures". Great care and experience were needed to use that method and the results depended on:

1) The choice of the different substructures: it was necessary to have a good idea (before calculation) of the main mode-shapes, in order to make the most suitable possible choice from the different substructures.

2) The number of calculated modes of each substructure: that parameter was very important for the accuracy of the modes obtained after the assembly of all the different substructures. It was necessary to take into account the number of degrees of freedom in the substructure concerned and, also, the different mode shapes which could be expected.

A suitable determination of the two above-mentioned parameters would ensure that a correct physical representation of the vibratory phenomena would be obtained by the final calculations.

Calculations of overall ship vibrations

One of the main interests of the calculations performed on the whole ship had been to investigate the coupling effects between different parts of the ship and to prove the presence of eventual forced vibration resonators. Due to their resonant response, their presence (the origin of secondary source excitations) provoked the elevated vibratory level of a large part of the environmental steelwork.

The call for FEM was needed in the calculations of overall ship vibrations because of particular circumstances, i.e. the uncommon open geometry steelwork of superstructures, which might vibrate in longitudinal, vertical and transverse directions. The authors agreed that such overall calculations were seldom needed for conventional ships, but might be imposed for an unconventional design.

The authors were glad that Mr Sole had endorsed their opinion that, today, the technique for investigating ship vibration characteristics, if commenced early at the design stage, would allow sophisticated analytical studies to be carried out within the design and production schedule. But it must be remembered that analytical production should be guided and supplemented by good practical research experience in the relevant studies.

Published for THE INSTITUTE OF MARINE ENGINEERS by Marine Management (Holdings) Ltd., (England Reg. No. 1100685)
both of 76 Mark Lane, London EC3R 7JN. Printed by Gibbons Barford, Wolverhampton.