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SHORE TRIALS OF A MARINE PROPULSION SYSTEM — Computer Controlled Manoeuvring Trials

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The paper discusses a means of performing simulated ship manoeuvres on a shore test facility. The system devised takes advantage of the propulsion machinery arrangement to eliminate the need for special purpose machinery and employs direct digital control to maintain shipboard loading conditions. The paper gives a brief outline of the plant and describes the progress of the control system from its conceptual requirements through to its final design, impletation and use on the shore test facility. The manoeuvring results are discussed with particular emphasis on the dynamic performance of the hydraulic couplings, and the phenomenon of main shaft stall. The trials carried out have demonstrated the ability of the machinery to withstand manoeuvring transients and have confirmed the effectiveness of the method used which represents a significant advance in shore testing technique. It is regretted that, for security reasons, no detailed information on the propulsion system trials can be included in the paper.

INTRODUCTION

This paper is concerned with a series of computer controlled manoeuvring trials conducted on a marine propulsion system on a shore test facility(STF). These trials featured the use of a novel dynamic loading system, controlled to simulate the loading conditions which would be experienced at sea during manoeuvring, particularly crash stop manoeuvres from high ahead ship speeds.

Background to the STF

A comprehensive justification of the shore trials concept is not within the scope of this paper; however, it is interesting to note that the cruiser propulsion plant represents the highest power per shaft used in any R.N. ship. A risk assessment of the problems likely to be experienced in the ship (Ref. 1) showed that the potential areas of concern which could be evaluated at a shore test facility were:

Gearbox;
Uptakes and downtakes;
Machinery controls.

Although none of the techniques and components used in the propulsion system are new, the combined use of clutches and fluid couplings at the high powers required was outside current R.N. experience. The potential problems associated with the design of uptakes and downtakes described in Ref. 2 are compounded in the cruiser because of the long ducting necessary due to the island superstructure. In order to reduce machinery manning requirements, the degree of automation used in the machinery control system is high. Consequently, extensive testing and evaluation was required prior to sea trials.

The principal advantages of the shore test facility were considered to be:

a) during sea trials, there would be many trials other than those associated with machinery and hence time for machinery trials will be at a premium;
b) if extended machinery trials on the ship were necessary, substantial costs would be inevitable;
c) there would be less scope on the ship for stopping and starting machinery and for making adjustments to control settings etc.

Previous experience on sea trials of first of class warships indicate that the more difficult problems arise during transient manoeuvres rather than during steady state running. Before the cruiser sea trials, the manoeuvring performance was examined in more detail than previously using computer simulation. These studies highlighted some machinery dynamic problems, particularly during crash astern manoeuvres, which were considered sufficiently serious to justify full scale trials on the STF. The simulation studies indicated that the most severe machinery conditions occured during a single engine per shaft crash astern manoeuvre and the problem was caused by the high power induced by the forward momentum of the ship acting on the transmission system via the fixed pitch propeller. The machinery torque/speed conditions produced by this injected power could not be provided on a shore test facility by a totally passive loading system (ie dynamometer), and therefore there was a need to inject power into the propulsion system to simulate the momentum induced torque. The dynamic loading system adopted to impose the transient loads has become known as the Power Injection System.

The main advantages of performing simulated "ship" manoeuvres on the STF were:

d) the machinery will have been demonstrated to have undergone a series of manoeuvres consistent with expected shipboard behaviour;
e) any propulsion system limitations should at least be known and eliminated or countered by operating procedures before sea trials;
f) extensive data will be obtained on gas turbines, hydraulic couplings and general transmission system behaviour which will allow design reappraisal and further studies to proceed with more confidence in machinery characteristics.

PROPULSION SYSTEM

Machinery Arrangement

Each shaft set of the propulsion system has two gas turbines which can drive together or singly into a reversing gearbox, the output of which is connected via the main shafting to a fixed pitch propeller. Manoeuvring drive ahead and astern is transmitted via scoop trimming double circuit hydraulic

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couplings and direct drive ahead via self synchronizing clutches is used for economical cruise and high powers. The shore test installation is a complete shaft set of main machinery, including full scale uptakes and downtakes, gas turbines, gearbox, ancillaries, mainshaft thrust block and a short main shaft directly coupled to a water dynamometer. The machinery layout is shown diagrammatically in Fig. 1. The propulsion controls are fully represented and local, ship control centre and bridge levers are fitted.

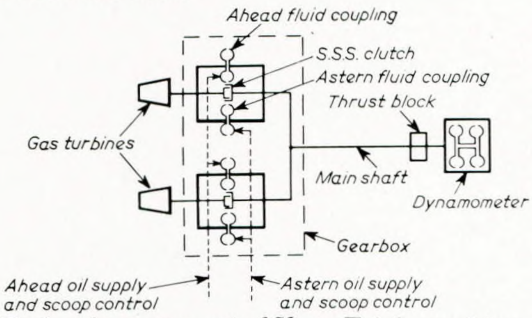

Fig 1—*Arrangement of Shore Trials machinery*

Propulsion Control System

In normal operation, overall control of the engine power and the machinery is obtained using a single lever (Power Demand Lever — PDL) for each shaft set from the bridge or the ship control centre.

Individual manual control of the propulsion plant is available at the local control centres in the machinery spaces. In normal operation, these will be unmanned.

A detailed explanation of the functioning of the propulsion control system is outside the scope of this paper. However, it is pertinent to later discussions to outline the control system functions and sequencing, and the machinery behaviour during a typical crash stop manoeuvre:

a) PDL moved to required astern position;
b) gas turbine reduces to idling;
c) (if in direct drive) clutch actuator moves to the 'ready to disengage' position;
d) (if in coupling drive) ahead coupling begins to empty;
e) astern coupling begins to fill;
f) (if in direct drive) reverse torque disengages the clutch (with transient brake assist if necessary);
g) power reapplied to gas turbine;
h) main shaft decelerates, reverses, and ship subsequently stops.

Potential Problem Areas

The above procedure gives the control system requirements in principle; in detail the control sequencing was investigated using computer simulation methods, and the following problem areas were highlighted:

Clutch Disengagement

The gear ratios of the gearbox, particularly between the clutch and hydraulic coupling lines, were designed to facilitate the engagement and disengagement of the clutch. To disengage the clutch during decelerations or crash stops, the reverse torque required to create a speed differential can, in most cases, be provided naturally by the reduction in power of the gas turbine combined with the ship momentum effects. Additional torque of the required sense is available from the filling coupling (ahead or astern) and, as a last resort, from a transient brake fitted on the intermediate speed gear line upstream of the clutch.

Power Turbine Speed

As the ahead power is taken off, and the transmission changes from ahead to astern drive, the point of reapplication of power must be chosen with care to achieve a compromise between high power dissipation in the filling astern coupling and low power turbine speed. Power reapplied too soon leads to a high rate of increase of oil temperature in the oil coupling; power reapplied too late leads to low power turbine speed and possible reversal which were expected to have undesirable effects on the power turbine bearings.

Coupling Oil Temperature

As the main shaft decelerates at a rate dictated by the applied astern power and the opposing (ship way generated) 'propeller' power, the power dissipation in the astern coupling builds up to a maximum (typically) just prior to shaft reversal, producing a high rate of increase in oil temperature. A balance between stopping performance and constraints on the allowable oil temperature (due to possible oil degradation) has to be obtained.

Main Shaft Stall

Gearing efficiency and friction studies indicate that (in common with other propulsion systems of a similar type) the main shaft will stall, ie remain at zero rev/min, for considerable periods during manoeuvres from high ahead speeds. The stalled propeller shaft will produce a lower astern thrust (thus resulting in a reduction in stopping performance) and could lead to mechanical damage to the transmission system. In addition, all the engine power is dissipated in the astern hydraulic coupling for the duration of the stall, thereby maintaining at a high level, or increasing, the coupling oil temperature.

The common factors in the above problems are the high power fed into the system via the propeller and the characteristic behaviour of the hydraulic couplings, particularly at high slip conditions.

The simulation studies dhowed that the worst case for these problems arose during single engine/shaft crash stop manoeuvres. This factor is of significance in the design of the power injection system, as will be seen later.

Manoeuvring Studies

Having established that problems existed, more detailed studies using computer modelling techniques were pursued in attempts to assess whether the problems could be designed out of the system. Manoeuvring procedures and control system parameters were determined to maintain the machinery within the known design constraints. Sensitivity studies were performed to assess the effects of uncertainty in the defined characteristics of the machinery system.

However, it was concluded that due to doubt about the reliability of some performance data used in these computer simulations it was necessary to obtain additional data before continuing simulation work.

An extensive work programme was initiated to procure performance data of relevance to the propulsion system. The programme included work in the following areas:

Hydraulic Couplings	—	manufacturers tests in ½ size coupling
	—	manoeuvring trials on County Class Destroyer which has similar type of propulsion system (½ size couplings)
Shaft Stall	—	manoeuvring trials on County Class Destroyer
Propeller and hull	—	tank tests on scaled model in steady and dynamic conditions.

Whilst these trials provided valuable data, the data obtained suffered from the disadvantage of requiring extrapolation to full size. Nothing more could be done on propeller characteristics until Sea Trials, but the STF was available for experimental purposes and it was decided to maximize the use of this facility to provide the full scale data required on coupling performance and shaft stall conditions and thereafter to examine the manoeuvring problems outlined above.

POWER INJECTION SYSTEM

Basic Concept of Power Injection

During a crash stop manoeuvre from a high ahead speed, ahead propeller torque derived from the ship's way is imposed on the machinery system. With the fixed pitch propeller, this torque combines with the inertia torque of the machinery tending to maintain the ahead rotation of the mainshaft. After the shaft reverses, and whilst the ship is still moving ahead, the effect of continuing propeller feedback is felt as an additional resistance to acceleration of the main shaft in the astern direction. Thus, for astern rotation the system will be operating at higher toque levels than those given by the astern propeller law. This high torque loading will continue until the ship reaches steady astern condition.

Simulation studies of the ship manoeuvring showed that the most arduous condition for the propulsion machinery occurred during single engine/shaft crash stop manoeuvres.

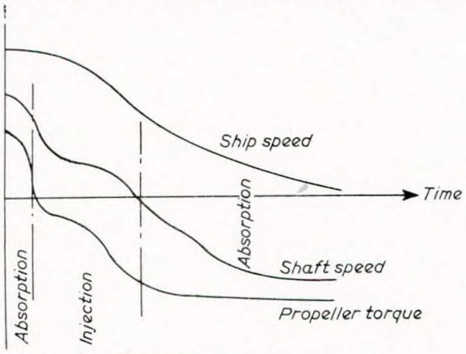

Fig 2—*Typical stopping transient*

Fig. 2 shows the speed, propeller torque and shaft speed transients expected during a typical crash stop manoeuvre. In terms of propeller loading, the manoeuvre can be divided into two periods of power absorption separated by a period of power injection. This can be more clearly observed from Fig. 3 which shows the propeller torque/shaft speed trajectory throughout the manoeuvre. Superimposed on Fig. 3 is a typical torque/speed trajectory which would be available using the STF machinery arrangement in normal test conditions. Of significant importance is the requirement to impose high torque loads on the system at very low and zero main shaft speeds.

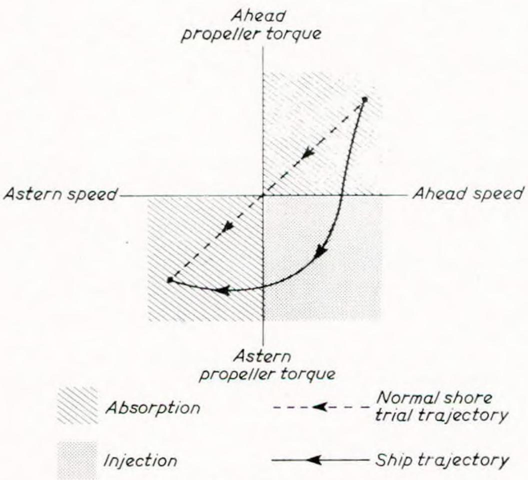

Fig 3—*Typical torque/speed trajectories*

Machinery Requirements

To reproduce these loading conditions on the STF for a wide range of crash stop manoeuvres requires the provision of controllable load absorption and power injection devices.

The double circuit water impeller dynamometer is bi-directional and in principle can provide the load absorption for ahead and astern operation. The natural torque/speed characteristics (obtained with preset control valves) of the dynamometer are not compatible, however, with the requirement that dynamic load control of both ahead and astern compartments is essential.

The power injection source adopted for the system was one of the installed gas turbines driving through its ahead fluid coupling, the other gas turbine providing the manoeuvring power. This arrangement was compatible with the intended usage of power injection — which is the simulation of single engine/shaft manoeuvring conditions, and had the advantage over other proposed devices and arrangements of requiring neither additional capital expenditure or major machinery modifications.

Power Injection Control System

The power injection control system is required to compute the load torque to simulate ship conditions and to control the load absorption and power output of the dynamometer and injection engine respectively so that this load torque is imposed on the manoeuvring engine and transmission during the manoeuvre.

The basic requirements of the control system were defined in broad terms by the knowledge of the required manoeuvring loads and of the capabilities and operating regimes of the dynamometer and injection engine. A mathematical model of the system was used to examine the feasibility and controllability of the proposed system. The model was formulated such that it could represent both the ship and STF configurations, providing the feature of almost immediate comparisons during the development of the control system. The computer based control system for power injection, indicating the main control function and the measurement and control signals involved, is shown in Fig. 4. Further details of the power injection control system can be found in Ref. 3.

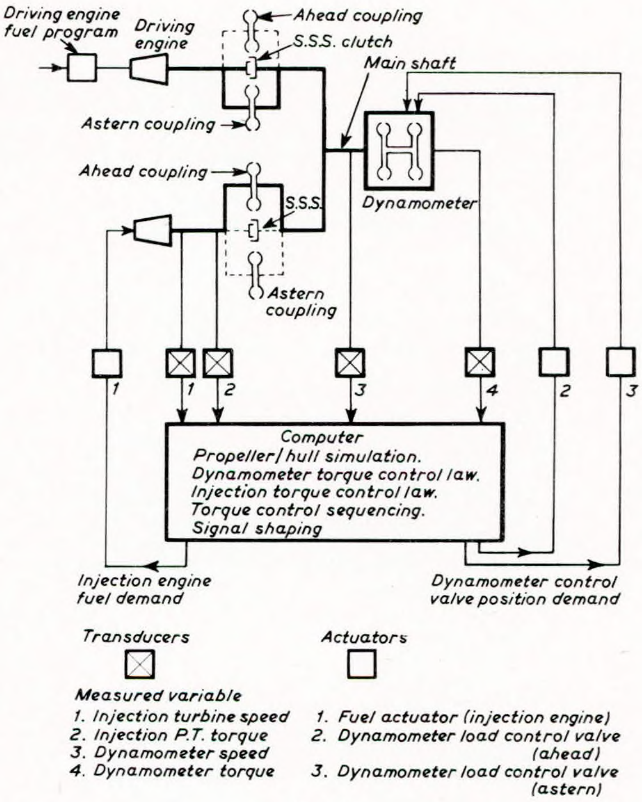

Fig 4

The control system comprises four basic sub-systems as shown in Fig. 5 and which are described below.

Torque Reference System

Since the power injection system is required to reproduce loading conditions on the manoeuvring engine and transmission consistent with those imposed by the propeller under equivalent ship manoeuvring conditions, the load torque imposed by the power injection control system must be made equal to the propeller torque experienced by the ship under these conditions.

The torque reference signal for the control system is obtained from the solution of the single degree of freedom ship motion equation, the propeller torque and thrust equations, and the hull/propeller interaction factors. The solution of the equations inherently formed part of the power injection control system and pointed the way towards a computer-based control system.

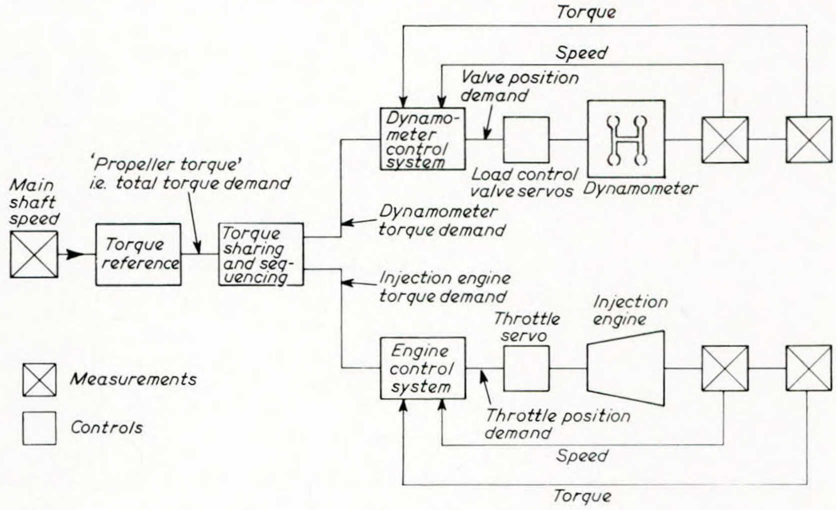

Fig 5

Dynamometer Control System

This system computes the dynamometer load control valve (LCV) position required to produce a given load torque at a given shaft speed. The demand LCV position signal is passed to the existing LCV position servo systems. Proportional and integral torque error signals are used to close the control loop.

Injection Engine Control System

This system computes the required injection engine throttle position from an input torque demand and the signal is passed to the existing on-engine fuel control system. A proportional torque error terms closes the control loop, and phase lead compensation is used in the feed forward path.

Torque Sharing and Sequencing System

This control system provides the necessary interface between the torque reference system and the individual torque control loops described above, to ensure that the overall torque requirements are satisfied and to allocate control responsibility and torque sharing between the dynamometer and the injection engine during the course of a manoeuvre. The main problem encountered in the development of the control system was concerned with the arrangements for timing the "entry" and "exit" points of the injection engine. From Figs. 2 and 3, the power injection requirement occurs in the middle phase of the manoeuvre, indicating that the injection engine is required only during this phase. However, because of the time factors involved in getting the injection engine and coupling into the system and capable of supplying the required injection power at the time required, there was no alternative to arranging that the injection engine and coupling be connected and supplying power to the system prior to the start of the crash stop manoeuvre. This required the dynamometer to absorb much higher power in order to dissipate that produced by the injection engine, always resulting in the net torque experienced by the driving engine being equal to the ahead propeller torque.

In addition, since the dynamometer is not capable of absorbing the required load at low astern speeds, the period of power injection had to be extended to compensate. At higher astern speeds, the dynamometer can absorb the total required astern load but it was considered that to allow it to do so would pose the difficulty of deselecting (ie taking out of circuit) the injection engine and coupling whilst maintaining overall torque control on the system. The most practical solution was to leave the injection engine in the system until the manoeuvre was complete, and to adopt a load sharing control policy for this phase of the manoeuvre.

In view of the above, an overall control and sequencing system was determined with the injection engine and coupling in circuit and supplying power to the machinery system throughout the manoeuvre. The system adopted, split the control requirements into three phases (denoted Phase I, II and III). The phase boundaries and typical machinery transients and control modes are shown in Fig. 6.

Application of System to Other Manoeuvres

Although primarily designed to simulate single engine/shaft ship crash stop manoeuvring conditions, the control system can be easily adapted to cater for acceleration and deceleration manoeuvres, which generally require only the control of the dynamometer torque. For these cases, single and twin engine operation can be covered, limited only to the maximum loading capability of the dynamometer.

COMPUTER CONTROL SYSTEM

The real time solution of the ship motion and propeller/hull equations required for power injection, and the general complexity of the control system, led to the adoption of a computer based control system. Consideration was given to both analogue and digital computers since both types could satisfy the primary requirement of torque control. Other secondary considerations, however, led to the selection of a general-purpose digital computer; these considerations included:

a) data logging availability;
b) reduced initial set up and commissioning period;
c) reduced set up and checkout prior to each series of manoeuvres;
d) reliability and program security;
e) application to other tasks on the STF and other projects.

Computer Software

Although the primary function of the control computer was to control the machinery during power injection manoeuvres, it was realized that other machinery control modes and system monitoring requirements would almost certainly arise during the course of the project. To allow considerable flexibility of control policies and to facilitate the incorporation of such additional requirements, the computer control system was implemented using a general purpose control and simulation package with subroutines tailored for each particular function. (In retrospect, this decision has been validated on innumerable occasions, and it allowed the Shore Trials program initially conceived to be considerably extended and modified with relative ease).

The computer/operator interface operates via the teletype and an extensive range of control commands, information requests and system monitoring functions are provided. All control actions and sequencing performed or required by the computer are checked and violations are flagged to the operator. The computer continuously monitors the state of the plant using a set of prescribed limit and rate values for particular machinery and control variables. Again, violations of these allowed values are flagged to the operator via the teletype and in particular cases to the plant operator via indicator lights as hazard warnings. The computer operator can request information on any or all system variables at the teletype. The computer software includes a data logging facility by which 60 channels of information can be recorded on a magnetic tape. This facility is intended primarily for detailed trials performance analysis, but can also be used in a more restricted manner for on-site data replay on a U.V. recorder as an aid in trials diagnostics.

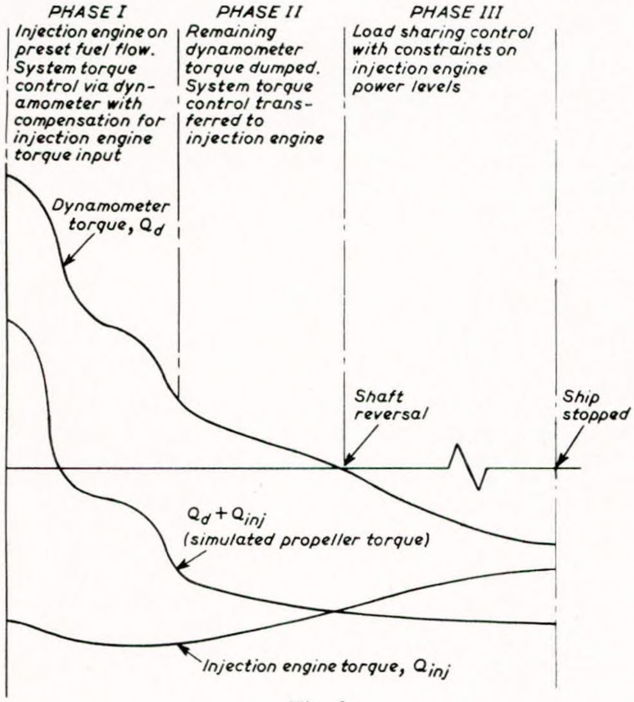

Fig 6

Instrumentation Systems

The installation of the computer control and monitoring system was achieved using, with one exception, existing actuators and transducers. The input requirements of the control actuators (dynomometer load control valves and injection engine throttle) were compatible with the outputs of the computer digital to analogue conversion system and direct connexions could be made via computer control/normal control switching arrangements. The only items of special purpose instrumentation required were power turbine torque-meters.

In addition to a wide range of machinery performance parameters, all significant control system variables (e.g. "ship speed", "propeller torque", "demand torques", etc) used within the computer were data logged for subsequent analysis.

Safety Considerations

In normal test house running, plant operators have total control over the plant, aided by the normal system protection/warning devices. For power injection manoeuvring using the system described, the operator has control only over the manoeuvring engine, and the computer exercises control over the dynamometer and the injection engine.

It was realized that, during power injection, malfunctioning of machinery control functions (including the computer control channels) could produce unusual failure effects (because of the non-normal operating conditions) which could render immediate diagnosis and operator intervention extremely difficult.

For these reasons, a safety assessment study was conducted using simulation techniques.

The study showed that there was little benefit in attempting to recover the manoeuvre in failure conditions and a single action "terminate manoeuvre" function was provided. This function effectively reduces the fuel flows on each engine to below idling in 100ms and deselects the engines from the transmission system.

The study also recommended the use of protection and warning devices in addition to those already in existence for normal running.

CONTROL SYSTEM TESTING

To fully evaluate and test the computer control system prior to power injection manoeuvring on the Shore Trials, the system was subjected to two series of comprehensive functional and operational checks.

The first series of tests were performed using the control computer in conjection with a hybrid computer model of the machinery system and control consoles, to exercise the control system in conditions as near realistic as possible, and thereby to establish the control system parameters for the various trials requirements.

The second series of tests were performed on the Shore Trials machinery. They examined the performance of the individual torque control loops and the torque reference system in normal machinery conditions (i.e acceleration, deceleration, steady state). Since the effectiveness of the control system depends somewhat on the operating characteristics of the machinery system, the control system tests were preceded by a fairly extensive series of machinery trials designed to establish those aspects of machinery performance relevant to power injection; the information obtained was used to update the control software as appropriate.

The particular functions of each series of tests are summarised below.

Simulated Machinery Tests

1) Specification of control system parameters;
2) Sensitivity of control system performance to fixed parameters;
3) Specification of trials requirements and operational procedures;
4) Trials data base for comparison with real trials.

Real Machinery Tests

A) Performance characteristics of dynamometer, engines and fluid couplings;
B) Evaluation of control system in real conditions:
   i) dynamometer control,
   ii) Engine control;
C) Evaluation of instrumentation systems.

Machinery Control During Trials

No particular problems were experienced with the engine control loop, the system response being generally in accordance with computer model predictions. However, considerable difficulty was found in establishing stable conditions on the dynamometer during the transient tests, particularly during slam accelerations from low shaft speeds. Since these machinery conditions are similar to the conditions that would exist during phase III of the power injection manoeuvres, the observed instability was of great concern. An acceptable response was obtained by increasing the inlet water flow rate, reducing the integral error gain and incorporating a phase lead compensator. The response of the dynamometer in decelerating conditions was satisfactory and consistent with model predictions, indicating that satisfactory torque control in phase I of the power injection manoeuvres would be achieved. The preliminary series of the power injection manoeuvres reflected the above findings. Control of the dynamometer during phase I (basically a load shedding operation) and engine control during phase II were satisfactory. No difficulties were obtained in the phase I to phase II changeover where control was transferred from the dynamometer to the injection engine and smooth transfers were obtained for all manoeuvres. Phase III (ie astern mainshaft) operation posed some initial difficulties in maintaining system stability. This astern instability was not always present, and poor repeatability was evident, suggesting that the load absorption characteristics of the astern dynamometer compartment were inconsistent. It is believed that the problem occurs because the astern dynamometer compartment becomes "choked" due to the relatively long ahead running period with high inlet water flows. This produces maximum increase of load astern initially (which is a desirable feature) but, as the astern speed increases the dynamometer "unchokes" and the water content reduces rapidly. The resultant reduction in dynamometer torque creates a transient unbalance in the astern load sharing which the engine cannot cope with because of its rate limited (throttle) response. The resultant effect in the system is to set up oscillations and continual load transfer between the dynamometer and the injection engine.

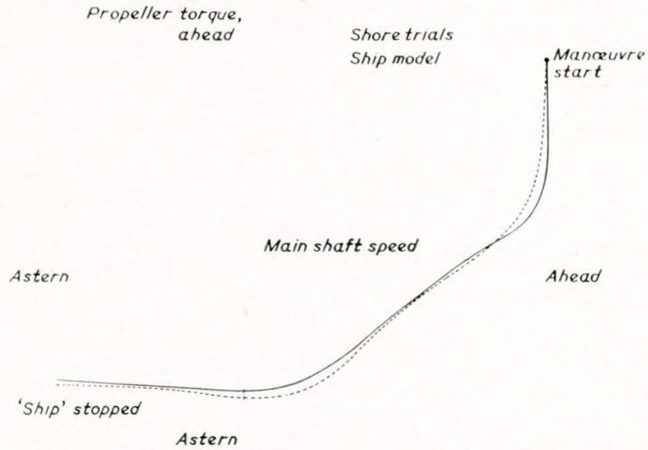

Fig 7—*Shore Trials/Model mainshaft trajectories*

Experiments with various water inlet flows proved unsuccessful and the problem was overcome by eliminating the astern load sharing controls and allowing the dynamometer to follow a natural astern load line by presetting the astern control valve and forcing the injection engine control system to control the overall torque requirements. The modification was effective for all manoeuvring conditions but required additional care to ensure that the proportion of astern load absorbed by each device was maintained at levels consistent with machinery safety.

This problem arises because the injection coupling will be operating at high slips since each side of the coupling will be rotating in different directions. The power dissipated in the coupling is the power output of the injection engine plus that proportion of the manoeuvring engine power not absorbed by the dynamometer. Thus to avoid high power dissipation (and hence high oil temperatures) on the injection coupling, the proportion of the astern load absorbed by the injection coupling must be maintained as low as possible. For each manoeuvre, however, there exists a lower limit to the usable injection power, because at low injection engine power levels, the ahead coupling slip will tend to approach 200 per cent (each side of the coupling running at the same speed but in opposite directions). Any reduction in injection engine power (to meet a lower torque demand) can only result in speed reduction on the injection engine, thereby increasing the coupling slip above 200 per cent. The coupling will now be operating in an unstable mode, and torque control becomes extremely difficult. The minimum torque that the injection engine and coupling can transmit to the system whilst maintaining the coupling slip below 200 per cent can be calculated for each manoeuvre. Using the dynamometer torque/speed characteristic, the required L.C.V. position could be found: a setting of 60 to 70 per cent closed gave satisfactory astern load distribution.

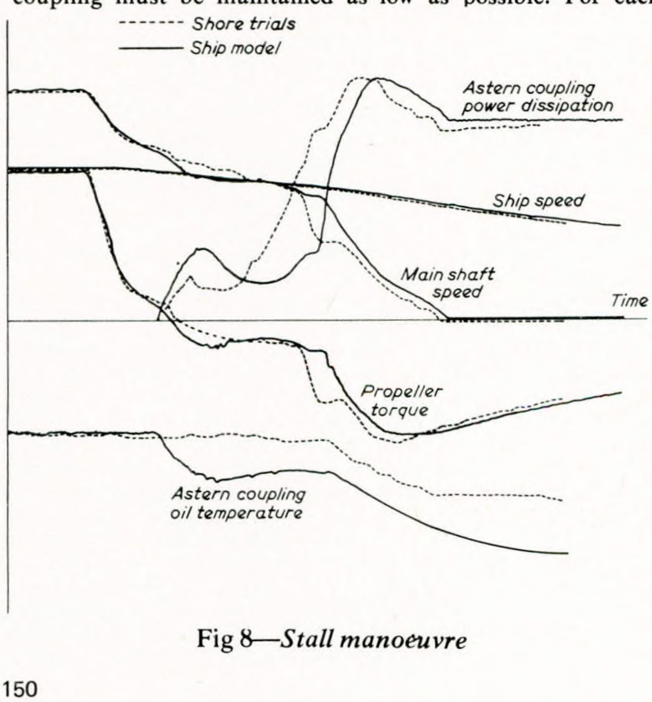

Fig 8—*Stall manoeuvre*

The preliminary manoeuvres were consistent with the predicted behaviour and provided sufficient operational information of the system to proceed with high power manoeuvring and shaft stall trials.

MACHINERY PERFORMANCE

Manoeuvring Trials

A total of 78 power injection manoeuvres were performed covering a wide range of single engine/shaft crash stop manoeuvres.

The first series of 20 manoeuvres was concerned with general aspects of machinery manoeuvring performance (eg clutch disengagement, power turbine behaviour, ship control sequencing) and to provide data on the transient performance of the hydraulic couplings particularly in high power, high slip conditions.

The second series of 58 manoeuvres was designed to further the investigation of the effects of main shaft stall on the propulsion system and to examine the effectiveness of proposed stall breaking and stall avoidance procedures.

The automatic transmission mode changeover (clutch ⇌ coupling drive) were examined in controlled slam acceleration and deceleration conditions.

Dynamic Loading

The trials conducted have demonstrated that the power injection system has adequately represented the expected shipboard loading conditions on the propulsion system.

Comparative torque/speed trajectories are shown in Fig. 7. The manoeuvre shown is a non-stall manoeuvre from maximum ahead/maximum astern power levels. (Further trials/model correlations are shown for the shaft stall trials in Figs. 8 to 11 inclusive).

General Machinery Performance

Clutch Disengagement

For all manoeuvres from the direct drive condition, the clutch disengaged satisfactory without assitance from the transient brake fitted on the upstream side of the clutch.

Power Turbine Speed

The power turbine speed droop during the ahead to astern transmission changeover was not as pronounced as predicted, and is acceptable.

Coupling Oil Temperature

The peak values of astern coupling oil outlet temperature was significantly lower than that predicted from simulation studies, the difference being in the range -7 deg to -1 deg. C (20 to 30 deg. F). Some uncertainty in the accuracy of the measured temperatures exists because of the location of the temperature transducers in a down pipe (forced by restricted access), where total oil immersion cannot be guaranteed. However, based on the measurements made, the problem of high coupling oil temperature is not sufficiently pronounced to cause any oil degradation problems.

Ship Control System

The ship and machinery control system performed satisfactorily during the manoeuvres and all torques, speeds, etc were maintained within the design criteria.

Shaft Stall Trials

The various types of manoeuvres performed in connexion with main shaft stall fall into the following categories:

A) Stall producing manoeuvres — to determine thrust block loads required to produce consistent stall conditions;

B) Stall breaking manoeuvres — to investigate three specific types of stall breaking procedures;
 (i) self break astern;
 (ii) stall break astern;
 (iii) stall break ahead;

C) Stall avoidance manoeuvres — to investigate a specific type of stall avoidance procedure in conditions which would otherwise produce a stall.

Thrust Block Loading

In order to relate the results of these procedures to the ship case, it was necessary to ensure that the static friction (stiction) loads in the two systems are compatable. There are several reasons why this is not the case: no main shaft plummer bearings on the shore test system, negligeable thrust at the thrust block, and the machinery bearing load reactions different due to the action of the power injection machinery. In order to overcome this problem, provision was made to increase the static frictions loads on the shore test system by applying pressure to the ahead and astern thrust block pads, thereby squeezing the thrust collar. Estimates of the required thrust block pressure were made using results of earlier main shaft breakaway trials — pressures in the range 10 to 20 bar (145 to 290 lbf/in^2) (depending on manoeuvring conditions) were calculated.

In practice, it was found that these pressure levels were insufficient to produce stall and consistent stalls could only be produced using pressures in the range 75 to 100 bar (1088 to 1450 lbf/in^2). It is thought that this situation arises because the dynamic build up of static friction in the STF is less rapid on the ship system. The factors relevant to this argument are:

1) On the STF, the stiction loads in total emanate from oil lubricated bearings. On the ship system, contributions to the stiction loads come from both oil and water lubricated bearings.
2) Typical viscosities of oil and water are 40 and 1cSt respectively and therefore the flow from the water lubricated bearings will be greater than that from the oil lubricated bearings, resulting in a more rapid rise in friction coefficient in the water lubricated bearings.

This difference in coefficient rise time is significant as the stiction torque associated with the water lubricated bearings is approximately 25 per cent of the total ship stiction, and in order to produce the fast initial rate of increase of stiction thrust block pressures in excess of those calculated from purely static considerations are required. In the stalled condition however, the high thrust block pressures will impose very much higher stiction levels than estimated for the ship, consequently reducing the correlation to ship conditions in terms of stall duration. The thrust block pressure was dumped to the lower values after stall had occurred in an attempt to maintain "static equivalence" to the ship case.

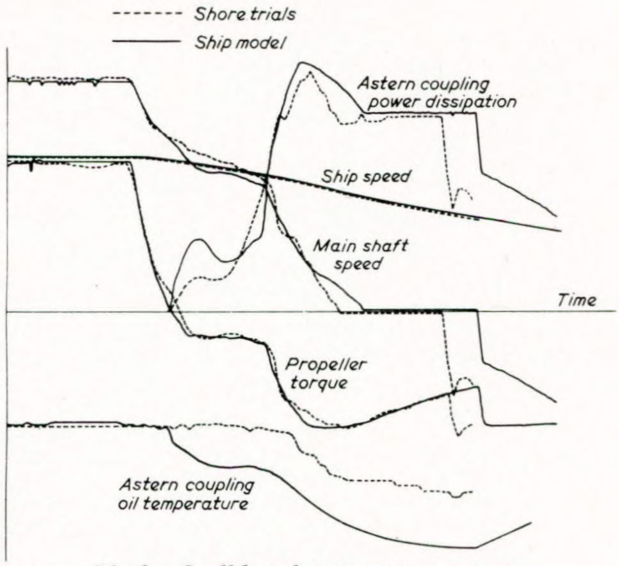

Fig 9—*Stall break astern manoeuvre*

Stall Producing Manoeuvres

These manoeuvres were performed to determine the thrust block pressures required in practice to produce consistent shaft stall condition for the various manoeuvring conditions required. An example of such a manoeuvre is shown in Fig. 8, which was performed with a thrust block pressure of 75 bar. The equivalent ship model is superimposed for comparison. The trials/model comparisons shown in Fig. 8 (and Figs 9, 10 and 11) were time synchronised at the start of the manoeuvre. The subsequent loss synchronization is due to the differences between trials and model control system sequencing parameters (ie, coupling filling rate, engine fuel application time etc). This difference could be resolved by running the computer model with the actual trial control system timings. A further point to note particularly in Figs. 9 and 10 is that the stall break procedures shown are manually initiated, resulting in additional timing differences. The significant feature is not

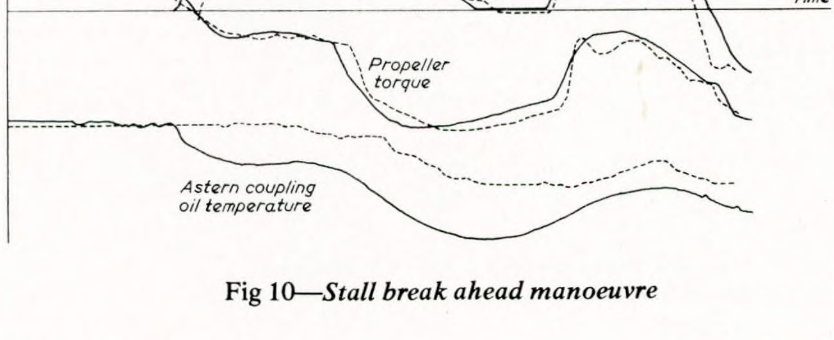

Fig 10—*Stall break ahead manoeuvre*

however, exact time synchronization but the general equivalence of corresponding model and trial variables in terms of magnitude, form, and relationship to each other. In these terms the results shown exhibit very close correlation with the exception of coupling oil drain temperature which has been noted earlier.

Stall Breaking Manoeuvres

Three variations of stall breaking procedure were examined:

i) Self break astern, where the main shaft is left in stall to breakaway astern of its own accord due to the reducing 'propeller' torque;
ii) Stall break astern, where the power level on the manoeuvring engine is increased after stall to produce a net increase in "astern driving" torque on the system;
iii) Stall break ahead where the power level on the manoeuvring engine is reduced (to idling) after stall to allow the 'propeller' torque to drive the main shaft ahead with minimal opposition from the manoeuvring engine; and thereafter to re-apply manoeuvring power to achieve astern rotation.

Examples of the stall break astern and ahead manoeuvres are shown in Figs. 9 and 10 respectively. The manoeuvring conditions and thrust block pressure are the same in all cases and the equivalent ship model responses are shown for comparison.

Stall Avoidance Manoeuvres

Computer simulation studies showed that main shaft stall can be prevented by an additional net decelerating torque applied to the system just before the main shaft reaches zero speed. The procedure adopted for stall avoidance is inherently simple, involving the application of additional power on the manoeuvring engine (by increasing the throttle demand) at a specified low ahead shaft speed. It is known however that the additional power applied and the time of application are fairly critical to the success of the procedure and to ensure repeatability on the shore test, the procedure was implemented as an automatic feature of the system.

An example of a successful avoiding manoeuvre is shown in Fig. 11.

Effectiveness of Stall Break and Avoidance Procedure

The consistency of shaft stall conditions achieved was sufficient to establish the effectiveness of the breaking and avoidance procedure. In terms of stall breaking capability, the ahead and astern breaking procedures were equally effective the stall time being approximately the same for both methods. The astern break procedure is less complicated operationally but produces the more severe machinery conditions (ie, higher torque levels, higher coupling power dissipation). On the other hand, if the ahead procedure is used, the ship stopping performance will be reduced (since astern thrust will be lost during the period of ahead rotation). On balance, the ahead break procedure would be recommended for the ship case.

The stall avoidance trials confirmed that there exists a fairly restricted envelope of additional throttle/application speed parameters within which the avoidance procedure is successful, but that a common set of parameters could be obtained to cover all manoeuvring conditions examined. The procedure is to be incorporated as an automatic feature of the ship control system, and is to be extended to cater for possible stalls coming from astern.

Hydraulic Coupling Characteristics

Performance data relevant to the operating characteristics of the hydraulic couplings were obtained from steady state and dynamic trials.

The torque characteristics, defined in terms of a coupling stiffness coefficient were obtained by driving the gas turbines through opposite sense couplings and, by controlling the power levels on each gas turbine independently, steady conditions were produced on both couplings over the slip range 5 to 200 per cent with various filling levels on the couplings.

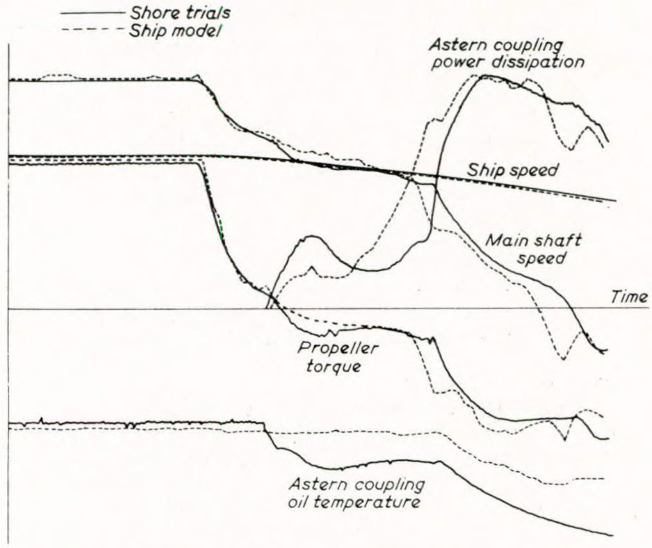

Fig 11—*Stall avoidance manoeuvre*

The torque coefficient obtained is shown in Fig. 12. In general shape and numerically, the data correlated with the previous data obtained from manufacturers and extrapolation from ½ size couplings tests for slips up to 200 per cent. The maximum stiffness obtained was approximately 10 per cent less than that predicted. The characteristics for slips greater than 200 per cent are defined experimentally for the first time. The trials results indicated that the stiffness of the coupling is also dependent on the coupling impeller speed and oil temperature, but these effects do not significantly affect the overall machinery performance.

The coupling torque coefficient was also calculated from measurements made during the power injection manoeuvres. The trajectory of the coefficient throughout a typical manoeuvre is superimposed in Fig. 12. The power dissipation capabilities of the coupling during crash stop manoeuvres is of importance in determining the manoeuvring power which can be applied by the engines. Previous tests on ½ size couplings indicated that there occurred a significant reduction in oil through flow of the couplings, implying an increased coupling pressure, as the main shaft approaches zero shaft speed and since this period coincides with the period of high power dissipation in the coupling, very high coupling oil temperatures would be obtained. There was no known way to extrapolate confidently to the full size coupling. The manufacturers advised, however, that the full size couplings would not exhibit this characteristic and this was in fact confirmed by the trials. From the results the oil throughflow is independent of coupling speed and slip, remaining practically constant throughout the manoeuvre (except for the filling/emptying periods).

From the tests conducted the hydraulic couplings have been shown to be adequately capable of performing crash stop manoeuvres and sufficient margins on coupling temperature exist. The comment made earlier on the accuracy of the oil temperature transducers must, however, be borne in mind.

Other Trials Using Power Injection Systems

A number of other investigations were carried out under simulated ship conditions and were:

a) Demonstration of correct coupling/direct drive changeovers under slam acceleration conditions;
b) Evaluation of the effectiveness of the coupling filling and the transient brake application in disengaging the direct drive clutch.

The changeover from coupling to direct drive was shown to be satisfactory and the trials confirmed disengagement of the clutch could be achieved under all manoeuvring conditions. The transient brake is only required to assist clutch disengagement during manoeuvres from low ahead speeds in maximum displacement and fouled hull conditions.

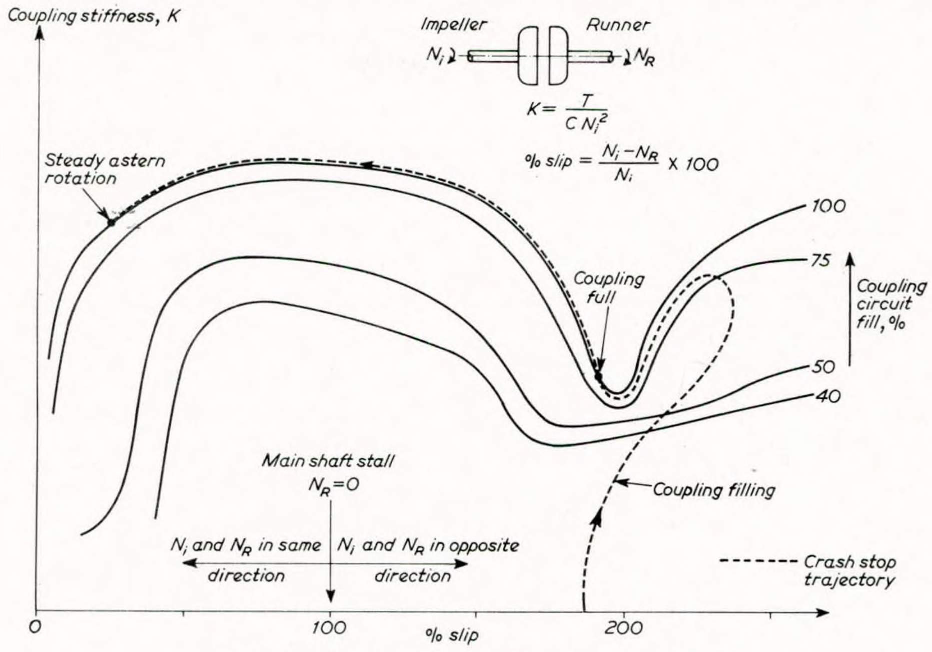

Fig 12—*Hydraulic coupling characteristics*

CONCLUSIONS

These trials have shown that the relevance of ship manoeuvring performance data obtainable from a normal shore trials facility have been greatly enhanced by the inclusion of power injection to simulate the effects of the hull and propeller characteristics on the propulsion machinery.

Accepting that the hull and propeller characteristic data used in the power injection control system are extrapolated from model tests (and which cannot be confirmed until sea trials), the trial results showed that power injection was successfully applied and that adequate close control of the machinery was obtained. Comparisons of trial results and computer predictions using the full sized data (i.e. engines and couplings) showed close agreement and will enable the significance of any variations to the ship performance equations to be assessed. The trials carried out have demonstrated the ability of the machinery to withstand transients and the control system sequencing to respond in the prescribed manner. The postulated problems of turbine underspeed and peak coupling temperature are not sufficiently pronounced to cause mechanical damage. Shaft stall during reversal has been thoroughly investigated, the likelihood is that it will occur during some manoeuvres on the ship. Shaft stall breaking and avoidance procedures originally derived from computer simulations were shown to work satisfactorily. The stall avoidance procedure has now been incorporated into the control system and will provide an automatic fuel addition near zero mainshaft speed. The procedures for breaking stall which can be evoked by the operators should the need arise have also been demonstrated. There are uncertainties and some facets of bearing stiction which cannot be fully explained, however, sufficient trials have been conducted to ensure that the ship case is adequately covered.

As a result of these trials a great deal of previously unconfirmed transient performance data relating to the propulsion machinery and ship control system have been obtained and may be used to assess any changes in ship configuration (eg, displacement).

These trials have confirmed the effectiveness of the power injection technique and represent a significant advance in shore testing technique. The experience gained indicates that computer control of machinery could be used for a wider range of trials in any future shore test establishment and would reduce the time required for running manually operated trials.

ACKNOWLEDGMENTS

The encouragement given by Director General Ships and the directors of Y-ARD Limited and Rolls Royce (1971) Ltd is gratefully acknowledged, as is the generous assistance of the author's colleagues and members of the Joint Trials Group. The views expressed in the paper, are, however, those of the authors.

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DISCUSSION

CAPTAIN A.W. WHEELER, R.N., M.I.Mar.E., said that, before opening the discussion on this paper, he must first declare his own interest. For the past year he had been the Ministry of Defence Project Leader for the Shore Test Facility. The work which had been described was in its closing stages by the time he took over, but he was very soon aware that many people from a number of different organizations were involved and that it had been essentially a team effort. He was sure that the authors would wish him to give due credit to all those others who had contributed and, since some of them were in the audience, he thought they could look forward to a lively discussion.

It would be appropriate for him to say a word about the cost-effectiveness of the Test Facility. The authors had no cause to deal with this question in their paper, but the fact was that the work which they described was not only a notable engineering achievement in itself, but it had also increased the inherent value of the Test Facility by a considerable measure.

Despite the very powerful arguments in its favour, there were some, in the early days, who had questioned the need for this expensive tool. They could derive a certain justification for their view by pointing to the absence of any proven ship dynamics simulation, but now there could be no doubt at all that the Test Facility had been value for money, not least because of the computer-controlled loading system which was under discussion here.

Having said that, he was sure that the authors would be the last to encourage any complacency about the work which would still have to be done during the sea trials of the first anti-submarine cruiser. There were a number of factors which, for various reasons, were excluded from the simulation and there were two in particular about which he would like to learn more.

Reference 3 explained that the inertia of the propeller and that portion of the shafting not duplicated in the Test Facility was discounted, so far as the torque reference system was concerned. Would the authors now say how significant they concluded this added inertia to be in terms of the crash stop manoeuvre ? Then, in the final section of the paper, the authors said that the control system now incorporated an automatic fuel addition, near zero main shaft speed, as the means of stall avoidance. This could not necessarily be regarded as the end of the story, however, because propeller slip had still to be taken into account. Trials in the County Class destroyers had shown that, with the ship's speed dying away it was possible, once the shaft had been reversed to reduce gas turbine power and still continue accelerating the shaft astern. Indeed, it might well be necessary to do so deliberately. Too much power after reversal tended to race the propeller and, when it did eventually bite, large, and possibly excessive, torque peaks occurred. This must clearly be a matter for investigation during sea trials, but he would invite the authors to expand on this point and to speculate on how the control system might be adapted to cope with the problem, should sea trials show it to exist.

In conclusion, they were told that, should stall occur, the operators would use stall-breaking procedures. He thought it would throw useful light on the control system philosophy if the authors were to explain precisely how this option would be exercised, assuming that the initial PDL movement was applied at the bridge control position.

MR. H.A. CLEMENTS said that the paper described a most interesting and clever concept of programmed power injection in order to simulate the water-milling effect acting on a propeller when reversing a ship.

Many papers had mentioned the various advantages of different reversing systems and, whilst most of the world's navies had adopted c.p. propellers for reversing gas turbine driven ships, the Royal Navy had vast experience of reversing gears, associated with the quieter and more efficient fixed pitch propeller. The power injection system, described by the authors permitted such gear to be shore tested and actually simulate a ship reversal, whereas such shore testing of a c.p. propeller system was not practicable.

His first experience of a form of power injection, was in shore testing R.N. County and Tribal class machinery. This was in a very simplified form, however, and was effective to simulate, for instance, the extreme rate of change in relative speeds, which might occur at SSS clutch engagement.

It was in connection with the County and Tribal class vessels that the effects of propeller shaft stall were first noticed, with the associated high heating in the hydraulic couplings. This was simply overcome, he believed, by raising the minimum speed of the gas generator, during such a manoeuvre, so that more power was available to pass quickly from ahead to astern rotation of the propeller.

The problem of shaft stall was most severe when stopping from high ahead speeds and, if the vessel's speed could be reduced rapidly before driving the propeller astern, the likelihood of stall occurring was greatly reduced.

One way to reduce the vessel's speed was to fill the astern hydraulic coupling with oil, whilst the ahead coupling still remained full of oil and the gas turbine was at idling power. This resulted in powerful dynamic braking of the propeller shaft for say, 15 seconds, then the ahead coupling could be emptied and the turbine power increased to drive the propeller astern. After the braking period, more power was available actually to drive the propeller astern.

This braking system was not likely to increase the ahead reach of the ship. In fact, some improvement might be achieved, compared with what might possibly happen if the propeller were reversed prematurely. There was, also, less likelihood of astern rotation of the turbine. In addition, the system avoided the difficult control procedures, described by the authors, to prevent propeller shaft stall. The ahead hydraulic coupling drive was used in the power injection system, described in the paper, whereas the direct drive could have been used with less variables to consider. He presumed however, that this was necessary to reduce the value of connected inertias and to avoid the power turbine rotating in the astern direction.

The gear arrangement with different ratios between direct drive and ahead manoeuvring drive was ingenious, but was not without its disadvantages. When in direct drive, with the SSS clutch engaged, the ahead hydraulic coupling components had relative rotation, i.e. there was not zero slip and the astern coupling components did not have exactly 200 per cent slip. Although the couplings were empty, they were full of air and the windage loss could be well away from the minimum values, as could be seen from Fig. 12. He appreciated that the object of the paper was not to cover such details of design, but it would be helpful to learn whether such windage was considered and measured.

How much more attractive such a gear would be, if the direct drive and coupling drive layshafts all had the same gear ratios, so that, when in direct drive, the ahead hydraulic coupling input and output components were rotating at the same speed in opposite directions (200 per cent slip), thus having minimum losses in both drives. Furthermore, the ahead coupling could then remain full while in direct drive, in readiness to achieve a quick transfer to the ahead hydraulic coupling drive. It would then also be feasible to have direct drive clutches in both ahead layshafts and thus share the turbine power between these layshafts for all cruising and high power operation. This would enable the gearbox weight to be reduced substantially.

MR. C.J. CHARLES, M.I.Mar.E., said that, whilst the title of this paper was concerned with the computer control of a mechanical system, sufficient information was given about the system itself to excite a more than casual interest in its behaviour.

During the presentation, they had learnt a little more about its working characteristics, but the absence of numerical scales on the figures and, in particular, a time scale, made it very difficult to appreciate its performance.

He noticed that power turbine speed and gas generator speed were not presented, which was a pity, particularly because the inclusion of fluid coupling temperature and power dissipation rate curves did lead to a puzzling result in Figs. 8 and 9, where, for constant power input, one saw an increasing temperature rise across the coupling. Was the cooling oil flow rate falling during this period ?

He was intrigued by a reference to a transient brake. How was this device to be integrated into the ship's control system, since he noted that its use was classed as a "last resort" ?

Turning to the core of the subject, computer control, he thought it was reasonable to ask the authors, whether, on due reflection, they were of the opinion that the costs involved, in the design, commissioning, and trials development of the system, were justified ? In other words, would manual control have been cheaper, overall, and in real terms, equally effective in determining whether the machine was capable of performing its design function, and also, enable its characteristics to be explored ?

After all, 78 manoeuvres, allowing five minutes for each, could be completed in one day. He assumed that the trials described here took several months to complete and that an appreciable part of the time was spent in taming the control system to exert a fairly simple form of restraint on the active engine system.

The fact that stiction torque was notoriously difficult to predict, and analyse, would support the view that manual control would have been more appropriate, being more flexible, more economic and more incisive, when a particular feature needed investigation.

He thought the wisdom of adopting the automated approach to the problem should be studied in this case, for the particular influence that the conclusions might have on future shore trials programmes.

Incidentally, the technique of overcoming shaft stall, by delaying the actual reversal until it could be accomplished swiftly, at moderate power, was one which, he believed, had been in use in the Royal Navy for several years.

In the case of the shore trials rig, did any mechanical damage, such as bearing wear, occur as a result of the stiction trials ?

Still in the context of shaft stall, he felt it was a relevant occasion to refer briefly to the possible advantages of using a compound epicyclic gear as a means of effecting propeller reversal in a naval installation.

Although, in terms of energy dissipation, exactly the same problems were presented, the kinematics of the epicyclic gear were such that the phenomenom of gears and bearings being subjected to potentially damaging high loads at low speeds, did not occur.

On the face of it, this removed a source of considerable uncertainty in the operation of high power gearboxes with fixed pitch propellers, and, in view of its other advantages of compactness and functional simplicity, rendered the epicyclic reversing gear a very attractive proposition for future naval application.

MR. B.C. RICHARDSON, M.I.Mar.E., said that the paper commenced with a discussion of the dynamic problem area namely, ahead clutch disengagement, power turbine stall, coupling oil temperature and main shaft stall, and these items were the main justification for providing the power injection system. A situation was depicted in which, during manoeuvring, a dilemma existed. On the one hand, dangerously low power turbine speeds were a consequence of maintaining acceptable coupling oil temperatures, and on the other hand, dangerously high coupling temperatures were a consequence of maintaining acceptable power turbine speeds. The simulation studies and analyses appeared very rigorous and one would expect that, where such trends had been identified, they would indeed occur in practice. However, later in the paper it seemed that, in practice, the dilemma never materialised, as both the power turbine and the coupling oil emerged from a manoeuvre unscathed. The explanation of this departure from prediction was not adequately covered. Did the answer lie in the third parameter, i.e. a relaxation of stopping performance, or was there a flaw in the original analysis.

Coupling oil temperature did not appear to rise drastically during stall and, if this situation did not change during prolonged stall, could one not live with the situation ? Although the traces did not indicate astern thrust at the inception of stall, he felt sure the thrust would be high and contributing a great deal to the ship's deceleration. Eventually, the system would break out of stall when the propeller torque, and therefore thrust, had fallen to a sufficiently low value. Why then was it considered unacceptable to have shaft stall during such manoeuvres. He believed it was more psychological than technical.

The paper stated that 25 per cent of stiction torque (and this was still a considerable amount) was contributed by water lubricated bearings. These bearings would have a stiction torque which was virtually independent of the thrust and torques being transmitted by the shafting and would, therefore, always appear during all manoeuvres. Was it, therefore, likely to cause prolonged shaft stall during slow ahead/slow astern operations when operating in confined waters ?

During transients, the system performance required very rapid changes of load on the propeller shaft. In the absorbtion mode, the conventional dynamometers were incapable of providing such a rapid response and, consequently, the torque deficit must be provided by other means - in this case, a second gas turbine and a good deal of sophisticated controls equipment. Would the authors indicate the amount of torque augmentation requirement from this back-up equipment, or better still, provide figures for the dynamic responses of the dynamometer ? His company had particular interest in this area, since this was one of their reasons for rejecting conventional dynamometers in favour of water cooled friction brakes, backed up with an electric motor (also of course for power injection), again controlled by an on-line digital computer.

The study into the dynamic performance of this propulsion system and in particular, the reversing gearbox, had been occupying a lot of effort for at least six years to his knowledge. Before this study began, many schemes were considered before embarking on a gearbox using reversing fluid couplings. With the experience now gained, would the authors consider that, for propulsion installations of this size, the thrust reversing mechanism described was still the best choice, when compared with alternatives such as c.p.p., partially reversible power turbines (allowing energy dissipation in the turbine as well as fluid couplings), or reversing power turbines ?

LIEUTENANT-COMMANDER W.B. HARRIS, R.N., B.Sc.(Eng.), M.I.Mar.E., said that, in the paragraph on "Application of System to Other Manoeuvres", the paper implied that simulated ship decelerations could generally be carried out by controlling dynamometer torque, without requiring power injection. Experience at Ansty, when carrying out these manoeuvres (mentioned in the paragraph on "Other Trials Using Power Injection Systems"), did not support this contention. To achieve simulated ship decelerations, power injection was required for all manoeuvres attempted. For slow decelerations, it would be possible to simulate adequately by controlling dynamometer torque only. Such slow manoeuvres were not of interest, however, since they did not highlight the extremes of transient hull momentum and propulsion machinery interaction.

MR. C.A. ROWNTREE noted that a figure of $10^{o}C$ to $15^{o}C$ difference, in astern coupling oil outlet temperature, between the updated ship model and the shore trials, was regarded as significant. Could the authors account for this difference, as the model was based on the latest shore trial information ? Were they convinced that the propeller and hull equations were sufficiently well defined for this difference to be significant, particularly as it was possible that similar differences would exist between the two gearboxes fitted in the same ship ?

On the same theme, was the mechanism of the build-up of stiction sufficiently understood ? What actual times were involved ? Did the computer simulation cover a sufficient range of possible errors, in these areas, to give assurance that the ship's case was covered, and could the authors give the degree of possible errors taken into account ?

During the initial power injection manoeuvres, without thrust block loading, three manoeuvres did in fact stall. Had the authors come up with a satisfactory explanation of this; in fact could the mechanism leading to stall now be accurately defined ?

MR. B.J. WRIGHT said that stall, being defined as a dwell at zero speed during an otherwise steady reduction and eventual reversal of shaft speed, was not necessarily more significant to stopping the ship than a dwell at some other point in the speed transition, and a dwell, at any point, might infer high heat dissipation in the fluid couplings. What must be important was that, if the shaft stopped for a finite period, then stiction came into play and prolonged the stall. Stiction must be critically important to the relationship between simulation and practice.

As it was admitted that there were some facets of bearing stiction which could not be fully explained and the STF yielded no information about shaft bearing friction/stiction (even the gearbox bearings were unrealistically loaded), there must remain considerable uncertainty about the likelihood of stall occurrence, its phasing or severity: therefore the stall avoidance system might be unnecessary, ineffective or perhaps only need adjustment to suit shipboard conditions.

To what extent was it envisaged that sea trials would be necessary to assess the actual stall potential and/or check the effectiveness of the stall avoidance system ? It had been said that stall avoidance timing was critical: what adjustment was provided and possibly required ? Alternatively, since it was now known that low p.t. speeds could be accepted and that coupling oil temperatures were conservative, could a stall be tolerated mechanically and what would be the penalty to ships' reach, etc, if application of astern power was delayed slightly, to avoid very high energy dissipations ?

AUTHORS' REPLIES

Dr. Cleland replied on behalf of the authors and in answering Captain Wheeler he stated that the inertia of the shore trials system, downstream of the fluid couplings, in the standard configuration, was approximately 70 per cent of the corresponding shipboard inertia (short main shaft, no propeller). The inertia of the injection power turbine and associated gearbox components was however such that the total downstream of the manoeuvring fluid couplings was:

$$I = I_{SHIP}\left(0.7 + 1.4\frac{\dot{N}_1}{\dot{N}_2}\right), \text{ where}$$

$\dot{N}_1$ and $\dot{N}_2$ are the injection power turbine and main shaft accelerations (referred to a common gear line) respectively.

Thus for true inertia equality, $\dfrac{\dot{N}_1}{\dot{N}_2} = 0.21$

This equality did not hold exactly nor continuously throughout the shore trials manoeuvres, nor was there any attempt to impose it on the system - hence there were "inertia errors" during the manoeuvre. However, the initial system feasibility studies (using computer simulation) showed that the effects of these "inertia errors" would not have a significant effect on the shore trials manoeuvre, and this was confirmed by the post-trials comparison of the ship model results and the actual shore trials results (see Figs. 8 - 11).

Referring to propeller slip after reversal: the propeller characteristics used in this work had been derived from a comprehensive set of steady state and dynamic tank tests, supplemented by measurements made on a County Class Destroyer. They were, therefore, the best characteristics that could be obtained at the present time.

Using these characteristics in computer simulations, it had been shown that very high shaft accelerations could occur immediately after shaft reversal, (especially after a shaft stall), but were not accompanied by torque overloads.

None of the County Class manoeuvring results known in detail to the authors exhibited the torque peak phenomenon referred to by Captain Wheeler, but this could well be due to the relatively low astern powers used in these instances. They considered it unlikely that the problem would arise in this particular case, but if required, speed governing could be used to compensate.

Of the two stall break procedures mentioned in the paper, the ahead stall break was the recommended method since it imposed the less severe torque loads on the system during the stall period. This procedure would be implemented manually by the bridge operator by manipulation of the Power Demand Lever (PDL) as follows :

"On recognition of stall, move PDL towards zero position (to reduce power level to gas turbine idle but maintaining astern drive mode).
When main shaft breaks away in ahead sense, restore the PDL to desired astern power setting, causing the main shaft to again decelerate and achieve astern rotation without stall."

The main shaft would break away ahead due to the ship momentum effect on the propeller, and would reach reasonably high (trailing) ahead speeds in a very short time. The procedure was however very simple, and had little effect on stopping distance. The occurrence of stall and subsequent breakaway would be adequately indicated to the operator by the standard bridge shaft speed indicators.

Dr. Cleland then replied to Mr. Clements and wrote that it was not immediately clear why it was not practicable to simulate the dynamic loading conditions of a CPP propulsion system on a shore test facility. Indeed the paper by Donnelly and Keyser (presented in Session P1 of the 4th Ship Control System Symposium in The Hague in 1975) demonstrated the hardware and control techniques required to simulate the loading conditions appropriate to the US Navy FFG-7 and DD-963 propulsion systems. It was not known whether the automatic load control system had been used in practice - the paper only described computer simulation results - but the technique did appear viable from a technical standpoint.

The County Class destroyers employed various methods of stall avoidance. Different procedures were employed dependent on the particular gas generator speed at which the crash stop manoeuvre commenced - the procedures were generally successful but required a fair degree of throttle manipulation by the operators.

It was true that stall could always be prevented by "delayed action" i.e. by waiting for the ship speed to fall to a level such that, when astern power was applied, the torque from the engines was largely in excess of the propeller torque - thereby producing rapid main shaft deceleration through zero speed.

However, the scheme proposed by Mr. Clements appeared to be an interesting additional mechanism. Without detailed consideration or simulation, full comment could not be made on the real advantages of such a system. The initial retention of an additional coupling in the line would, in principle, provide an additional ship braking effect, but serious consideration would be required on many other aspects, in the main concerned with the physical constraints of this particular system (e.g, volume of couplings, coupling empty/fill rates) but also on the requirements of additional sequencing complexity of valves and scoops.

They had not tried Mr. Clements scheme, but did experiment with the possibility of using the astern coupling on the non working engine (this obviously limited this technique to single engine/shaft operation), with the impeller of that coupling locked. The results showed that the torque developed in this coupling was comparatively small, and reduced very quickly in the course of the manoeuvre - but the method reduced the peak oil temperature by 20°F and the ahead reach by one per cent. The procedure was not however taken further, since alternative control system sequencing and higher oil flow rates provided more significant improvements.

The authors would doubt whether Mr. Clements' scheme would, by itself, produce a significant enough change in ship deceleration to prevent main shaft stall, and would reject the suggestion that the stall avoidance controller was complicated - it was, they suggested, very much simpler in practice than the control sequencing that would be necessary for Mr. Clements' mechanism.

Mr. Clements was correct to assume that the injection power turbine was used driving via its ahead fluid coupling, and not via the direct drive clutch for the reasons given. A slipping link was essential.

Fig. 12 did not show fluid coupling characteristics with an empty coupling, as implied by Mr. Clements. The windage losses were of the order of tens of horsepower, and might therefore be ignored in relation to the design power levels of the system.

It was not possible to measure these coupling windage torques since the power turbine torquemeters had to be calibrated, on site, at speed with no load, but of necessity "no load" included all upstream side components of clutch and couplings - which included windage. The torquemeters were retrospectively calibrated on a special test rig and some information on windage might therefore be calculated from consideration of these latter calibrations with the on site measurements.

The "equal gear ratio" system proposed by Mr. Clements was indeed interesting, and was currently being assessed for viability in a frigate context.

A reply to Mr. Charles followed, and the authors apologised for the lack of numerical data on the Figures - but as Mr. Charles would appreciate, all current and future R.N. projects were subject to security classifications.

The "puzzling result" referred to by Mr. Charles in Figs. 8 and 9 was entirely explained by the fact that the oil temperature rise in the fluid coupling was governed, in common with most physical phenomena, by differential equations. Thus, even for constant power dissipation and oil throughflow, the oil temperature would continue to rise until a static heat balance was obtained. In Figs. 8 and 9, this steady state condition had not been achieved.

The transient brake was fitted on the power turbine intermediate shaft, and as its name suggested, was not for sustained duty but was applied, in sequence with other interlocks, for a very short period (a few seconds) in order to disengage the direct drive clutch - if all other disengaging mechanisms failed. In practice, the brake was required in this role for a restricted set of manoeuvres with a deep and dirty hull condition.

The question on the overall cost effectiveness of the computer controlled manoeuvres was complex, and could only be answered by firstly questioning the technical effectiveness of the computer controlled, system, and then, by somehow equating the technical benefits against the cost of achieving them compared with other less technically complete methods.

It being impossible to make this comparison with any certainty, the authors could not properly comment on the cost aspect, but the following comments might answer the technical aspects of the question.

It was not conceivable that even skilled operators could have handled the required manipulations of two gas turbines and a reversing dynamometer to produce the performance data required to satisfy the objectives of the trials.

The authors were satisfied that all available options in the control complexity sense had been investigated and assessed correctly, and the final decision on a fully closed loop system was the only way of ensuring consistent, repeatable and relevant data on the propulsion system performance in a wide range of manoeuvres. (It was interesting to note that another U.K. project in many ways similar to the present one was also to be tackled using computer control of shore test machinery).

The importance of the trials under discussion should also be realised. The overall machinery trials programme, of which the computer controlled manoeuvres formed only a small, albeit very important, part, was the most complex and detailed ever tackled in this country, and was expected to fully prove the propulsion system capability prior to installation in the ship. These trials were not a "back of the envelope" variety where a few assorted results would suffice, and it was totally outside comprehension that they could be completed in a single day. The 78 manoeuvres were in fact completed in a total of 16 days, covering a period of six months. The trials were planned over such a period to allow as much information to be derived for input to each following series of manoeuvres.

The authors accepted without apology the fact that any control system, not necessarily computer based, needed to be "debugged" and thoroughly tested prior to actual use. In this case, the failure of the control system could have had serious effects on the safety of the machinery, and hence the attention paid to control system design and testing.

While the emphasis of the paper was on computer control manoeuvres, there was a considerable number of trials, associated with the dynamic performance evaluation, which were performed manually - these being basically information gathering trials on gas turbines, fluid couplings, dynamometer and gearbox friction/stiction characteristics and were performed prior to the computer controlled trials.

Mr. Charles had not understood the principle of the stall avoidance control discussed in the paper; with the stall avoidance control active, shaft reversal occurred a few seconds earlier, not later, than it would normally occur. (See previous comments on stall avoidance in answer to Mr. Clements).

There was no mechanical damage which was attributable to the stiction trials.

Epicyclic gearboxes had been considered in the past as reversing mechanisms. They did not have the problems of parallel shaft reversing gearboxes, but they did have their own problems. To date the schemes proposed had been considered and rejected on technical/cost grounds. This was not to say however that such schemes would not be viable in future applications.

Referring to the departure from prediction outlined by Mr. Richardson, it was not thought that this was so significant as suggested. In the initial stages of the simulation work - concerned with the development of the propulsion control system - the low power turbine speed/high oil temperature dilemma existed but could now be resolved by control sequencing due to a relaxation of the restriction on low power turbine speeds. Also of influence in this connection were that the on-engine control unit, and the fluid couplings had been shown to be less stiff than design information suggested - both factors contributed to the less severe (than originally predicted) power turbine speed droop. The oil temperatures were always lower than expected.

There had been no relaxation on stopping performance targets, which remained consistent with the specified constraints in system torque and oil temperature levels.

Continuing their reply to Mr. Richardson the authors stated that in stall manoeuvres, it was true that continuing high astern thrust was being generated and that eventually the main shaft would rotate astern due to the reducing speed, and hence reducing propeller torque. However, for some manoeuvres, the stall time was considered excessive (approximately one minute), and one of the main worries in stall was the possibility of (long term) wear on the bearings, etc., caused by the shaft breaking out of stall with poor lubrication.

For harbour manoeuvring, where the effect of ship's way was less significant, shaft stall was not expected to be a problem; if it did occur:

(a) the stall would be very much less severe (due to lower torque and thrust levels);

(b) the stall avoidance control would easily cope with the problem.

In deceleration and reversing manoeuvres, the dynamometer could not by itself satisfy all the torque requirements if shipboard conditions were to be simulated - hence the need for an active power source, or "torque augmentation", as Mr. Richardson put it. The maximum torque input from the injection engine might be taken as occurring at main shaft reversal, where the dynamometer exerted no load on the system. At this stage, the "propeller torque" must be provided by the injection engine alone. Security reasons prevented the authors from quoting numerical data in power levels, but it could be taken that at reversal, the power and torque levels of both the driving and injection engines were of approximately equal magnitude, but of opposite senses (in terms of torque application to the main shaft). The dynamic response of the dynamometer could not be uniquely defined in terms of simple numbers. The characteristic behaviour was non-linear, dependent on present load, water inflow rate and outflow rate (which in turn was controlled via load control and back pressure valves). The response to increasing load conditions was very much slower in general terms than for load reduction, and this had been taken into account in the design of the control system. The dynamometer used for this project was of much higher load absorption capacity and of different type than the dynamometer which was initially considered for Mr. Richardson's application, and would have significantly different characteristics of operation and dynamics.

On fluid couplings: much work had been done, and was being done on the various reversing mechanisms for naval warships. Each proposed system would have its "pros and cons" which must be balanced for each particular case.

Lieutenant-Commander Harris had corrected the false impression given in the paper that simulated ship accelerations and decelerations were generally possible without power injection. Power injection was required for all reasonably rapid deceleration manoeuvres, where overrun occurred, particularly those associated with the investigation of the mechanisms of clutch disengagement.

To Mr. Rowntree the authors said that the differences existing between the predicted and measured maximum values of coupling outlet oil temperature during the simulated crash stop manoeuvres could arise in either of two ways, viz. either the predicted values were higher than reality, or the measured values were lower than reality.

The oil temperature model used to predict temperature rise was a lumped parameter model and effectively calculated the temperature of oil within the coupling, assuming a uniform temperature distribution within the coupling. This was an approximation and would lead to temperature errors, principally in dynamic conditions.

It was however considered that the major source of temperature error arose from measurement inaccuracies - due mainly to the placement of the temperature transducers in the coupling outlet drain. A series of further experiments were to be conducted on the Shore Trials Facility to determine the most suitable transducer location.

Differences in temperature rise between the two gearboxes on the ship would probably exist (due to non symmetrical power application, differences in outflow rates etc) but were not expected to be of significant importance, and certainly not as high as the present model/measured differences.

The authors did not claim to be able to fully explain the intricate mechanisms of stiction, but considered that they sufficiently understood the phenomena to draw the conclusions that they had. They knew from the behaviour of the propulsion system - both in power injection trials and in a series constant acceleration/decelation reversals through shaft zero - that the effects of stiction were consistent with the cause/effect model developed from work on County Class destroyers. The mechanism of stiction was complex, being directly caused by the breakdown of the oil film between rotating and static elements of the system.

They had had discussions with various "Tribology" institutions, but had not been able to obtain any clearer or more definite description of this mechanism.

In broad terms, the stiction model assumed that the build up of friction coefficient in the system as the rotational speed approached zero was related to the torque being transmitted, the thrust at the thrust block, and the weight of the shafting - these factors all tending to destroy the oil films between the rotating and static elements in the system. It was assumed that the build up rate from the normal running (dynamic) friction levels to the stalled shaft (static) levels was exponential and related to the system torque levels i.e.

$$Qf = f(Qt, T, W)\, e^{-t}/Tc$$

Where Qf is friction torque

Qt is transmitted torque

T is thrust at block

W is weight of rotating elements

Tc is build up time constant and is inversely related to Qt

The model was conceived and validated using County Class trials data, such that the model could reproduce the stall/no stall situations achieved during these trials.

The authors considered that sufficient degree allowances for the effects of errors in the model had been taken into account. It had been shown, for example, that the main shaft would stall even:

- if the static friction coefficients were 60 per cent less than predicted.
- if the build up rate was 50 per cent less than predicted (if faster than predicted, stall was even more certain).

Additionally, the stall avoidance and stall break procedures had been shown to be effective using coefficients of 2 x predicted values. This had also been shown, to a lesser degree, in practice by the use of extremely high thrust block loads (see paper).

Analysis of the manoeuvres which stalled without thrust block loading had shown that the difference between driving engine and injection engine torque levels, immediately prior to stall, for these particular manoeuvres was considerably less than other manoeuvres. This net torque had to be controlled to within typically two per cent of engine torque to maintain machinery conditions appropriate to the expected shipboard conditions and this was not achieved for the three manoeuvres referred to by Mr. Rowntree.

Answering Mr. Wright: As discussed earlier, shaft stall did not have a severe penalty on ship stopping distance. The more significant effects were likely to be increased wear of the gearing, etc. and reduced ship handling.

Stiction was critically important to the relationship between simulation and practice; and ipso facto, this was reflected on the attention paid to the subject in this study.

The authors would dispute the statement that "the STF has yielded no information about shaft bearing friction/stiction". This was simply not the case. They would accept that, of necessity, due to the power injection configuration, the bearing loads on some gearbox elements were different from the normal ship case, but provided this was known, and given that they had measured all appropriate torques and speeds in the system, the gearbox manufacturers and the shipbuilders should be able to relate the bearing loads actually experienced to the appropriate ship case with little difficulty. As they admitted, there was still uncertainty but not, they suggested, as considerable as Mr. Wrignt would imply. The various breaking and stall avoidance procedures had been shown to be effective over a wide range of stiction parameters to cater for these uncertainties. Provision had also been made in the hardware associated with the stall avoidance controller to allow a reasonable degree of on-ship tuning.

Ship trials would still be necessary to finally prove the machinery system in its real environment, and they expected to have to rethink some of the ideas on stall/stiction. They looked forward to the prospect.

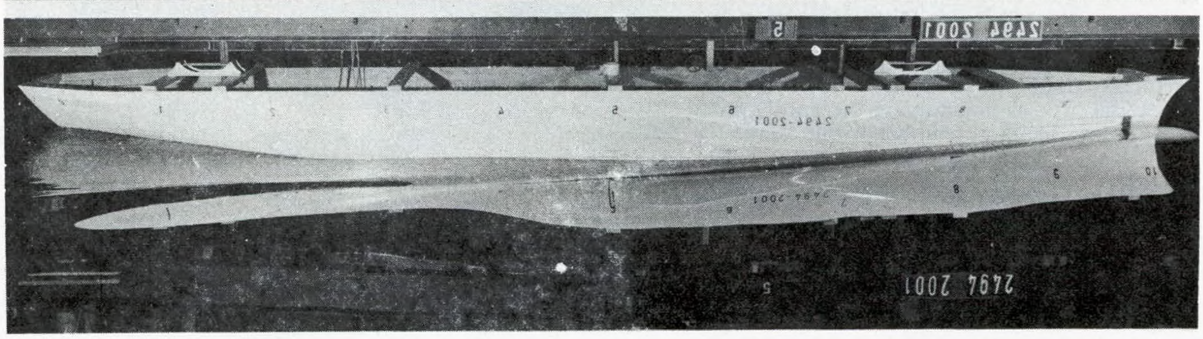
Paper read at the Institute on Tuesday 18 January 1977

DESIGN OF A HIGH SPEED SINGLE-SCREW CONTAINERSHIP

Dipl.-Ing. Hans Langenberg* and Dipl.-Ing.
Gerd O. Andersson*

The authors show that propeller excited vibrations are a main problem in single screw containerships — it must be considered in the early stage of design and given the same importance as speed and stability. This is the lesson they learnt from the production of six of these ships which have low vibration levels. Both model and full scale measurements substantiate this statement.

Additional investigations on potential future containership designs have been carried out and the authors conclude that the fast single screw containership of limited size is likely to become the ship of the early eighties.



INTRODUCTION

About ten years ago the first cellular containerships were ordered and built at European Shipyards, representing a courageous step by shipowners into the unknown economic environment of containerization in which the ships would operate. Containerships being specialist ships are sensitive to environmental changes (physical, economic, regulatory, etc.) which were difficult to predict before the initial dramatic expansion of the container system. Some of the original ships have some limitations due to this lack of information.

The course of development of containerships can be represented by a curve AB taken from reference (1) and which is shown in Fig. 1. This curve shows how an advanced concept—such as containerships—improves its efficiency

* Blohm + Voss AG, Hamburg

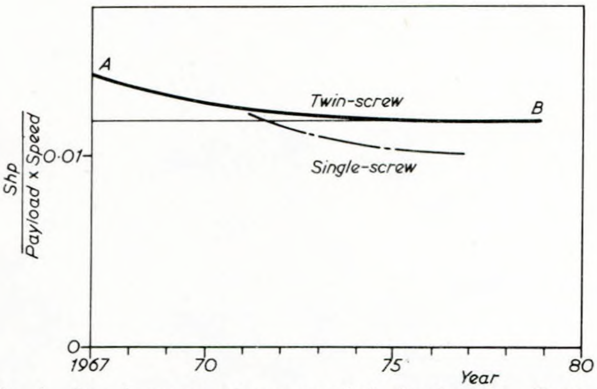


Fig. 1—*Development of a new technical principle—Curve AB represents the improvement in a new technical principle versus time of development as applied to high speed containerships (24·0 kn)*

rapidly in its first phase then it approaches asymptotically the maximum attainable efficiency shown in curve AB. The maximum efficiency can only then be improved upon by innovation or change in basic technology. As ordinates the inversion of a kind of "economic efficiency" (2) is taken which is represented by the engine power per speed times payload on a basis of time. The authors hope that the year 1976 has been allotted the proper position, and they assumed that significant improvements in profitability of 'conventional' containerships is improbable and will only occur if there is a change of either technology or the economic environment.

A change in technology is the replacement of the well tried twin screw propulsion plant for high powered containerships by the less expensive and more efficient single screw arrangement (above 35000 hp and 24 kn). Blom + Voss has designed and delivered between 1973 and 1976 six high speed single screw containerships (25·3 kn under trial conditions of 9·6 m draught with 36000 shp—26480 kW) for the company 'Aktieselskabet Dampskipsselskabet Svendborg/Dampsskibsselskabet av 1912 Aktieselskab'—Kopenhagen (Maersk—Line).

These "compact"—ships demonstrate, apart from other interesting features, that propeller induced problems (3-5) in high powered single screw ships can be solved up to 40000 hp (29420 kW) and that there is a good chance for success at even higher powers.

Main Steps in the Design Procedure

The paper includes a report of the main steps in design procedure of the Maersk-ships and their respective considerations on the background to the yard's prior investigation into future containerships. Results of tank tests and sea trial measurements are given particularly with reference to propeller excited vibrations. Finally, the authors dare a prognosis on a possible next generation of economic containerships by presenting some recent tank test results.

The authors frankly confess that their company's research and investigations described in this paper do not have the perfection and accuracy of a scientific work as the design was dominated by the task of maintaining the yard's production and in fulfilling the building contract within minimum time and cost.

DESIGN CHARACTERISTICS AND DESIGN PRINCIPLES AS SEEN BY THE SHIPYARD

The Design Process

Background

The yard entered into discussions with the owner early in 1973 based on the following information and criteria:

a) Technical experience gained during the development and construction of containerships for about 11000 t.e.u. in total.

b) Fundamental investigations carried out on containership design resulted in guidelines for cheaper production and the potential reduction of engine power for future ships. Leaving out intricate mathematical analysis of available (or not available) economic data, this work was done by selecting competing designs or configurations using simple technical criteria regarding the "sailing vessel" — such as lighweight or fuel storage per ton payload and the "power per speed times payload ratio mentioned above.

c) Tanktest results of a single screw hullform (under the designation HSVA 2437) with low block coefficient ($C_B = 0·5$), low L/B (5·8) and high KM_0/B-ratio (=0·475), and with a pronounced V-type character of frames.

The results of resistance—, propulsion—, manoeuvring— and seakeeping tests confirmed the capability of this hullform-concept in general (results of resistance and propulsion tests are given in Fig. 15). Fig. 16a shows the afterbody of this model and the streamlines indicate that the propeller-inflow in way of the upper disc area calls for improvement.

The yard started with the general conception that considerable advantages for both the shipyard and the owner could be expected by application of the single screw propulsion system for high-powered fast containerships with a hullform of low resistance and highest possible KM_0/B ratios (0·45—0·47). The latter statement being supported in the following paragraphs.

Stablility is an extremely important requirement for containerships as the cargo has a high stowage factor and, therefore, must be piled high. Stability therefore has to be considered in any technical comparison of container carrying ships.

The ratio P_D /payload x speed requires for its calculation a loading condition for the ready-for service, seaworthy and loaded ship which must be based on fixed data such as:

i) weight of a container;
ii) possible deviation of the cargo's centre of gravity from its volume centre (since unhomogeneous stowage can be expected in general);
iii) tolerable draughts (Panmax);
iv) steaming range;
v) minimum GM_0.

This "standard loading condition" could be a method of defining the actual "container capacity" of containerships.

Returning to the ratio P_D payload x speed:

The cargo weight which has to be considered is mainly dependent upon the lack of stability. The higher the ship's form-stability, the more cargo can be loaded without the need of extra ballast water or, in other words, the higher the KM_0 the less ballast water is required. The resulting smaller displacement would be of advantage for the required propulsive power. This has a particularly strong influence if the main dimensions of the ship are subjected to restrictions. For this reason the desirable extension of KM_0-ratios is favoured by a low block coefficient C_B providing this is combined with a large waterline coefficient C_{WP}. This usually leads to extreme V-type frames in the aftbody which—in spite of benefiting ship's speed creates unfavourable or even intolerable wake peaks in way of the propeller's upper disc area (7).

An alternative giving a more reasonable solution is the "pram-type" aftership with its wide U-frames (8). Before deciding on either slender V-frames or wide U-frames the

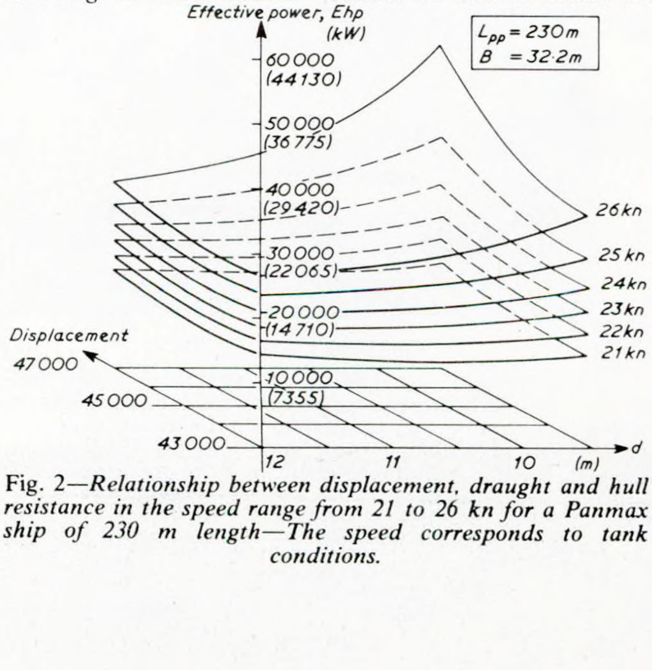


Fig. 2—*Relationship between displacement, draught and hull resistance in the speed range from 21 to 26 kn for a Panmax ship of 230 m length—The speed corresponds to tank conditions.*

ship's strength must be considered briefly.

Low lightship weight is of advantage for low ship-resistance since in case of restricted dimensions each additional tonne to the displacement increases the block coefficient. This in turn adds a whole chain of extra weights caused by the higher engine power required as a result of higher C_B. Fig. 2 shows the effect of draught and displacement on ship's resistance for a 230 m Panmax-ship.

Thus, higher fuel consumption requires larger fuel tanks which adds to the steel weight and increases the deadweight. Ultimately a change to twin screw propulsion may become unavoidable if the maximum propeller thrust is not to be exceeded.

The steel weight for a given ship, will depend more or less on the external loading of the hull; i.e. the static and dynamic forces and moments (9,10) which can be controlled by good design. The combined stresses occuring in the ship's upper girders arise from the still-water bending moments, which for this type of ship will be of a hogging character (tension in the upper deck). The still-water bending moments can be reduce the cavitation problems of the propeller and, therefore, its excitation forces.

b) Increasing the draught and/or reduction of light ship weight means finer lines. This improves the flow to the propeller in general—provided the angles of run do not exceed about 20 degrees. Improvement of flow is essential for low level propeller excitations (11).

c) Reduction of steel weight in way of the afterbody enables the ship's LCB to move forward which again can help to improve the afterbody-lines and ensures a good flow to the propeller.

Details of the Maersk Ship Design:

The owner's inquiry for a ship of about 210 m length, able to carry 600 x 40 ft.-containers weighing 20 tons each, to be arranged in a total of eight tiers (two of them above hatches), was answered with an offer for a ship of about 190 m length but increased beam to carry the specified number of containers in nine tiers (seven below hatches) (12).

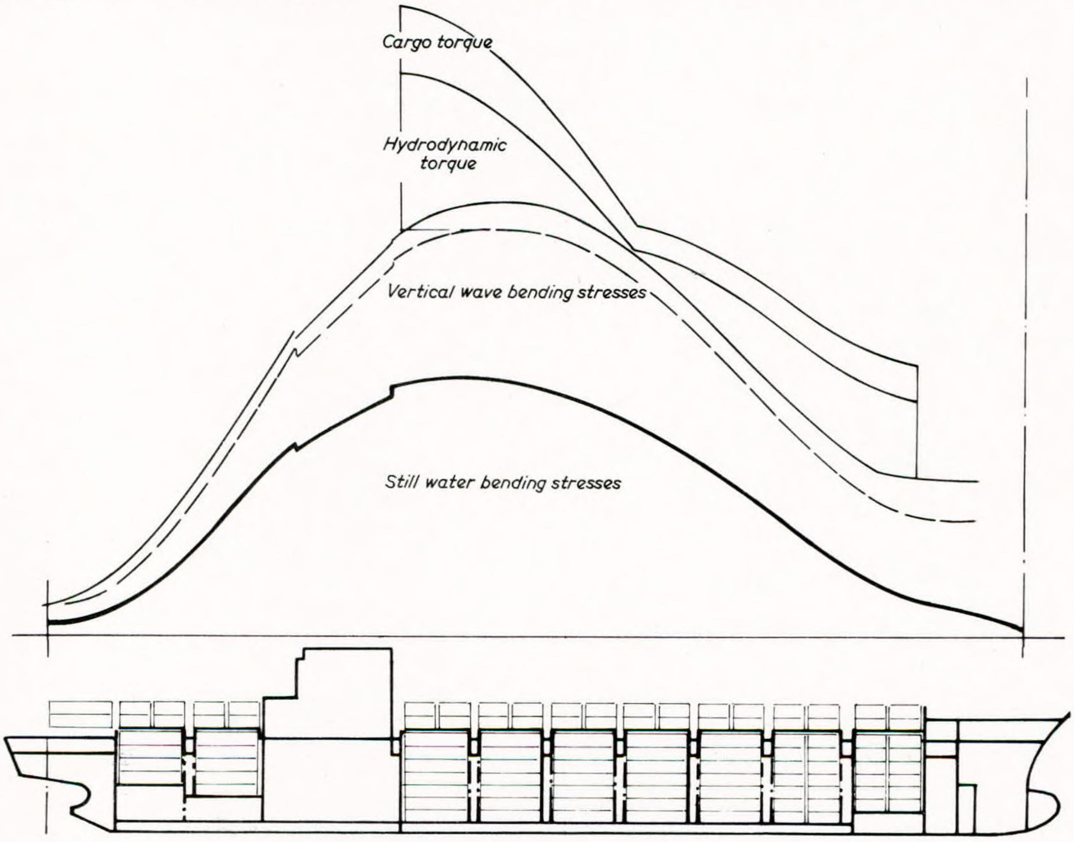


Fig. 3—*Prior state of the Maersk ships' development showing composition of total stresses in the upper deck for the critical condition—Oblique sea from aft.*

minimized by avoiding large weights at the ship's extremities. Thus, these parts of the hull should be designed with slender sections with stores accommodated amidships (Fig. 3).

Reduction of flare in way of the forward part of the hull can help to reduce the dynamic torque (9). In the authors' opinion therefore slender ship ends would be of help to reduce the steel weight in general.

It must also be remembered that the dimensions of the major part of the hull's steel will be determined either by local strength requirements or in accordance with the needs of vibration rigidity, which is of particular importance. Here again a sharp V-framed afterbody seems to be advantageous.

The relationship between a vessel's strength and her hydrodamics has been briefly described and the following general remarks apply to the solution of the propeller excitation problem:

a) Increasing the draught and/or reduction of light ship weight will help to reduce considerably the ship's resistance. The resulting thrust-reduction helps to

It was on this basis that a series of tank tests was started at the "Hamburgische Schiffbau Versuchsanstalt" (HSVA). A total of 15 model versions was tested. Table I gives a survey of the results:

A detailed description of the process of hull form optimization under the aspect of resistance is given in ref. (8).

The ship's final dimensions and main data are given in Table II.

Results of Resistance and Propulsion Tests:

A comparison with the well known Taylor curves (the chosen hullform corresponds to a certain degree with the Taylor-models)—showed that the first version had unexpectedly high residual resistance coefficients, particularly at lower Froude's numbers (13). The explanation found during an advanced stage of investigations was that in spite of the low block coefficient C_B partial separation occurred at the afterbody as a consequence of the low L/B-ratio. Consequently,

TABLE I—CHARACTERISTICS OF MODEL VERSIONS

	HSVA-Model No.	L*/Lpp	L*/B_{WL}	BWL/T	C_B*	CM	Cs	Xv (%Lpp)	KMo/B_{WL}	Bow Bulb l (%L*)	Bow Bulb b (%B)	Bow Bulb h (%T)	Bow Bulb F (%Θ)	P_E/P_D for 25.3 Lu	Aftership Version	Scale of Model
1	2494-1001	1·070	6·88	3·10	0·498	0·928	2·62	49·5	0·448	3·41	18·8	58	12·5	0·740*)	long conical stern bulb	29
2	2494-2000	1·070	6·88	3·10	0·498	0·928	2·62	49·6	0·448	3·41	18·8	76	14·5	0·738*	"	
3	2502-1000	1·060	6·79	3·12	0·503	0·919	2·57	51·3	0·455	3·07	17·4	68	12·8	0·729*	narrow skeg	29
4	2502-2000	1·070	6·84	3·15	0·500	0·919	2·55	51·4	0·454	3·41	18·4	74	14·0		"	
5	2502-5000	1·089	6·98	3·15	0·493	0·919	2·53	51·5	0·453	5·50	18·4	74	14·0		"	
6	2511-1000	1·080	6·88	3·14	0·490	0·922	2·60	50·3	0·454	4·46	18·8	77	13·8	0·738*)	AG "Weser"-bulb	29.32
7	2511-1001	1·080	6·88	3·14	0·489	0·922	2·60	50·3	0·455	4·46	18·8	77	13·8	0·750*)	narrow bulb	29·32
8	2524-1001	1·080	6·90	3·15	0·488	0·922	2·55	50·4	0·455	4·40	18·3	73	14·0	0·743*)	narrow skeg	29·55
9	2524-2001	1·081	6·93	3·15	0·482	0·922	2·55	50·5	0·454	5·10	18·2	73	14·0		"	29·55
10	2524-4001	1·101	7·03	3·17	0·478	0·922	2·53	50·6	0·453	6·41	18·2	73	14·0		"	29·55
11	2524-1002	1·080	6·93	3·18	0·490	0·920	2·58	50·8	0·463	4·40	18·3	73	14·0	0·694*)	stretched afterbody narrow skeg	29·55
12	2524-5002	1·080	6·93	3·18	0·489	0·920	2·59	50·8	0·464	4·40	12·3	73	9·5	0·717*	stretched afterbody narrow skeg small funnel	29·55
13	2532-1000	1·075	6·93	3·18	0·490	0·920	2·59	50·8	0·463	4·08	15·7	74	12·0	0·698*)	narrow skeg	30
14	2532-2000	1·081	6·97	3·18	0·492	0·920	2·59	50·9	0·463	4·76	16·5	74	13·5		"	30
15	2532-4020	1·081	6·97	3·19	0·493	0·920	2·58	50·9	0·462	4·76	16·5	76	13·7	0·731	small B+V—sternbulb	30

* With Stock Propeller

TABLE II—MAIN DATA OF THE MAERSK SHIPS

Length overall	210	60 m
Length pp.	194	50 m
Beam moulded	30	50 m
Depth to upperdeck	18	70 m
Depth to freeboard deck	14	86 m
Draught (scantling)	11	19 m
Corresponding deadweight	257	10 t
Trial draught	7	80 m
Container capacity:		
20 ft. units below deck	102	
40 ft. units below deck	339	
20 ft. units on deck	436	in 2 tiers
Total capacity on basis of 9 tiers	1216	t.e.u.
Tank capacities: Fuel	60	60 m³
ballastwater	60	70 m³
Complement	41	
Propulsion Plant:	A. G. Weser/General electric steam turbine	
Output m.c.r.	36 000 s.h.p. (26480 kW)	
Propeller speed	985 rev./min.	
Class	Lloyd's Register of shipping + 100 A1 + LMC, UMS	

the form was stretched by 1·5 per cent of Lpp from 0·25 L aft. This obviously improved the flow in way of the afterbody to such extent that apart from the advantage for manoeuvring qualities and cavitation risk, the prior resistance curve was surpassed by about 1000 ehp (735 kW).

Fig. 4 shows some results of resistance and propulsion tests on four typical model versions (referred to as versions I to IV) indicating the course of development for the "Maersk" ships. The differences are described in Table 1 and (8). The afterbody plans are shown later in Fig. 6. The test results are valid for tank conditions. a power requirement of about 3000 hp (2200 kW) has to be taken into account to allow for bilge keels, wind (BN 2), and shaft losses. Propulsion tests of versions I to III were run with a stock propeller and version IV with a final wake adapted propeller. Furthermore, the displacement of version IV was about 700 tonnes greater than the displacement of the preceeding versions due to lengthening and the additional requirements of the owner and the classification society.

For version IV a small B+V sternbulb was adopted. A patent was obtained later and is described below. As this version was not tested with a stock propeller, it is uncertain whether or not the considerable power savings with version IV were mainly due to the final propeller.

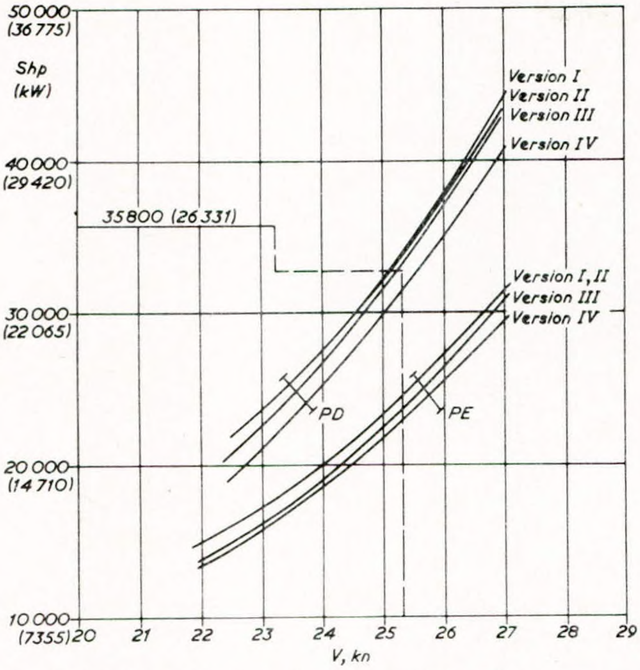


Fig. 4—*Tank test results of typical model versions*
Version I: HSVA-Model 2494-2001
Version II: HSVA-Model 2511-1000
Version III: HSVA-Model 2532-1001
Version IV: HSVA-Model 2532-4020

Speed Trials

The speed trials were performed by using the 'Decca-method' in the area "Skagen-North". The depth of water was about 300 m. As *Anders Maersk* happened to find very fine weather conditions during her speed trials (BN 1 — 2), the results of these measurements are described.

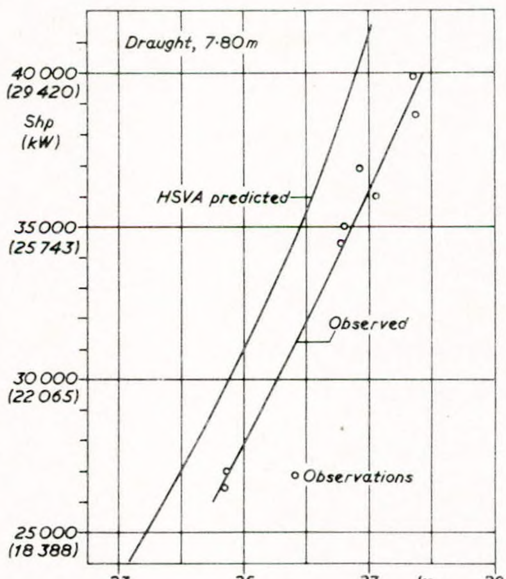


Fig. 5—*Power curve of speed trials for draught T=7·8 m —The observed points on trial runs deviate considerably from the tank predictions demonstrating the uncertainties of scale effect experienced with almost every new type of ship*

The vessel was ballasted down to a draught of 7·80 m and floating on an even keel. Eight single runs were carried out and the observations corrected — so far as necessary — for each run separately. Fig. 5 shows the power versus speed curve for the trial draught. The calculated extrapolation to contract draught of 9·6 m reveals that the ship will be able to sail at about 26·0 knots at nominal power.

INVESTIGATIONS OF PROPELLER INDUCED VIBRATIONS

Excitation

The shape of the afterbody was developed not just to produce satisfactory resistance and propulsive properties. From the beginning close attention was also paid to achieving uniform propeller inflow with respect to vibrations. The stern bulb finally applied aims directly at avoiding excessive radial and tangential wake components. To render this possible, the stern bulb was given a shape which would generate a partial negative pressure immediately before the propeller, so as to straighten the flow in the axial direction. The slimness of the afterbody is believed to generate only small transverse wake components.

Since the Maersk ships would be filled with a steam turbine plant, vibrations would mainly be excited by the propeller. Investigations on vibrational behaviour concerning the excitation were concentrated upon questions of wake, cavitation, and propeller excited forces, i.e. shaft forces and hull forces.

The Wake Field:

Fig. 6 shows the velocity distribution in the wakes of the four model versions mentioned above. The figure also contains the afterbody plans in order to convey an idea as to how the frame contours determine the wake fields.

It is not yet possible to give a non-controversial interpretation of the wake velocity distribution with regard to causing vibrations. The authors would like, however, to comment on the wakes.

In wake field I, one should be concerned about the wake peak in the lower part and the dent above the hub. The first is generated by the dead wood beneath the bossing. The latter implies the assumption of a vortex originating at the bilge (3). Both phenomena cause non-uniform flow into the propeller. Locally changing loads on the propeller blades may give rise to vibration exciting forces. Due to its rotational symmetry wake field II will commonly be accepted as "good". Wake fields III and IV refer to the hull form with lengthened afterbody, with and without the final stern bulb. Field IV shows the effect of the bulb wake with properly rounded wake lines in the lower part. The wake lines of both fields are more extensively flared as compared to the others shown thus reducing the circumferential wake gradient and consequently the dynamic impact loads of the propeller blades. The shape of wake field IV is considered advantageous as it produces small propeller excited force fluctuations providing one is free to choose a "suitable" blade number.

Wake Analysis:

The amplitudes of the first ten harmonic components of the axial velocity of wake field IV are shown in Fig. 7. It should be noted that absolute values are given, but as a matter of fact only the sixth component changed sign at about $r/R = 0{\cdot}5$. These amplitudes are compared with the wake harmonics of a twin-screw container vessel. The comparison is considered instructive since it reveals how far vortex free propeller inflow on Maersk models has been obtained.

The values serving as a comparison are taken from ref. (4) where the respective wake field is referred to as No. 6.

It can be seen that only the first two or three components of the Maersk wake are considerably higher than those of the twin-screw vessel, there is, however, a tendency towards lower values with increasing order of components. It is thought that, apart from a higher wake number, a remarkable similarity to a twin-screw axial wake field is indicated.

The three spatial velocity components of the twin-screw wake field are reproduced in Fig. 8. A three dimensional wake survey of Maersk models is only available of model III. Though the results of these measurements cannot directly be applied to the final model IV (main alteration being another bow bulb and stern bulb attached to the stern frames) they may well serve to estimate the orders of magnitude of the two transverse wake velocity components.

Radial and tangential velocity components of the wake field III are depicted in Fig. 9. The maximum values of both

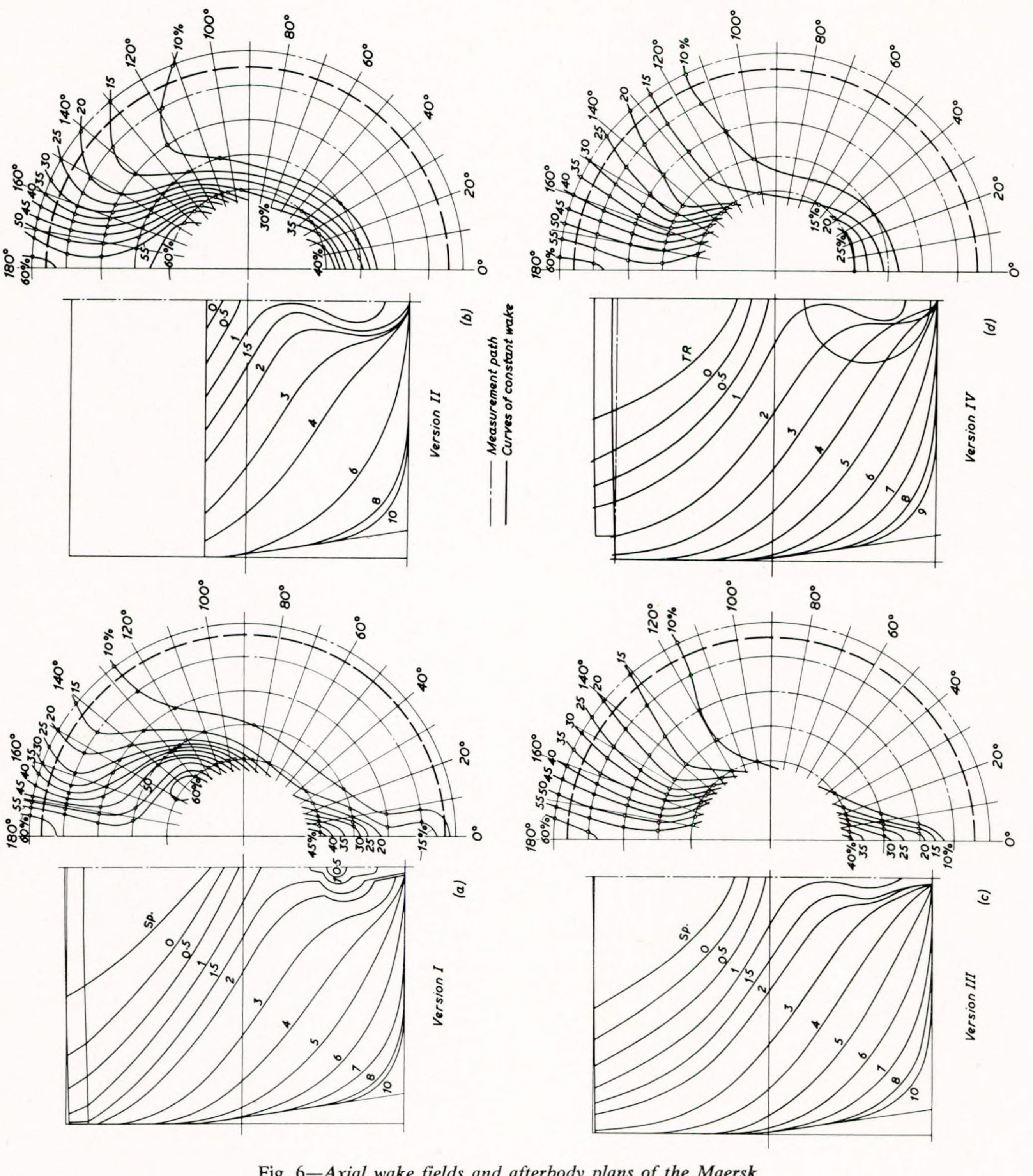


Fig. 6—*Axial wake fields and afterbody plans of the Maersk model versions I to IV*

components are of the order of 10 per cent of the undisturbed velocity. These may be compared with Fig. 8 which shows maximum values of both components of 15 to 20 per cent of the undisturbed velocity in case of the twin-screw vessel.

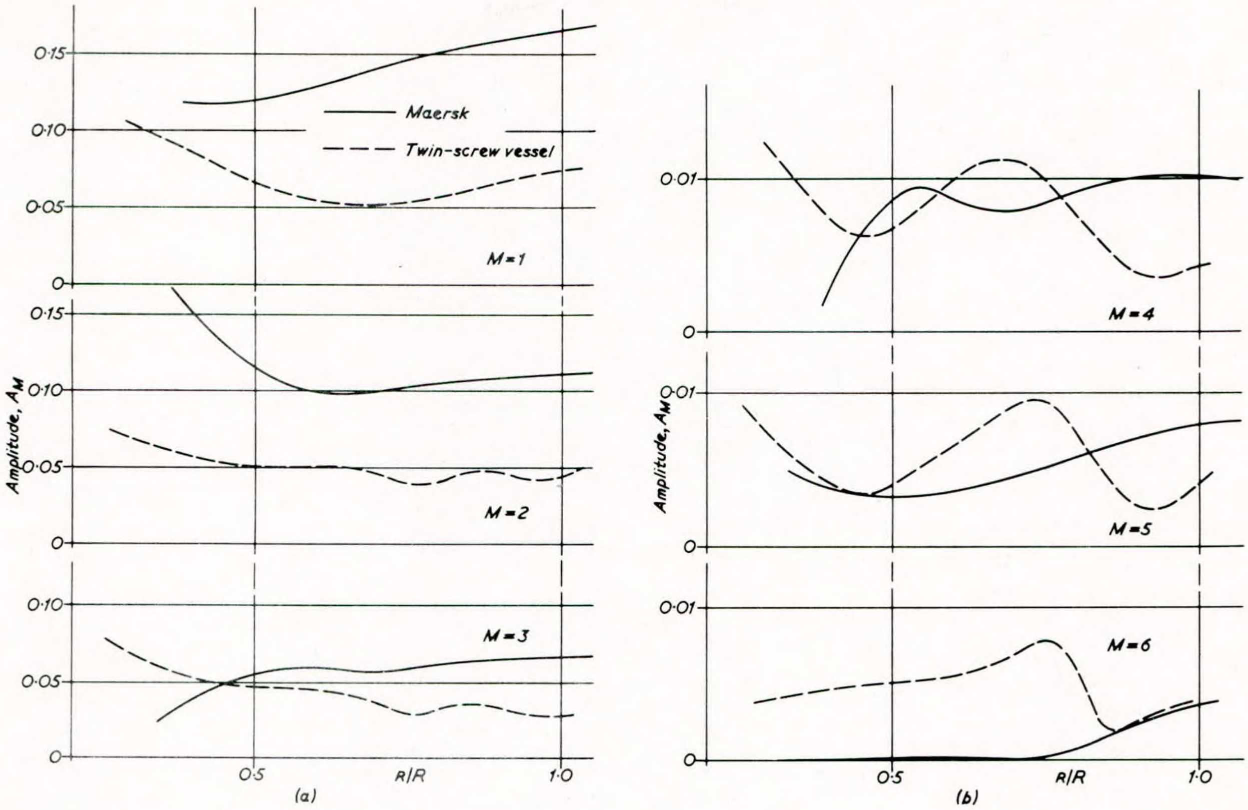
It is concluded that the final model wake shows reasonable uniformity of both axial and transverse velocity components. The smallness of higher order harmonics indicates a propeller inflow with rather small local disturbances.

Calculation of Propeller Excited Forces

When in 1973 the yard started designing the Maersk ships the available information regarding the amplification phenomenon of propeller excited hull forces by cavitation (5) was scarce as were the guidelines for the practical design procedure. It was also of little help that a small first order wake harmonic had been recommended for small pressure fluctuations on the hull as well as small thrust eccentricity. The quality of afterbody lines producing the required wake property was totally unknown.

A means of controlling the magnitude of hull forces was applied by allowing generous propeller-hull clearances. Table III shows the clearances of the Maersk ships as compared to the recommendations of Lloyd's Register. The latter being exceeded by about 50 per cent.

Moreover it was considered advantageous to choose



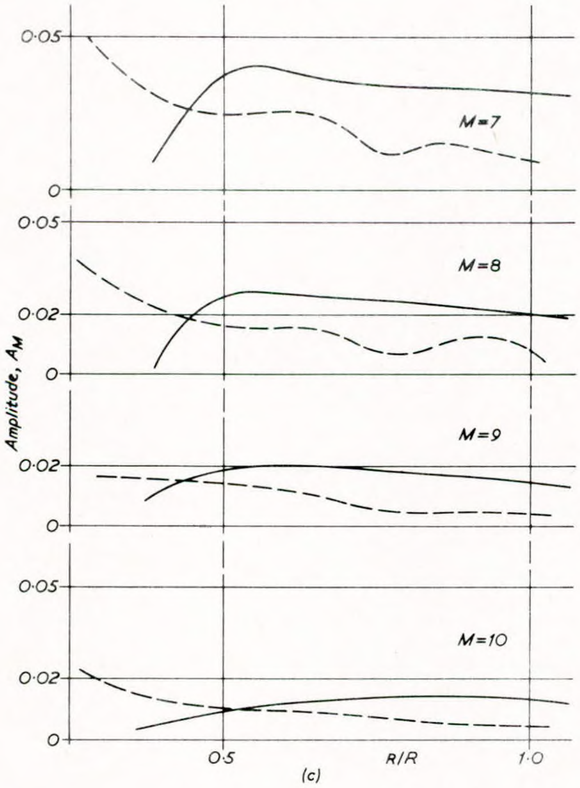


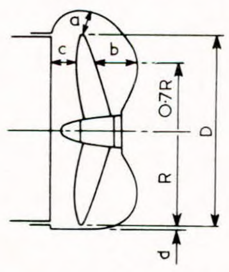
Fig. 7—*Comparison of the amplitudes of the axial wake harmonics of the final Maersk model version IV and a twin-screw containership (m = order of harmonic component)*

V-shaped frames above the propeller (Fig. 6), with object of removing areas of the hull exposed to pressure fluctuations as far as possible from the pressure source.

In effect no measured information as to how Maersk hulls are affected by pressure fluctuations is currently available. Measurements on scale models could not be carried out due to lack of time and the results, due to unknown scale effects, would also become questionable.

An evaluation of the propeller induced shaft forces has had more influence on the design concept than considerations on hull forces. According to the HSVA method, the hydrodynamic propeller forces were calculated for 5-bladed and 6-bladed propellers. The relative importance of the calculated forces and moments can be judged by virtue of values *Kx* (combines thrust and torque Fluctuations) and *Kyz* (combines the fluctuations of the two transverse forces and bending moments) which are opposed to limiting values. These values are based on the experience of HSVA and mark upper limits beyond which vibrations have caused complaints.

TABLE III— PROPELLER-HULL CLEARANCES IN PERCENTAGE OF D



	LR	Maersk
a	19	30
b	28	55
c	12	26

Table IV shows *Kx* and *Kyz* values of model IV for the 5- and 6-bladed propeller respectively, and the limits of complaint.

The 6-bladed propeller evidently evokes smaller shaft forces than the 5-bladed propeller, and can be regarded as tolerable. Furthermore, it can be concluded from Table IV that, in the same wake, a 6-bladed propeller is acceptable whereas a 5-bladed propeller must be rejected. Consequently, a 6-bladed propeller was chosen for the Maersk ships. This choice had the advantage that the overall induced vibrational energy of the propeller would diminish due to the high blade frequency since a decrease with increasing number of blades can be presumed.

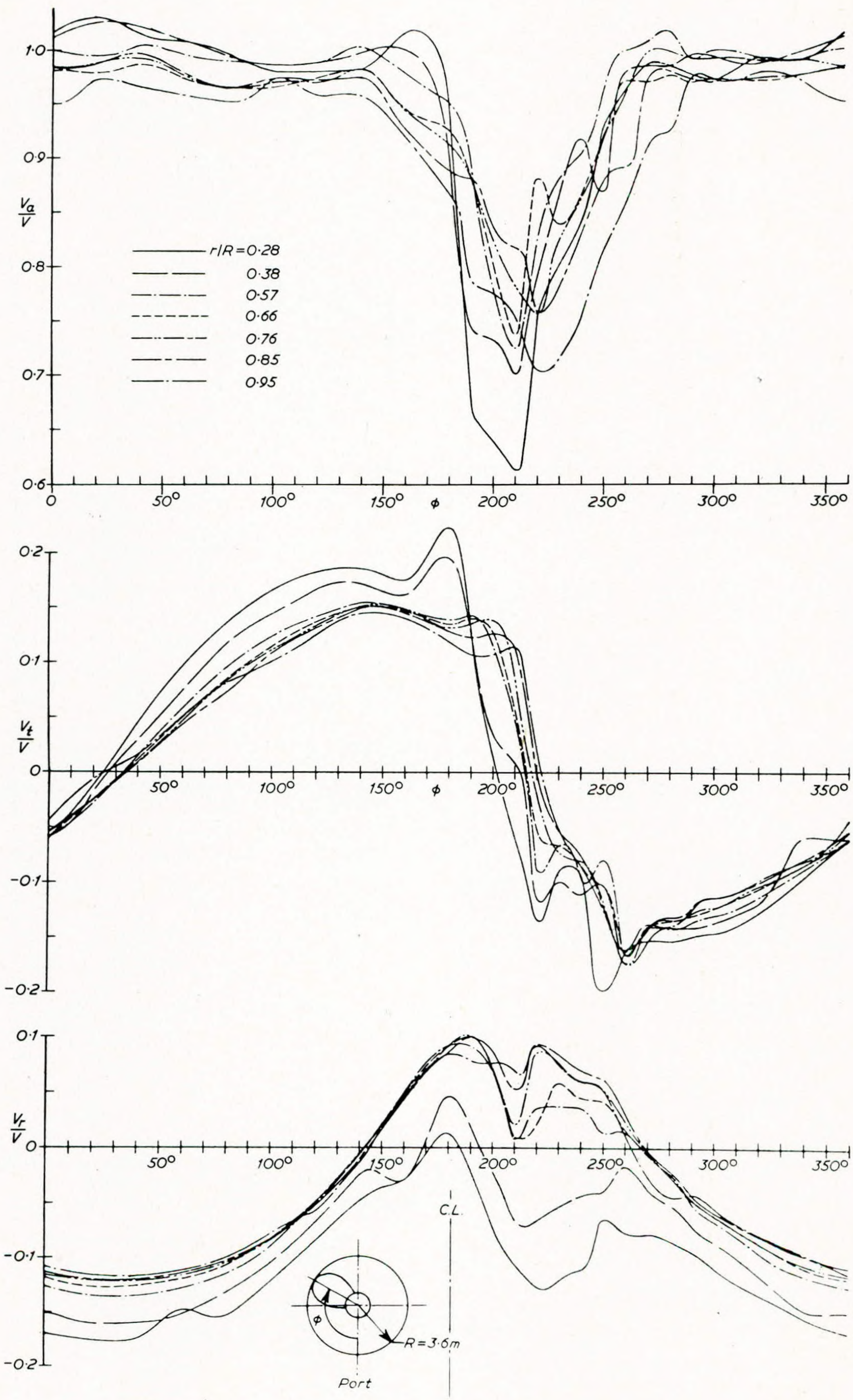


Fig. 8—*Three-dimensional wake survey of a twin-screw containership*

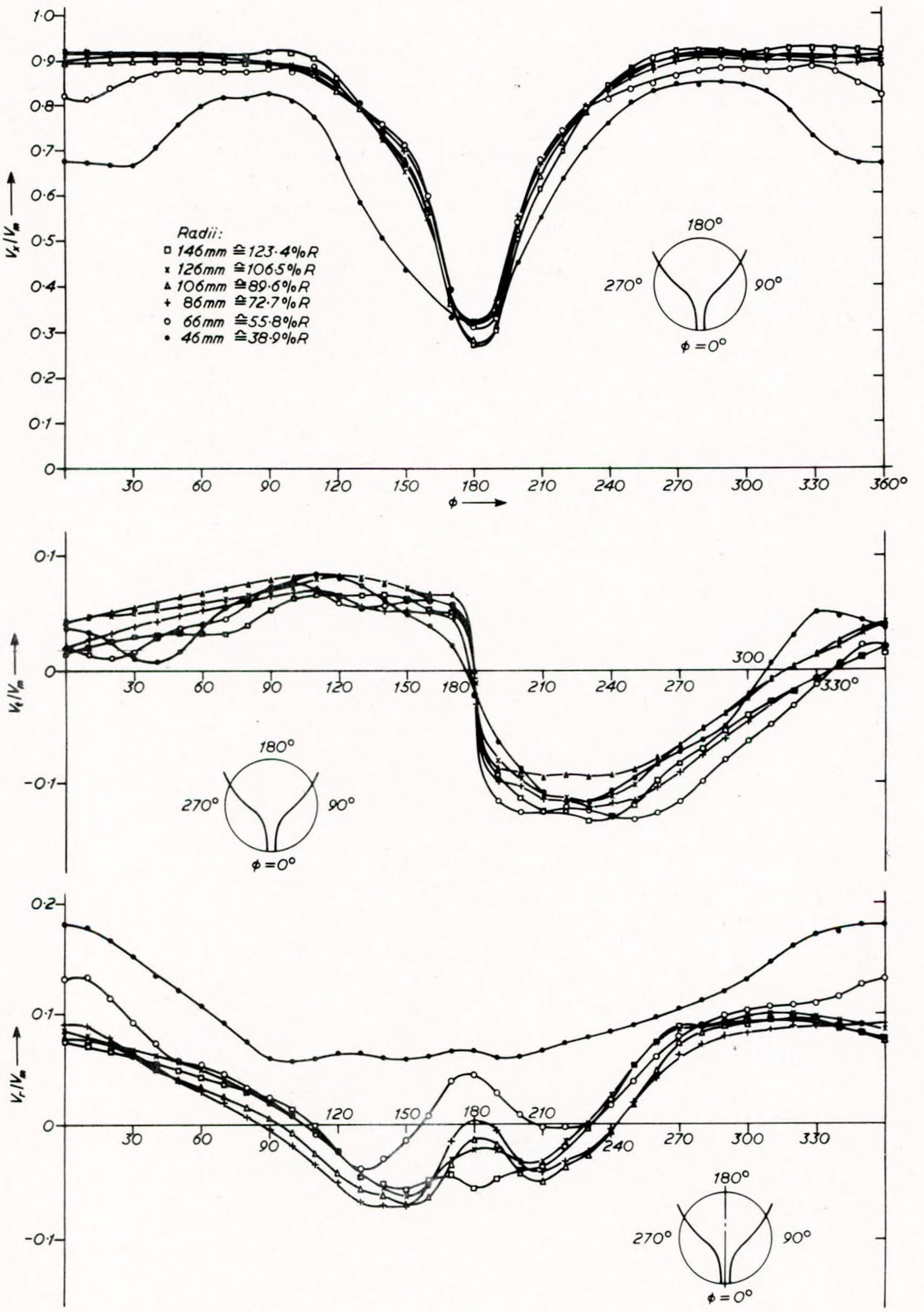


Fig. 9—*Three-dimensional wake survey of Maersk model version III*

TABLE IV—VALUES OF Kx AND Kyz OF A 5-BLADED AND A 6-BLADED PROPELLER IN AXIAL MAERSK WAKE FIELD IV (ACCORDING TO HSVA-METHOD)

Ƶ	5	6	Upper limits
Kx	0·17	0·11	0·25
Kyz	0·41	0·23	0·30(0·35)

Cavitation Tests

The original propeller designs from two competitive manufacturers were gradually improved during extensive cavitation tests carried out by HSVA.

For these tests the axial wake field IV was produced in a cavitation tunnel using a dummy afterbody and mesh wire. The alteration of the angle of attack caused by the tangential wake component was taken into account by reducing the number of revolutions. Because of the resemblance between the wake fields of full load and ballast condition, the same velocity distribution for both draughts could rightly be used.

TABLE V—MAIN DATA OF FINAL MAERSK PROPELLERS (TWO MANUFACTURERS).

	A	Б
Diameter D(M)	7·1	7·1
Pitch (mean) P (M)	8·97	8·85
Pitch ratio P/D (1)	1·263	1·246
Disc area ratio A_E/A_O (1)	0·920	0·947
Blade number Ƶ	6	6
Propeller speed n (rev/min)	100	100

The tests were primarily aimed at recognizing and eliminating major erosion problems. It was the aim to establish steady sheet cavitation on the back of the blades when passing through the wake peak. Bubble and cloud cavitation should be avoided, as well as any kind of face cavitation. Cavitation observations were pursued for the following conditions:

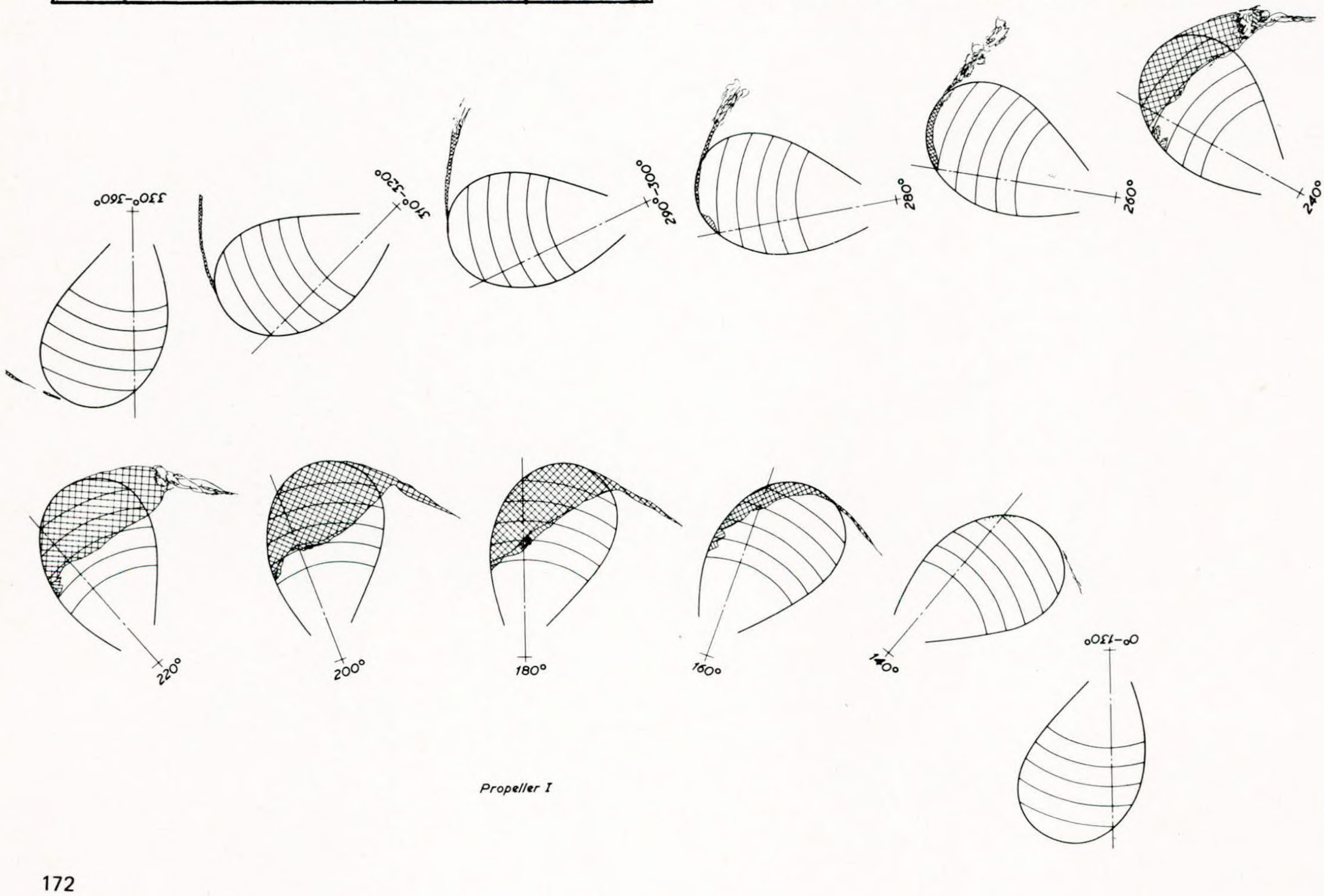
T = 9·6 m, trim = 0 m:	39 600 hp (29 125kW) and
	35 900 hp (26 400 kW)
T = 6·5 m, trim = 2·5 M:	35 900 hp (26 400 kW)

Fig. 10 shows some cavitation patterns on the final model propellers from both manufacturers.

HSVA reported on the cavitation behaviour of both propellers and said that the cloud cavitation near the trailing edge, specially around 240° found for all tested conditions, might represent a slight danger of erosion. Signs of bubble cavitation near the leading edge of the blades when entering the wake peak were considered removable by small form alterations. The same held for the observed face cavitation. On the whole the observed cavitation phenomena would have no influence on number of revolutions and thrust.

As a result of these observations on models the final full scale propellers were once more slightly altered. Their main data is given in Table V.

Full scale observations of propeller cavitation are not



available. A diver's report states that the propeller blades do not show any sign of erosion after one year in operation.

Response

The effort was concentrated on the response side of the vibrational problem and not with an attempt to optimize steel weight, steel structure, and natural frequencies. In most cases, the steel structure was dependent on local boundary conditions and classification rules. The exception was for the superstructure and the afterbody near the propeller where the stiffeners have on the average been thickened. Essentially, the results of the vibrational calculations show the actual conditions since the opportunity of optimizing is scarcely possible in yard business.

Complicated structures were as far as possible reduced to simple substitute systems. It was afterwards proved by measurements that the natural frequencies have been predicted with sufficient accuracy. It should be mentioned that contrary to the view of both the owner and Lloyd's Register, certain sub-structures (see next paragraphs) were designed to have a lower first natural frequency than the blade frequency. In fact, some structures due to their size and scantlings exhibited low first natural frequencies in the domain of the blade frequency. However, strengthening the structures in order to remove this undesirable effect would be rather costly, the main drawback being increased steel weight affecting the ship's economic performance.

Shafting

The computed natural frequencies of forward whirl and counterwhirl of the shafting up to 25 (15) Hz are shown in Fig. 11(a). The bearing distances were arranged so that the first order blade frequency was about 1·4 times higher than the first natural whirl frequency. Measurements (Fig. 11a) coincided well with the calculations.

Fig. 11(b) shows the calculated and measured deflexions of the tail end of the shafting at $n = 100$ rev/min. The calculation was based on the predicted hydrodynamic propeller forces taking into account bearing elasticity but not damping. As a consequence of the fact that the required frequency was far from the first natural frequency, a rather good agreement has been found.

Natural frequencies of transverse and longitudinal vibrations were also calculated. To comment only on the more important longitudinal vibration, a first natural frequency of 11Hz was calculated. The measurement yielded 10·4 Hz. No dangerous effect of that near-resonance condition has been reported to date.

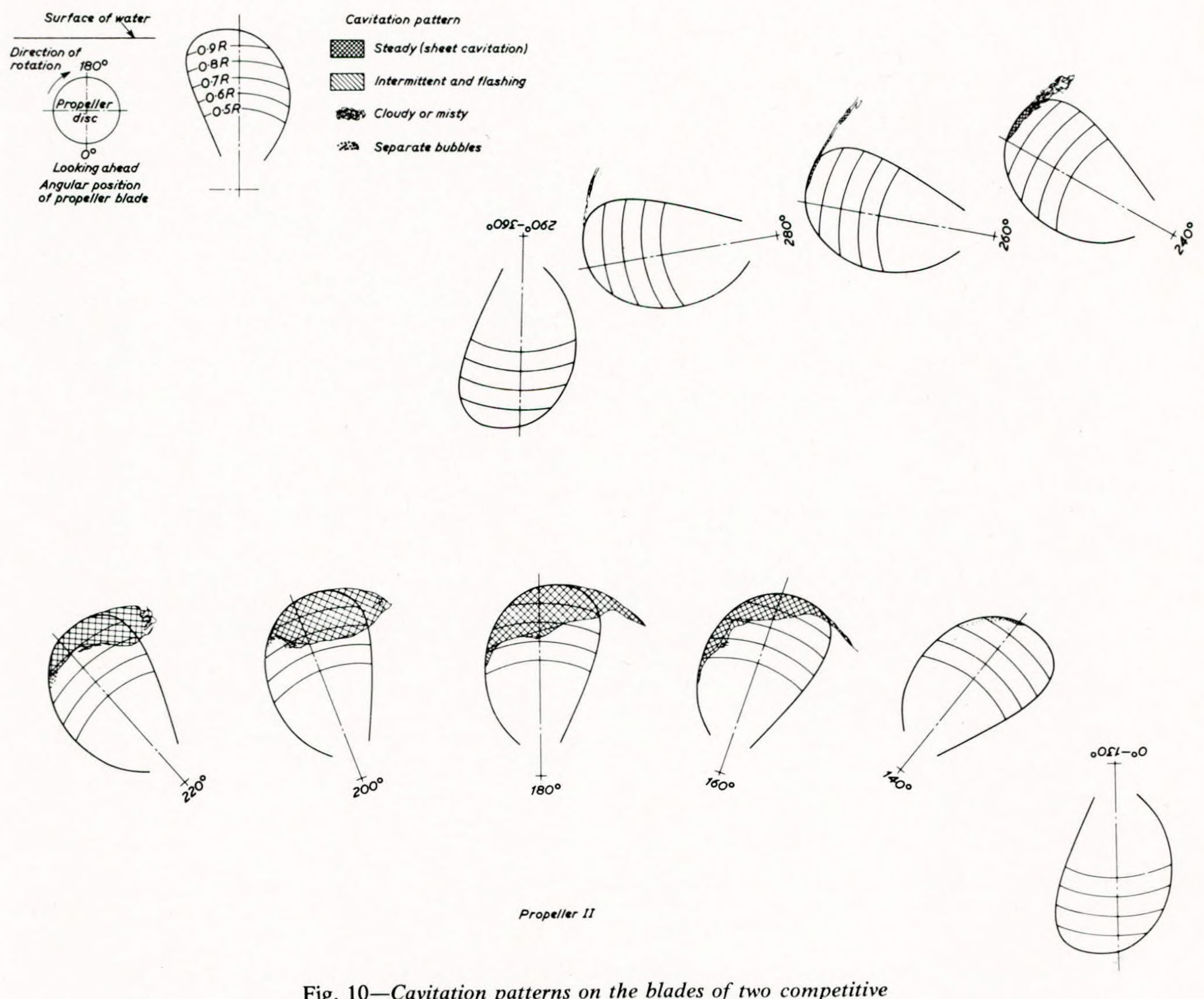


Fig. 10—*Cavitation patterns on the blades of two competitive propeller models observed in the axial Maersk model wake field IV*

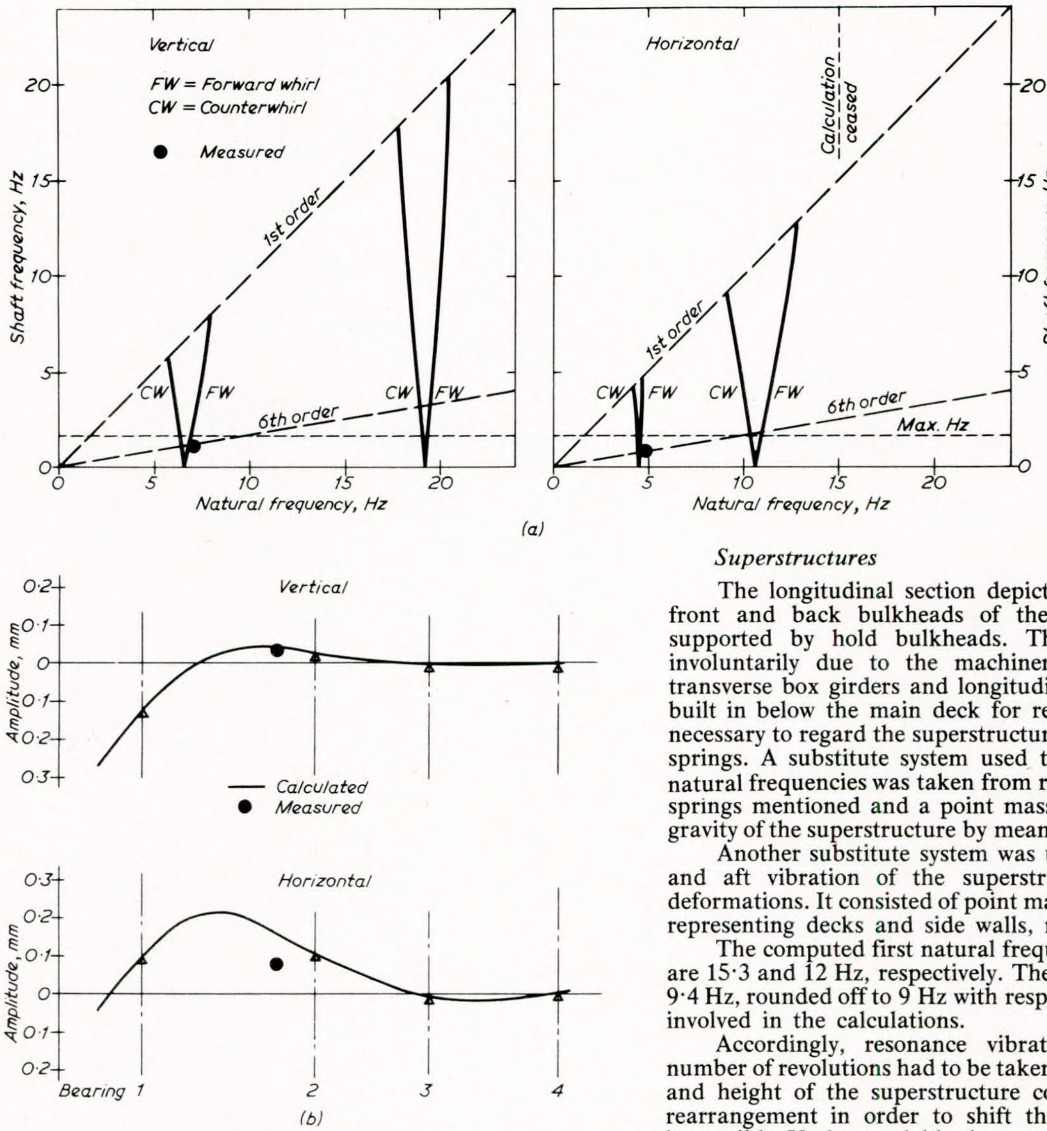


Fig. 11—*Natural whirl frequencies (a) and (b) deflexions of shafting (n = 100 rev/min)*

Stern Bulb Area

An initial rough calculation of the natural frequencies of the extremely slim part of the afterbody in front of the propeller yielded values so near to the main exciting frequency that a more precise examination was necessary. A substitute system was developed consisting of an elastically supported beam with variable moment of inertia in the shape of the stern bulb. The spring elements presenting bending and shear flexibility were also made variable according to their different "lengths" between beam and assumed clamping in the fifth deck (see Fig. 12). For this system of 26 m length, a first natural frequency of 7 Hz was calculated which is well below the blade frequency of 10 Hz. Confirmation by measurement is not available.

Vertical Hull Bending Vibration

The natural frequencies of the vertical hull bending vibration were calculated using a substitute system of forty discrete discs. Results for several draughts and essential loading conditions were obtained, and some can be checked up against actual measurements. The first two natural frequencies at trial draught occurred at 7·5 and 13·5 Hz, the respective calculated values being 6·8 and 13·9 Hz.

Based on the agreement obtained it is reasonable to conclude from the calculations that there will be no propeller excited resonance at or near the nominal number of revolutions of one of the first four degrees of vertical hull bending vibration at any of the loading conditions considered.

Superstructures

The longitudinal section depicted in Fig. 12 shows that front and back bulkheads of the superstructure are not supported by hold bulkheads. This has to be accepted involuntarily due to the machinery arrangement. Though transverse box girders and longitudinal wing bulkheads were built in below the main deck for reasons of stiffness, it was necessary to regard the superstructure bulkheads as resting on springs. A substitute system used to evaluate the rotational natural frequencies was taken from ref. (14). It consisted of the springs mentioned and a point mass located in the centre of gravity of the superstructure by means of rigid, massless levers.

Another substitute system was used to calculate the fore and aft vibration of the superstructure caused by shear deformations. It consisted of point masses connected by springs representing decks and side walls, respectively.

The computed first natural frequencies of the two systems are 15·3 and 12 Hz, respectively. These were combined to give 9·4 Hz, rounded off to 9 Hz with respect to the rough estimates involved in the calculations.

Accordingly, resonance vibrations near the nominal number of revolutions had to be taken into account. As the size and height of the superstructure could not be altered, any rearrangement in order to shift the natural frequency was impossible. Understandably close attention was paid to the trial measurements.

A slight but noticeable resonance caused by the blade frequency was measured at $n = 85$ rev/min. In fact, this was the only vibration of the superstructure worth mentioning throughout the whole range of shaft revolutions, and the amplitudes decreased markedly towards the nominal revolutions.

The satisfactory vibrational behaviour of the superstructure may to a great extent be due to locating *the housing remote* of the propeller, but it could probably not have been obtained if the exciting forces had been large.

Afterbody Details

Between stern-tube bulkhead and rudder horn plate floors, spaces 600 mm apart and 15 mm thick, have been provided (Fig. 12). The thickness of hull plating in this area ranged from 33 to 21 mm, and no special vibrational investigations into this rather strong construction or parts of it have been carried out.

Several other elements of the afterbody, e.g. decks, bulkheads, and foundations of machinery plants, have, however, been thoroughly examined. Some of the calculated first order natural frequencies (main motion direction) are given in Fig. 12 and it should be pointed out that the natural frequencies of most of the investigated structures are fortunately much higher than the main exciting frequency of 10 Hz. The rudder horn being tuned to 6 Hz is the exception.

Two additional longitudinal bulkheads, marked "add" in Fig. 12 were demanded by the classification society. In order to increase the bending and shear stiffnesses of the afterbody they

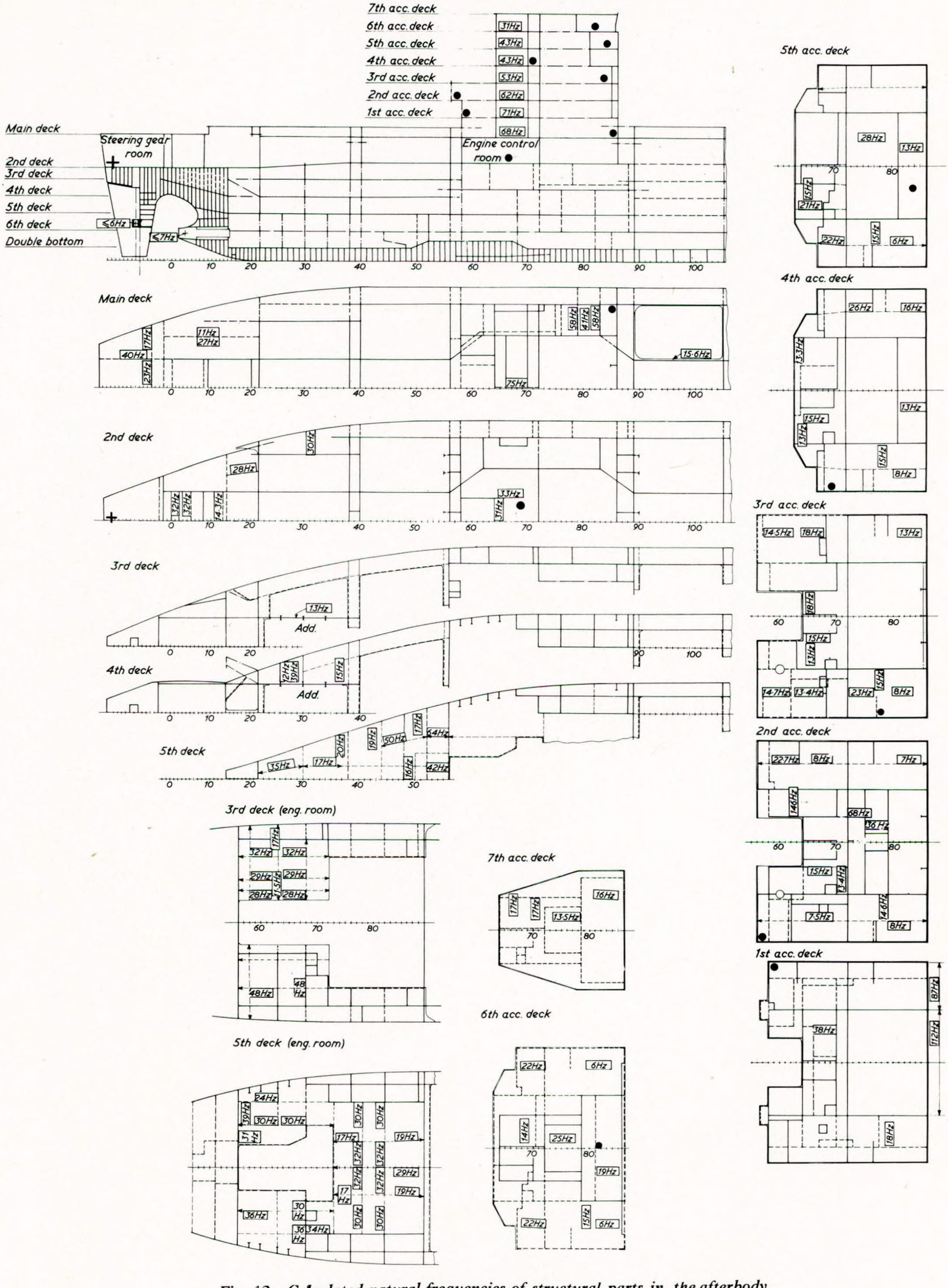
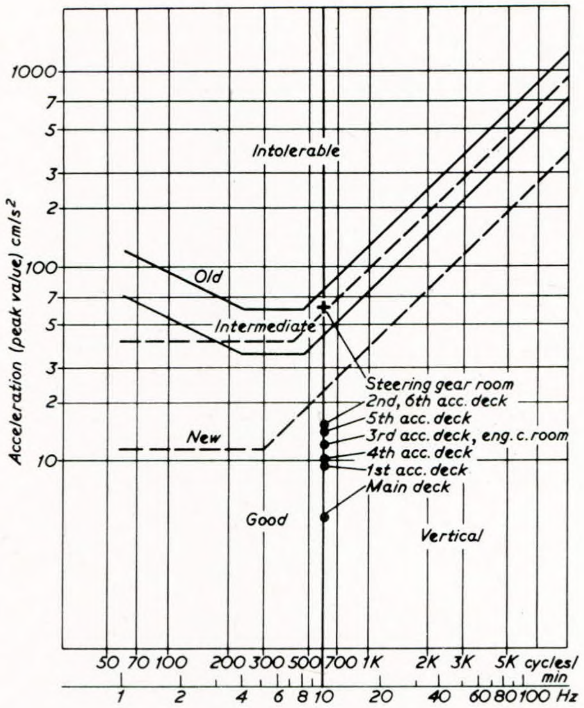
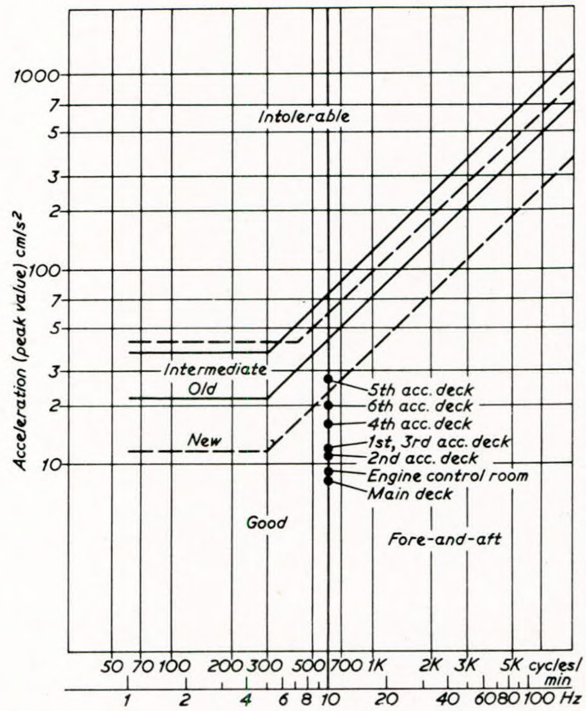


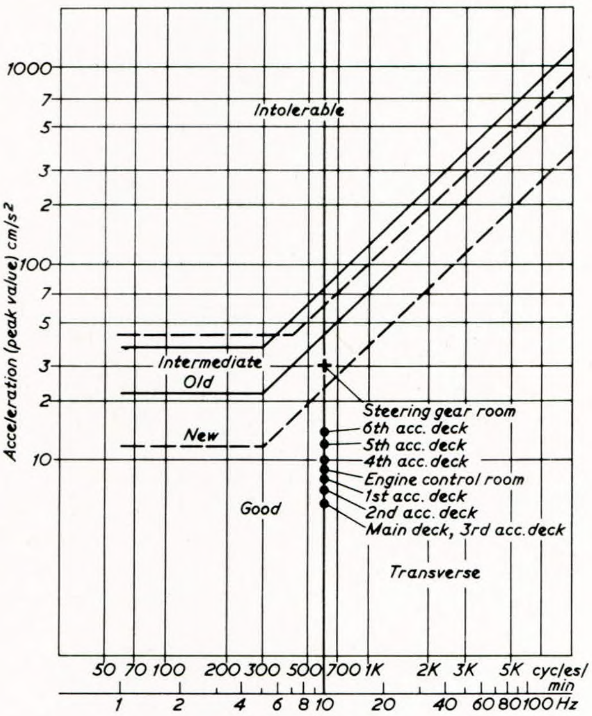
Fig. 12—*Calculated natural frequencies of structural parts in the afterbody*



(a)



(b)



(c)

Fig. 13—*Comparison of measured vibrations in the afterbody with the fatigue decreased proficiency boundaries (ISO/DIS 2631 = old; new specification presently being prepared = new)*

were originally planned to reach up to the end bulkhead of the machinery space. It could, however, be proved by calculation that their contribution to stiffnesses decreased rapidly beyond the length now scheduled.

To avoid resonance vibrations, single structural parts like stiffeners and girders were regarded as vibratory systems from the outset. Sketches showing the natural frequencies of stiffeners as a function of length, strength, and loading were prepared to help the constructor select scantlings at an early stage in the design. Owing to this procedure the weight of some structures has increased, and, in particular, the superstructure.

Fig. 13 is intended to show the level of vibration on board the ships. Measurement results at trial draught and nominal power output representing the accelerations in vertical, fore-and-aft, and transverse directions are given at control points on each deck. From several control points the most annoying have been selected for presentation; their locations being shown in Fig. 12. Heavy lines in Fig. 13 indicate the limits of human comfort according to ISO/DIS 2631 and the new specification presently being prepared (15). The majority of the control points is found in the region defined as "good". Even the measurements in the steering gear room yielded "tolerable" results.

CONCLUSIONS AND FURTHER DEVELOPMENT

Succeeding Tank Tests

The tank tests for the Maersk ships were followed by others, with the aim of optimization and of investigating their effect on new ship designs.

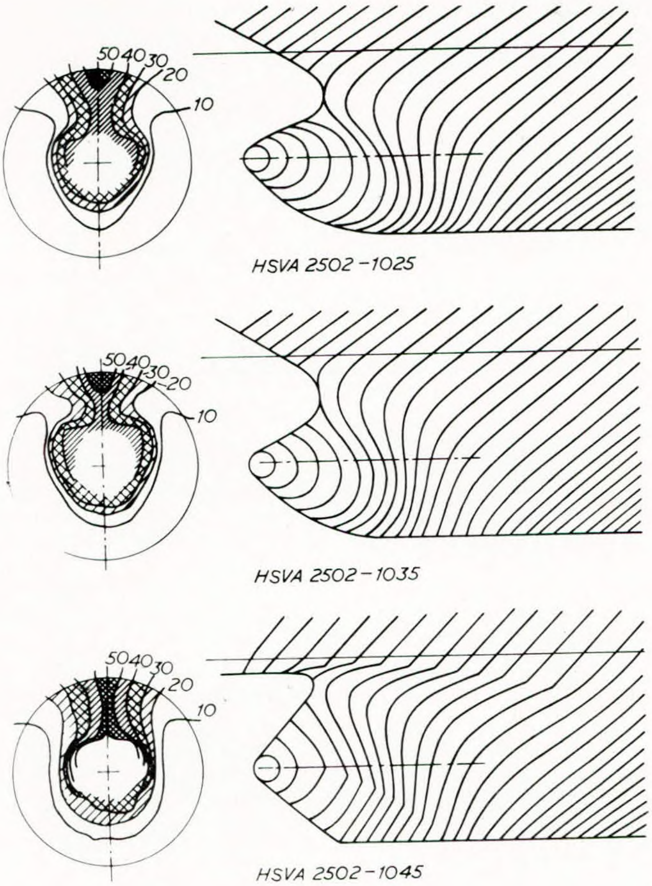
A model (HSVA 2502-1014) from an earlier test series was used which had the same size and same C_B as the Maersk ships, but with the centre of buoyancy further forward. Thus, it had finer afterbody lines and the propeller was arranged at the end of a long sterntube. This model was taken as a basis for the following alternatives (Fig. 14):

Alternative HSVA 2502-1025 : The propeller was moved three meters aft and the sterntube was converted to a barrelled, rotationally symmetric stern bulb.

Alternative HSVA 2502-1035 ; To avoid the incipient vortex above the centre of the prepeller disc area in this alternative, the bulb was thickened above the shaft.

Alternative HSVA 2502-1045 : The upper part of bulb was further thickened and a knuckle line was produced running up to the top edge of the propeller disc to increase the flow towards the propeller's twelve o'clock position.

Apart from the gradual improvement of the axial wake field there was a gradual reduction of 5 per cent in the required propulsion power probably due to a reduction in the ship's resistance. Unfortunately, resistance measurements were not



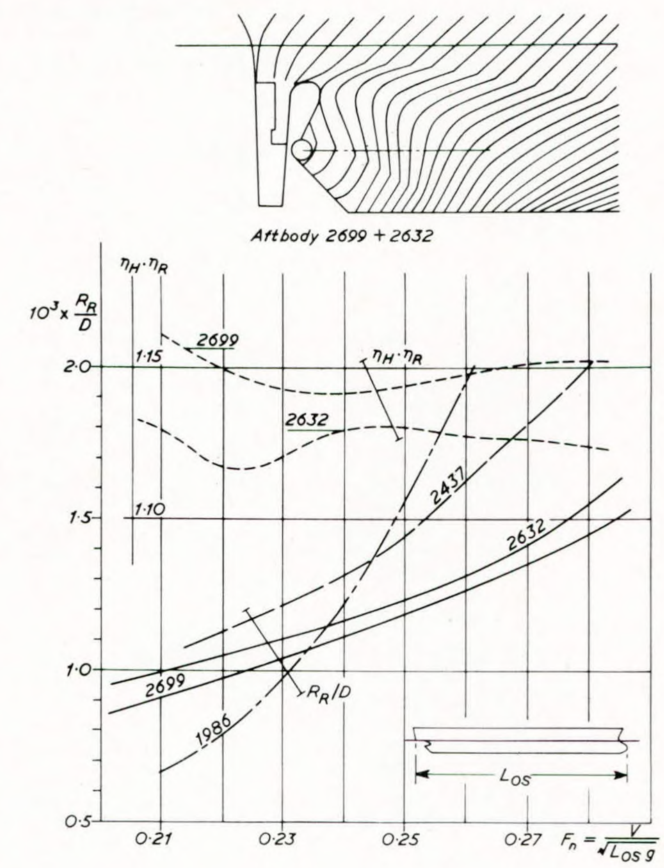


Fig. 14—*Stepwise development of the B+V sternbulb—Version 2502-1025 has completely rounded sections whereas Version 2502-1035 is slightly thickened above the tailshaft—The tailshaft—The final Version 2502-1045 has plain areas in the lower part, rounded surfaces at its upper part, and is faired to the hull with a nearly horizontal tangent—Thus, a kind of channel is formed to increase the fluid velocity to the propeller's twelve o'clock position.*

Fig. 15—*Comparison of residual resistance* R_R/D *and propulsive efficiency* $Z_H \times Z_R$ *for several models—The numbers refer to models with following characteristics:*

1986 : Los/B = 6·62; B/T = 3·13; CB = 0·571
2437 : Los/B = 6·12; B/T = 2·76; CB = 0·473
2632 : Los/B = 5·37; B/T = 2·61; CB = 0·496
2699 : Los/B = 6·85; B/T = 3·20; CB = 0·480
Models 2632 and 2699 have a B+V sternbulb.

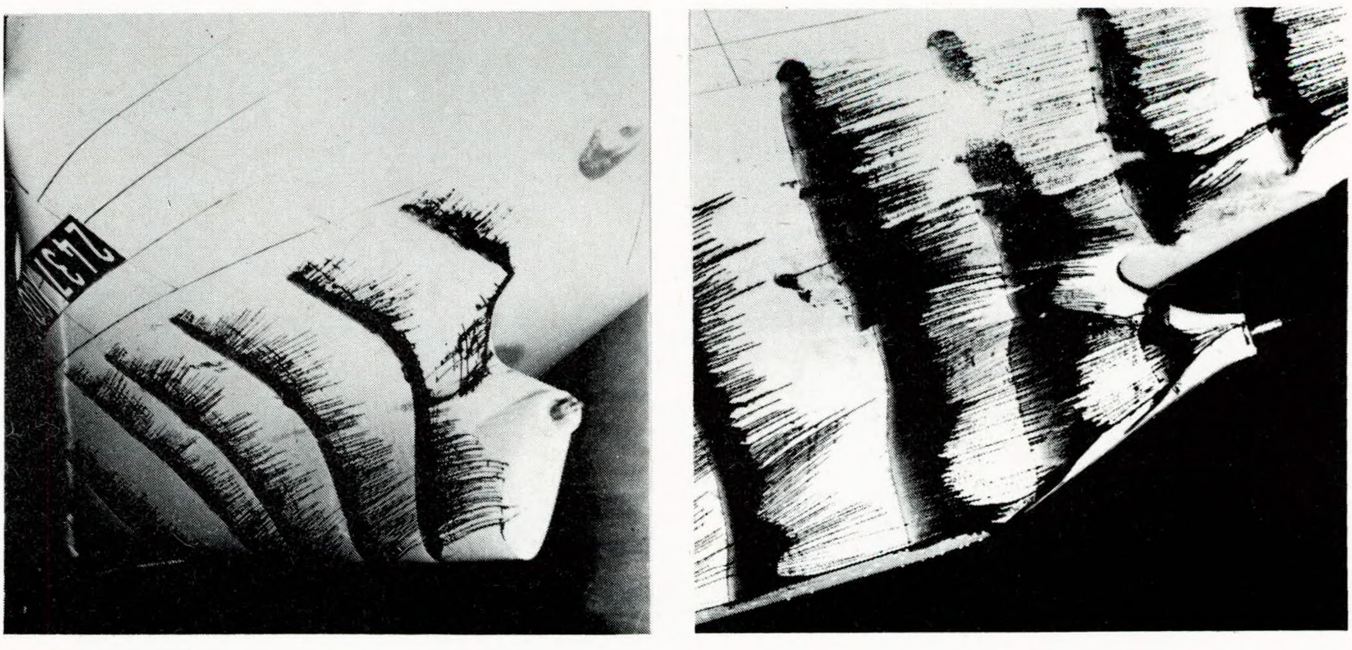


Fig. 16 (a, b)—*Streamlines on model 2437 (a), and model 2632 (b)—The intensive flow above the sterntube of model 2632 is typical for the B+V bulb arrangement.*

taken. The streamline studies with paint show a very intensive flow in way of the upper propeller disc area, and the calculation of the shaft forces for the same propeller design as of the Maersk ships gave the following values:

$Ky = 0{\cdot}04$

$Ky_z = 0{\cdot}12$

The corresponding values of the Maersk ships were:

$Ky = 0{\cdot}11$

$Ky_z = 0{\cdot}23$

After afterbody-form (Fig. 15) obtained in this way was adopted practically unchanged for two containership proposals which were tested by the HSVA with model Nos. 2632 and 2699. The first with its extremely small L/B-ratio (= 5·0) and high KM_0/B (= 0·475) and low block coefficient (= 0·50) was intended for smaller types having no restriction of beam and draught, while the latter was for a typical "Panmax"-containership. Both models were tested with a short faired-in bow bulb.

Results of tank tests for model Nos. 2632 and 2699 showed that for both hull forms the resistance curves rise exceptionally gradually and the coefficients for propulsion efficiency (ZH ZR) are kept consistently high over the whole tested speed range. The streamline tests indicated an extremely intensive flow in way of the upper portion of the propeller disc area. Wake measurements have not been taken up to now, but it can be assumed that the wake patterns are similar to those of HSVA model 2502-1045 (Fig. 16).

These results show that for future containerships a service speed of 26 kn at fully loaded draught with 50 000 shp (36 770 kW) can be expected, even for smaller (200 m) ships, providing the single-screw arrangement is adopted.

Such ships could then operate in competition with the existing twin-screw ships their lower fuel consumption per unit (Fig. 17) being a result of the single screw arrangement. Their medium size can be of advantage for sailing frequency and shore organization; in other words: these "compact ships" could help to approach the desirable "pipe line effect" in container rotation.

Problems Which Need Further Attention

Apart from vibration problems explained in this paper which are subject to worldwide investigation, there are still many unsolved problems which require further attention, such as:

a) Low resistance ships suffer a disproportional loss of speed in heavy weather conditions —particularly in transverse and oblique seas and winds ([11]).
What is needed are guidelines so that designers can take care of this problem in the early stages of hullform design. Similar information would be appreciated by the propeller manufacturer.

b) An acceptable minimum GM_0 is not solely governed by safety requirements against capsizing, but also in making allowance for the operations requirements such as a constant angle of keel in a lateral wind and the angel of heel when the ship is turning.

c) Investigations into various configurations for the hull structure to improve rigidity, reduce vibration levels and reduce the weight of steel.

ACKNOWLEDGEMENT

The assistance of Mr. K.-H, Kluge, Mr. H. Schonfeldt, and Mr. P. Behr (all Blohm + Voss AG) during the preparation of Section 2 of this paper is acknowledged with thanks. Most of the data given herein has been taken from their paper ([16]).

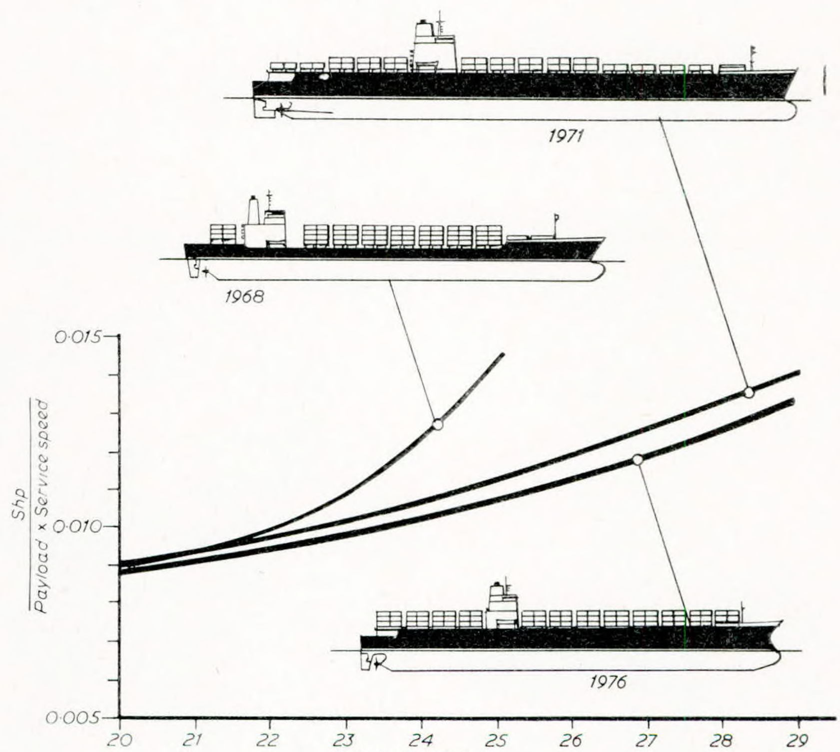


Fig. 17—*Technical development of containership design in the last decade—The fraction P /Payload x V is based on the following assumptions:*
Weight of 20 ft container: 10 t (homog. stowage)
Maximum draught: 11·0 m
Range: 12000 n miles.
Service condition: Tank resistance + 20 per cent
The advanced single-screw ship indicated with "1976" is based on the hull form of model 2699.

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DISCUSSION

CAPTAIN T. DILLING said that this paper was almost as well executed as the ships were. It was amazing to see how clearly cut and well defined one could be when explaining how these vessels were developed. His company, as owners, went through a very worrying time when developing these ships. The six container ships, together with their three sister ships in the Flender yard, were the biggest single investment they had ever made and the demand was that they had to be successful. They knew when they contracted for these vessels that, with about 40,000 hp on a single propeller, they were moving in an area where there were unknown factors and where other people had run into problems. Therefore, they had begun by concentrating on this particular problem area and it was satisfactory for them to meet people with such dedication as Mr. Langenberg who worked on this job with an open mind with a view to not only fulfilling the contract, but also with a desire to obtain the best possible result.

It was fair to say that the end result was also due to real co-ordination and co-operation with both the Classification Society and the propeller manufacturers.

Today these ships were in service and had in their performance proved all the owners expectations and, with regard to vibration, they had even exceeded those expectations. He did not think it wrong to say that there were hardly any ships afloat with as little vibration as in those ships. In seaworthiness they had proved to be satisfactory and, in general, under all operational conditions. It would be wrong for an owner not to have a few things to point a finger at, otherwise the naval architects might think they had reached the optimal solution; his colleague Mr. Kappel would dig into some of these technical questions. He would, however, like to point out that the one area in which there had been some disappointment was that of the speed in loaded condition. Here they did not attain the speed that had been calculated, but he was sure there would be some explanation for that later on.

He would like to mention that Mr. Langenberg in his paper had strongly advocated light steel weight, and no owner could be opposed to that idea, but at the same time there were a number of owners and also Classification Societies who had run into very serious problems due to light steel weight, so one should be careful to reach a happy medium.

He was sure it would interest the audience to know that the company had carefully watched the propeller blades for cavitation during the operational period and, so far, had not been able to detect any cavitation or erosion on the blades, by divers' inspection, after one year in service, or more than one year in service with at least the first of the ships.

MR. B. RAPO, M.R.I.N.A., said that successful designs were rarely discussed by professional bodies. The limelight was usually reserved for specific solutions which had been developed in response to a given problem.

It was gratifying to be able today to discuss the success of a particular design and, if the authors deserved congratulations for an excellent paper, then they deserved even greater recognition for an excellent design in the first instance.

Container vessels of about 200m in length, commonly referred to as second generation containerships, were normally diesel or turbine-powered, and developed about 30,000 h.p. (22,371 kW) at 110 to 140 rev/min for a service speed of about 23 kn. There were quite a few single-screw vessels of this size, in the 36-38,000 h.p. (26,845 - 28,336 kW) range, built in Japan and in Europe along conventional lines. The single-screw container vessel "Japan Ambrose" (215 x 32.2 x 19 x 11.03; container capacity about 1600 20 ft units; engine room three-quarters aft) was a notable exception. She was delivered by IHI in 1972 with a turbine propulsion plant of 50,000 s.h.p., sufficient to give the vessel a speed of 25.1 kn. Hence, the statement made in one of the concluding paragraphs that "a service speed of 26 kn at fully loaded draught with 50,000 s.h.p. (36,770 kW) can be expected for ships of 200m" was fully agreed with an indeed, had already been achieved. The vessel under discussion, "Adrian Maersk" was not a standard "off the peg" design; she had very fine form and a block coefficient of 0.54 which was below the usual range of values of 0.58 to 0.68. The vessel was fitted with a propulsion plant capable of developing 36,000 s.h.p. (26,845 kW) at 100 rev/min.

At the time of initial design appraisal, Lloyd's Register of Shipping had examined the wake pattern and carried out a harmonic analysis of the circumferential axial wake distribution for this vessel, the results of which were shown in Fig. 18. These results reflected exceptionally good afterbody characteristics. The maximum wake peak was of the

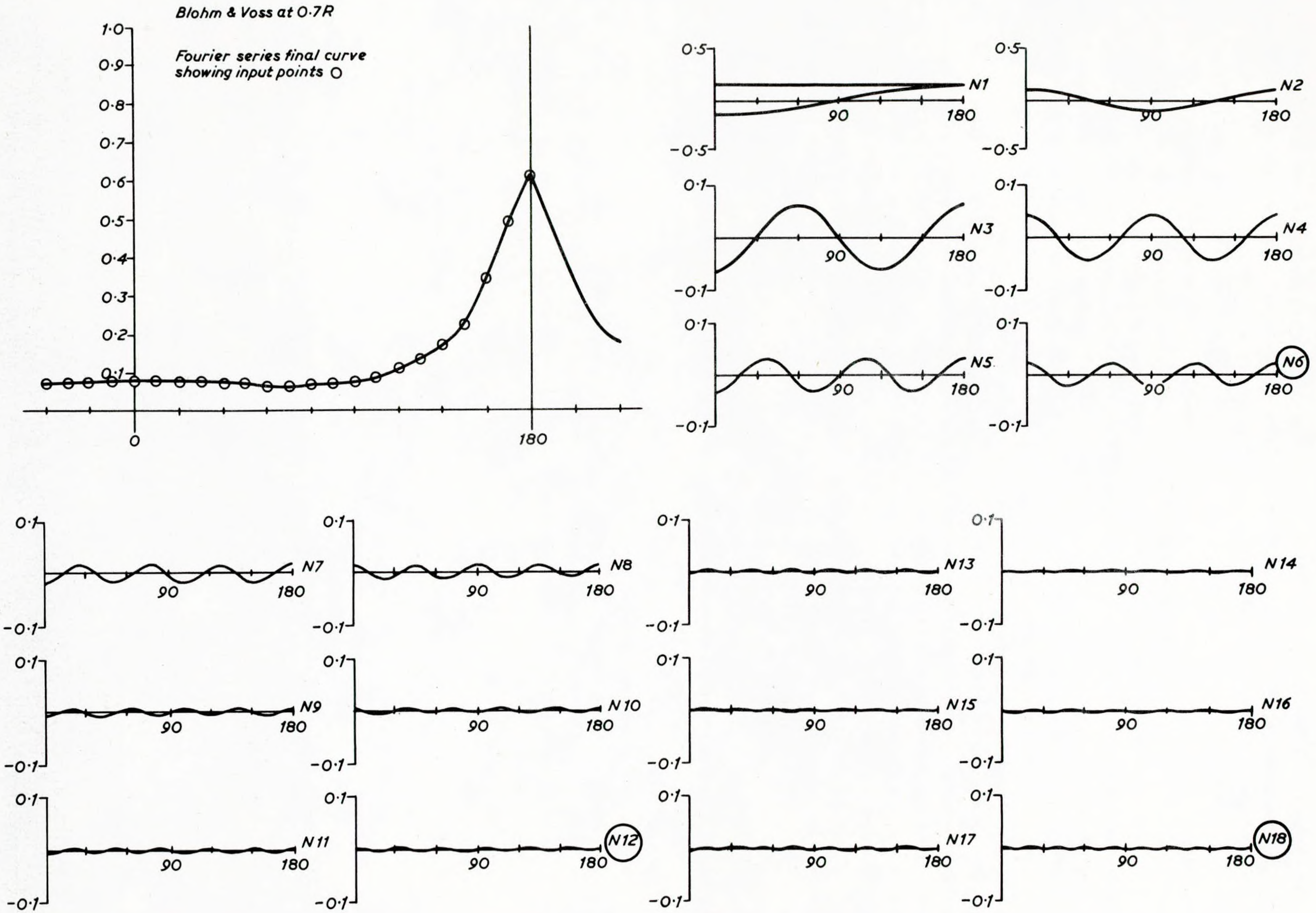

Blohm & Voss at 0·7R
Fourier series final curve
showing input points O
1·0
0·9
0·8
0·7
0·6
0·5
0·4
0·3
0·2
0·1
0
180
N1
N2
N3
N4
N5
N6
N7
N8
N9
N10
N11
N12
N13
N14
N15
N16
N17
N18
90
180
0·5
-0·5
0·1
-0·1

Fig.18

order of 60 per cent in the top blade position, and the gradient was moderate. Whilst propeller blade harmonic amplitude showed relatively small fluctuations, all higher order harmonics were virtually non-existent, and this was remarkable.

Wake field pattern, however good, could not, on its own, guarantee vibration free performance. For non-cavitating conditions it determined the intensity of pressure oscillations. Amplification, due to cavitation-induced high frequency components in the pressure field, depended upon the character and extent of the cavitation. These two aspects and the interaction between the hull and the propeller determined the general level of excitation forces.

The authors had rightly pointed out that it was impossible to give non-controversial interpretation of the wake measurements results. Also, it was virtually impossible to produce a universally valid recipe for a vibration free design. Part of the solution might be found at a stage when the vessel was still on the drawing board. It was generally considered that good hydrodynamic characteristics of the ship's afterbody required the angle of run of critical waterlines to be kept in the range of 20 - 22 degrees. Also, there were indications that concave shaped waterlines tended to slow down the axial flow. Propeller clearances should be sufficient, but indiscriminate increases might spoil the angle of run of waterlines and a proper balance must be found in each case.

On the response side, the authors stated in the paper that satisfactory vibrational behaviour of the superstructure might be, to a great extent, due to locating the housing some distance from the propeller. This was readily agreed with. It was remarkable how few complaints, concerning vibrations in way of accommodations, were heard from ships where the deckhouses were separated from the aft end by one or two cargo holds.

Further, the incorporation of longitudinal bulkheads in way of deck girders at the extremities of the vessel had, no doubt, made another beneficial contribution to the overall response aspect.

MR. D. GARRETT, B.Sc., C.Eng., M.R.I.N.A. regretted that he was not conversant with some of the symbols used in the paper and would therefore like to ask the authors if they could include a list of symbols with their definitions in the final version of their paper.

Having been involved for some years in the hydrodynamic aspects of the design of various ship types, he would very much appreciate the authors' views on a few specific questions which their paper had raised in his mind.

Firstly, were vibrations measured on trial at frequencies other than blade rate ? If so, it would be interesting to see them added to Fig. 13 because, in his experience, problems could be encountered in single-screw ships at higher frequencies, even if the blade rate vibrations were acceptable.

Secondly, referring to the model cavitation sketches shown in Fig. 10; the two propellers clearly had different cavitation performances and it would be of great interest to know why. What were the design differences between these propellers which might cause this effect ? For instance, was the lift generated more by blade camber than by angle of attack, and what was the radial lift distribution ? Also was there any difference in the open water performance of these two propellers ?

Thirdly, what was the Hamburg Tank's calculation method for hydrodynamic propeller forces referred to in the paper ? What assumptions were involved ? What wake pattern was applied ?

Finally he wondered if the authors could comment on the reasons for the choice of 98.5 rev/min for this ship - was it largely an attempt to reduce vibrations, or was it to increase the propulsive efficiency ?

MR. G.C. VOLCY, M.Sc., F.I.Mar.E., commented that the authors had shown a judicious way of solving vibration problems by acting on the excitation forces. However, the question might arise, at the building stage of the ship, of how to cope with vibration problems when rather important hydrodynamic excitations were present.

Mr. Volcy thought he could provide an answer by presenting an example which proved that by the adoption of an unconventional approach, consisting of the research and detuning of forced vibration resonators*, it was also possible to combat vibrations by convenient action on response forces.

The two biggest Ro-Ro ships in the world each of 23,481 dwt and equipped with three medium-speed diesel engines, and driving through gearing the biggest c.p. propeller in the world, of 46,000 bhp (the weight of which, with its four blades together with tailshaft, was 170 tonnes). The inboard profile of the hull and layout of the propulsion plant of these ships were shown in Fig. 19.

These ships, named "Australian Emblem" and "James Cook", had been delivered in 1975 by Kawasaki Kobe

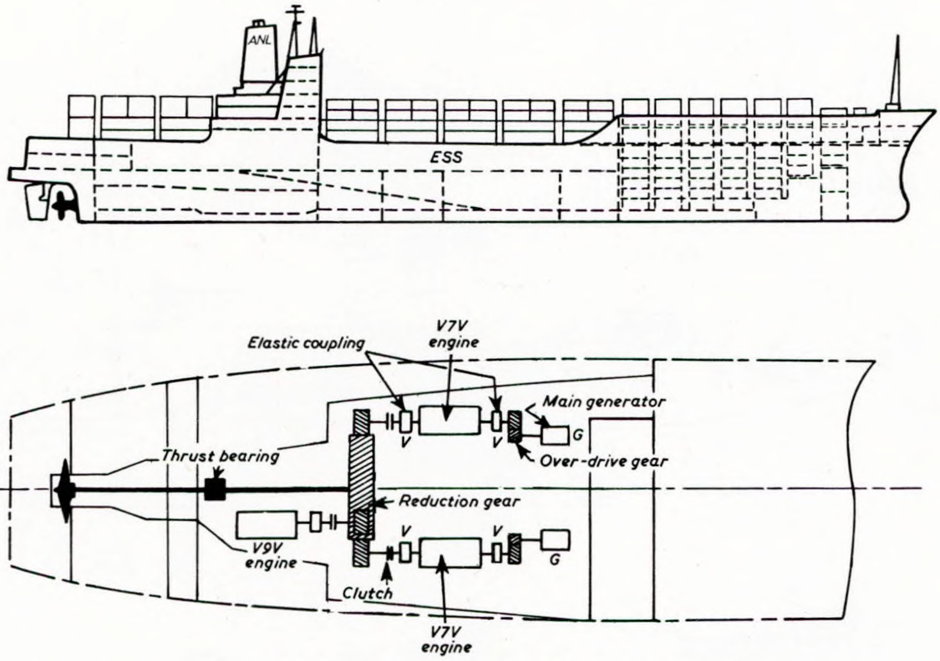


Fig.19 — Inboard profile of the hull and layout of propulsive plant

Shipyard to Australian National Lines.

The operational constraints imposed the presence of complicated internal hull steel-work, especially in the aft part of the ships and in the engine room. Also, however, they required the wide flat form of aft underwater design (ending with a transom stern) and unfortunately this latter part had to be subjected to severe hydrodynamic excitations due to the high output of the c.p. propeller.

They had been determined experimentally by NSMB at Wageningen, showing values largely exceeding those normally admitted, some of which were precisely explained as follows:

- the first harmonic component of thrust variations was 13 per cent of average thrust;
- for torque variations the first harmonic was found to be equal to 10 per cent of average torque;
- the steady thrust excentricities amounted to 17 per cent of propeller radius and their fluctuations of first harmonics have led to a horizontal bending moment of $\pm$ 4,5 tm the vertical one being $\pm$ 17,8 tm;
- as to hull surface efforts and due to the presence of cavitation phenomena, first harmonic efforts of $\pm$ 30 tonnes had been measured vertical.

All these values largely exceeded those normally admitted from the point of view of vibration phenomena.

Faced by these values, the task of Bureau Veritas who had been requested to proceed with calculations and to give advice as to how to keep a low vibratory level was rather arduous.

This had been done by applying the Bureau's philosophy of simultaneous treatment of static and vibratory phenomena by researching the location of forced vibration resonators and detuning them. Thus, a three dimensional F.E.M. model of the aft part of the ships had been built up, the forward part being represented by equivalent masses and beams.

This procedure had then been followed by static calculations - to determine steel work deformations and calculate rational alignment - as well as response in free and forced vibrations. The model in question was constituted of 3,200 elements having 3,000 degrees of freedom which, for vibratory

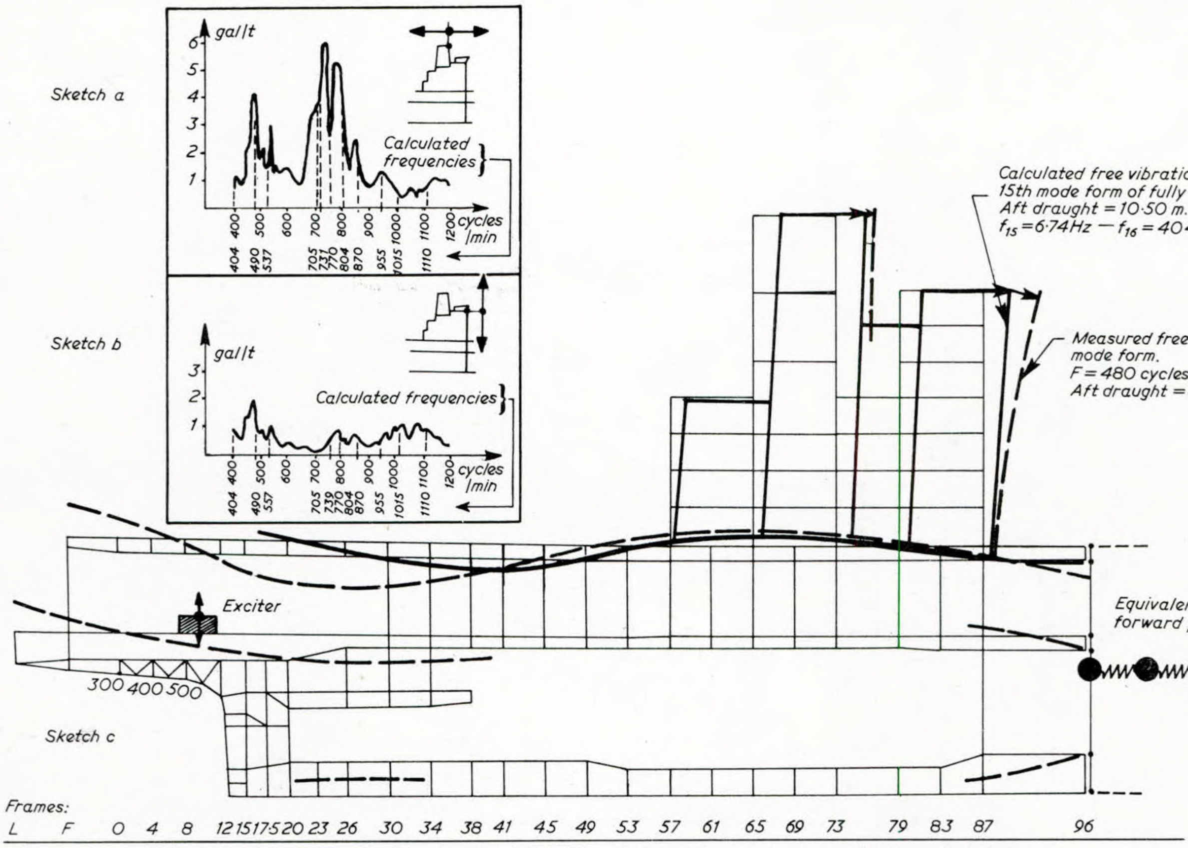


Fig. 20— Elasto-dynamic model of aft part and correlation of exciter tests with free vibration calculations

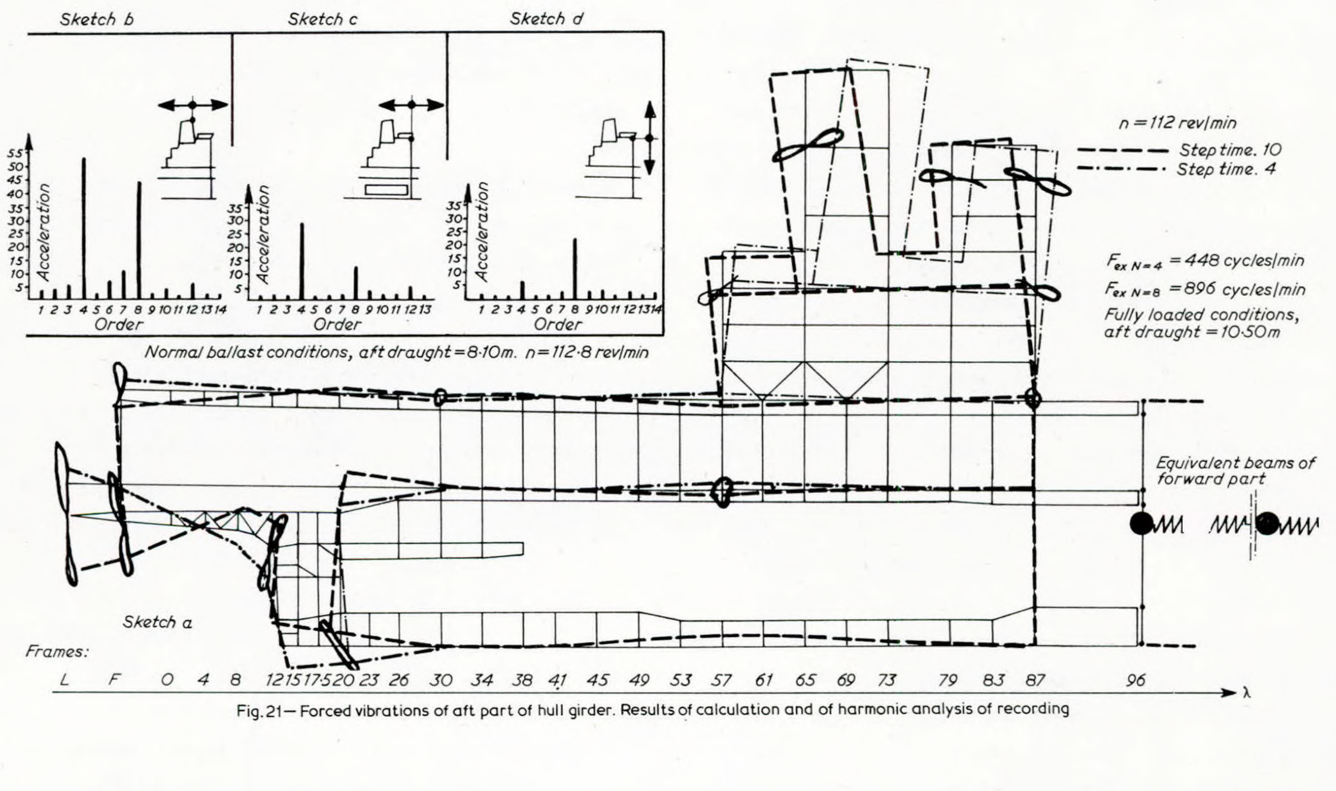


Fig. 21— Forced vibrations of aft part of hull girder. Results of calculation and of harmonic analysis of recording

calculations, have been condensed to 311 master degrees of freedom.

Fifty natural frequencies of free vibrations were calculated, covering the zone of possible resonances of propulsive plant (active resonators) and of steelwork (passive resonators) due to the two lowest harmonics of propeller excitation. On the basis of these calculations much advice had been given to the shipyard (all of which was adopted) for detuning forced vibration resonators.

Fig. 20 showed the correlation, between calculations and results, of exciter tests made by the ship shipyard (upper left side).

A rather good degree of correlation could be seen.

In Fig. 21 the results of forced vibration calculations were presented, as well as the spectral analysis of vibration recordings made by the shipyard during trials (see upper left part of the Fig.).

As to results obtained, it could be stated that the ships were practically free from vibration and noise, the measured vibratory level being 15 to 30 per cent lower than the calculation.

The details of this study could be found in a paper** recently presented in Sydney.

In conclusion Mr. Volcy wished to state that this example confirmed the results obtained by the authors of the paper, namely that today there was enough technical knowledge (on both the soft and hard-ware side) to cope effectively with ship vibrations, which could be called the "cancer of mechanics and shipping".

* G. BOURCEAU and G.C. VOLCY "Forced Vibration Resonators and Free Vibrations of the Hull" 1969

** G.C. VOLCY and M. NAKAYAMA "Studies leading to vibration and noise free ships"
Conference and Workshop on vibration and noise control engineering, Sydney (October 1976).

DR. -ING.K.J. MEYNE said that from the propulsion point of view those single-screw containerships designed by Blohm + Voss and Luebecker Flender were to be considered as a special class with regard to speed and power. The six vessels built by Blohm + Voss had been designed for an output of 36,000 h.p. (26,480 kW) and about 98.5 rev/min, whilst the three ships built by Luebecker Flender were designed for 40,000 h.p. (29,828 kW) and about 102 rev/min. For both types of propellers it had been a requirement that the propeller designer should take into consideration the possibility of absorbing the highest output.

In the early stages it had been possible to have serious and intensive discussions with the Blohm + Voss shipyard regarding the wake field and propeller design. The good technical results obtained with those vessels, especially the low vibration level, was based upon these early-stage discussions.

For the design of those propellers, considerable calculations had to be made.

With their computer program group 1, Mr. Meyne's organisation had analysed the wake field for model scale and estimated full scale with regard to harmonic components and local wake fluctuations at different radii.

The next step in the design work had been an investigation in designing the propellers with a view to considering all important parameters, such as diameter, number of blades, area ratio, radial pitch distribution, profile camber and skew of blades. With this computer program group 2, the design of the propellers had been wake adapted, taking into consideration the efficiency point of view.

However, for single-screw vessels of this high output, the propeller efficiency was not the main design point, because of problems of greater importance, such as: cavitation; thrust and torque fluctuations; bearing forces; bending moments; thrust eccentricity; and propeller-induced hull pressure fluctuations.

With the help of their computer program group 3, the forementioned different propeller designs were analysed for comparison purposes. As a basis for these calculations of the different designs they had used the axial wake field in model scale and in estimated full scale, which had been calculated by NSMB, Wageningen, on behalf of the Blohm + Voss shipyard.

Furthermore, tangential wake components from the three-dimensional wake measurements had been taken into consideration.

With regard to cavitation, it was important to prevent the occurrence of pressure (face) side cavitation and to find out a favourable shape and characteristic of suction side (back) cavitation. For this, the computer program had calculated local angles of attack, local lift coefficients and local cavitation numbers for all radii, at every five degrees of blade position.

These results allowed an estimation of local cavitation behaviour of each special design and supported the design decisions.

Computer program 3 had given the information that the chosen six-bladed propeller would not produce problems with regard to thrust and torque fluctuations, bearing forces, and bending moments.

Because of the fact that the vessels were designed for turbine propulsion, possible vibrations could only be excited by propeller induced pressure fluctuations. So the next step was to try to optimize the propellers with regard to avoiding hull pressure fluctuations.

The most decisive influence upon pressure fluctuations had been the extent and thickness of the cavitation sheet and particularly the time-dependent modification of this sheet while passing through the wake field peak. The loading or unloading of the blade tip was to be considered as the main design parameter.

In the case of a loaded blade tip, the cavitation sheet was comparatively thicker. It was, however, more even on a wide range of the propeller field. During the passage of the blades through the wake peak, there would be only a small modification of the sheet. However, as a comparison, propellers with unloaded blade tips had a cavitation sheet of comparatively smaller extent and thickness; but while passing through the wake peak stronger time-dependent modifications of the sheet had to be expected. This was why propellers designed by them for containerships had been provided with loaded blade tips for some years, with best success.

Lastly, it was necessary to mention the importance of special dimensioning of the propellers, with regard to their strength, for these high powered, single-screw container vessels. With the forementioned computer program, calculations of the stress variations, in model and estimated full scale axial wake and tangential wake had been made.

MR. J.J. KAPPEL said that the involvement in the design and building process of these ships and working together with the authors and the authors' company had been a great pleasure and that he would like to touch on a few points regarding some of the decisions and problems as these had looked to his company before knowing whether they were solved or not. The decision 3 in Fig. 22 showed that ships with a capacity from 900 to 1500 containers and speeds from 22.5 to 27 kn. were considered in the first planning stage. Out of this first survey came the request of a fast single-screw container-vessel. For the conditions of the day the turbine plant was by far the best choice and the company thought that the steam turbine, with today's fuel prices, could also compete in the high-powered, fast, single-screw containership.

After the single-screw version had been chosen it became clear that the most important single problem, to be faced was to avoid undue vibrations. Reasonably strict limits in the contract specification had been defined but it did not help very much if this left one with a vibration problem at the time of the delivery of the ship.

A survey of 25 fast single-screw ships indicated that a high portion of those ships suffered from vibrations to such an extent that their operation was affected.

Faced with those alarming facts the company had set up the Table shown in Fig. 23 which simply indicated the problems which they had felt should be looked for and where they could find people with whom such problems could be discussed. The valuable discussions which had followed were here acknowledged with thanks. The most important thing about Fig. 23 was to recognize the problems early enough and that meant during the model testing stage - where the wake field could still be improved.

The request to the shipyard for further investigations had been well received but Mr. Kappel thought it was fair to state that during these investigations no one was able to state that the ships would be free from undue vibrations. This could be illustrated by reference to Fig. 24, and comparing the order of magnitude of axial and transverse forces K_x and K_{yz} of the ships described by the authors and three ships of the same class built by Flender Werft. From these excitation numbers one had to conclude that the Flender ships should be even less affected by vibrations than the Blohm + Voss ships. It was, however, the other way around. Whilst the Flender ships behaved within the limits set for vibrations and could be considered as exceptionally quiet and free from vibration.

The paper mentioned that the owner and the Class of vessel wanted the shafting system so arranged that the whirling critical speed was well above the entire operating range. It was good practice to arrange matters so that no critical frequency was excited within the operating range. As it could be seen from Fig. 25 this was not possible. In the case of the shafting system, where the whirl frequency was greatly influenced by the propeller overhang and the distance from the stern tube bearing to the next bearing, the achievement of a subcritical system would have been possible, but it would have made a

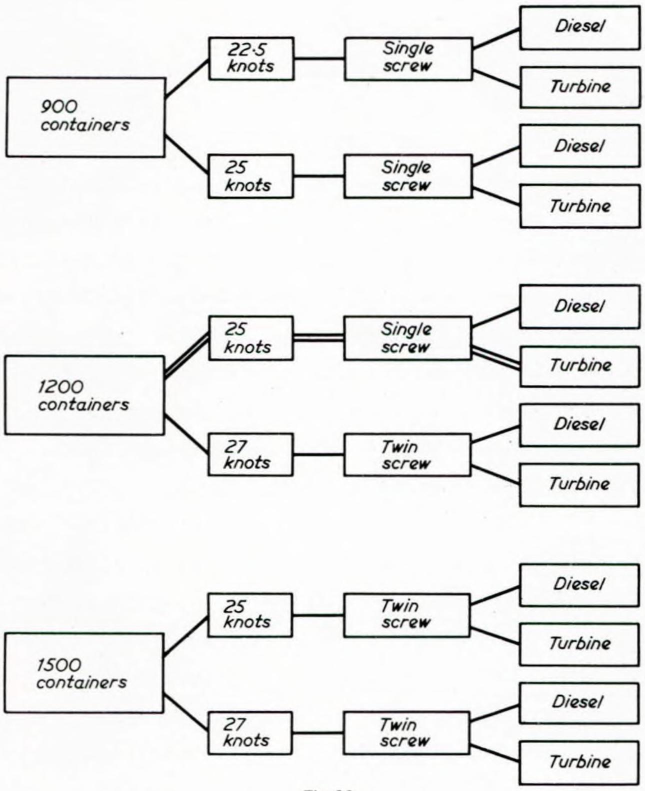


Fig. 22

	Shipyard	Model basin HSVA	Model basin Wageningen	Model basin NPL	Lloyds	NV	Stone	Zeise	Shipyard consultant	
Propeller-Hull clearance	X				X				X	
Wake diagrams	X	X	X	X	X	X	X	X	X	
Blade number, 5-6-7	X	X					X	X	X	
Propeller diameter	X						X	X	X	
Propeller loading	X	X			X		X	X	X	
Propeller cavitation	X	X					X	X		
Tip-Hull vortex cavitation			X			X				
Whirling	X				X					
Shaft torsional vibration	X									
Shaft axial vibration	X									
Stern tube forces and moments	X	X			X		X	X	X	
After ship vibration	X				X					
Deck house vibration	X				X					
Local structure vibration	X									
Pressure fluctuation on hull	X		X			X				
Rudder vibrations	X				X					

Fig. 23 — A-class containerships

	B & V 887-892	FW 609-611
Kx	0·11	0·04
Kyz	0·23	0·16

Fig. 24

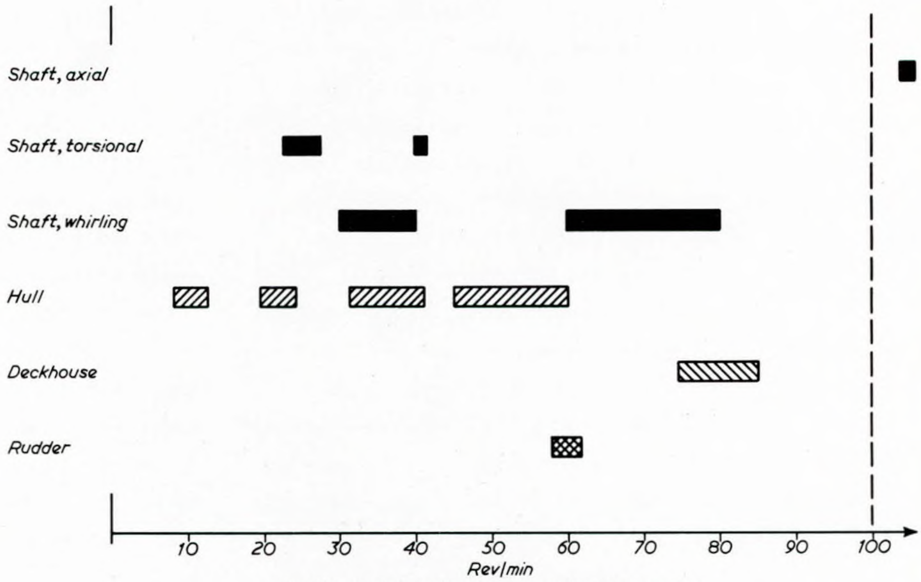


Fig. 25 — A-class containerships — natural frequencies

good alignment of the propeller shaft in the stern tube bearing much more difficult.

It was, however, important to note that in spite of the critical frequencies indicated in Fig. 25 the vessel could operate in the entire propeller speed range without any restriction or inconvenience.

The Blohm + Voss ships had been designed to run at a maximum continuous power of 36,000 s.h.p. (26,480 kW) which this Class of ships was actually using in service. It had, however, occasionally been proved that the ships could run trouble-free even at 40,000 s.h.p. (29,828 kW). This presented a convincing conclusion that the authors were correct in stating that good results could be achieved with single-screw ships with a service speed of 26 Kn. with 50,000 s.h.p. (36,770 kW) provided that careful attention was paid to the flow conditions to the propeller and to the structural design of the ship.

MR. L. SINCLAIR, F.I.Mar.E., considered that the authors had provided a valuable contribution to the transactions covering the performance of a most outstanding ship type. The writer had had the privilege of being associated with certain aspects of the propulsion of those ships and could thus testify to the prodigious amount of detailed prior planning that was involved with every aspect of the design and the very close collaboration which had existed between the various interested parties and, in particular, the very forward-thinking ship owner.

While agreeing that vibration could in some single-screw vessels be an important problem, it might be better if it was called "wake induced" rather than "propeller excited" and the authors acknowledged this in accepting that it was a problem of hull rather than propeller design. In approaching the design in this way, success had undoubtedly been achieved and must stem from the favourable flow conditions to the propeller achieved by a well developed hull form forward of the screw.

As far as single and twin-screws were concerned, it was difficult to understand the authors' surprise to find an advantage in the former against the latter. Single-screw would always have an advantage providing that the power required could be carried without detriment for other reasons, i.e. inability to secure a propeller of sufficient diameter or to avoid overloading manifested by harmful cavitation.

When the speed was such that a power over 40,000 s.h.p. (29,828 kW) was needed, single must give way to twin and beyond this point it was worth considering triple-screws as a satisfactory and efficient alternative to twin.

In considering a simple criterion of performance, it was suggested that the "old fashioned" Admiralty coefficient, i.e.

$$\frac{\Delta^{2/3} V^3}{SHP}$$

would be a much more reliable and sound parameter to use than the function s.h.p./payload x speed. Power was more nearly proportional to $V^{3)}$ than V and more related to wetted surfaces than to displacement or deadweight, i.e. b.h.p. $\propto$ to $\Delta^{2/3}$ rather than to Δ. Furthermore, it was difficult to understand the significance of plotting the function against the year rather than against speed length ratio. Perhaps the authors could comment on this.

Fig. 10 showed the nub of the propeller problem in single-screw vessels of this power. It was impossible for the propeller designer to avoid cavitation and he must therefore come to terms with it. In other words he must make sure that however advanced the condition, attempt must be made to achieve stable sheet cavitation, hopefully without impinging on blade surfaces and causing blade erosion. This would appear to have been achieved in this case but it was doubted that cavitation tunnel tests were yet developed to a sufficient degree for use in a competitive manner as an instrument of propeller choice. In fact, to the writer, competitive model propeller tests either in the cavitation tunnel or in self-propulsion remained a prostitution of the art.

It was now well accepted that cavitation itself had had a magnifying influence on the pressure field around the propeller and it might yet be necessary to tackle the cavitation rather than the erosion problem in later ships. A propeller design might therefore have to be cavitation free rather than erosion free which at the moment was one of the principal requirements of design.

MR. C.F.W. EAMES, M.R.I.N.A., drew attention to the authors' very detailed design work in trying to improve the flow into the propeller and by using contours of wake for correlation between aft end lines and the effect on the propeller. It was good to see that virtue had been rewarded by propellers which had shown no signs of erosion and by ships with very low vibration levels.

This was an example, the results of which provided a considerable incentive for others to follow.

Fig. 14 was interesting and provoked a question.

It was said that the final sternbulb increased the fluid velocity at the propeller's 12 o'clock position.

Careful inspection would appear to indicate the reverse, an actual reduction in the average V_a (i.e. an average increase in wake). The beneficial effect of this particular form might well come from the clearly more gentle entry and exit from the 12 o'clock position as indicated by the wake. Here again there might be a worthwhile lesson for others to follow.

MR. J.S. CARLTON, M.I.Mar.E., said that it was very encouraging to read of the considerable effort, consistent with the restrictions and limitations of commercial design practice, that the authors had devoted to obtaining a good wake flow into the propeller.

Unfortunately, we all too often found from our full scale investigations into ship propulsion and hull and machinery vibration, that this vital consideration in the design of a new ship was undertaken only in a superficial manner - if at all. Consequently, this resulted in many cases in high levels of forced vibration throughout the ship due to propeller excitation which caused, at best, discomfort to the crew, and at worst the failure of some part of the ship's structure or its associated machinery components.

However, in the case of the ships under discussion in the paper, the use and development of this particular type of stern bulb had undoubtedly contributed greatly to the suppression of some of the more unsatisfactory wake characteristics in the propeller aperture, thereby high-lighting the importance of attention to the details of the waterline endings and their effect on the water flow into the propeller disc at the design stage.

The authors in the opening paragraphs of their paper, mentioned the consideration that they had given to the sea-keeping and manoeuvrability characteristics of the ships as being a prime feature of the design study along with the propulsion, vibration and strength aspects which they have dealt with at some length in the paper.

It would, however, be interesting to learn more of the details of these other areas of the design study and how the ships performed on trial.

The cavitation patterns for the two propeller designs shown in Fig. 10 of the paper, both exhibited similar fairly stable back sheet cavitation characteristics with only some minor signs of the least welcome types of cavitation.

In the paper no mention was made of the relative state of the tunnel water with respect to its nuclei content at the time of the tests. Since this parameter has been shown to seriously alter the nature of the observed cavitation patterns in the cavitation tunnel, could it, therefore, be assumed that the tunnel contained a similar nuclei content at the time of each test. If this was the case then it would be interesting to know which of the two propellers were finally chosen and upon what basis the authors had eventually reached their decision.

Correspondence

PROFESSOR Dr. -Ing. O. GRIM wrote that the successful construction by the authors and the shipyard Blohm + Voss of a high speed single-screw ship, whose level of vibration was unusually low, was remarkable, and justified the large amount of space admitted in this paper to the endeavours for freedom from vibrations. Once more it had been proved that a ship with low vibration levels could be built, when all available possibilities were used.

It was not possible to balance the various described arrangements against each other and to determine the success quantitatively. He would like to point out the large number of blades of the propeller and the large propeller-hull clearances with regard to their effect on the exciting pressures and forces on the hull of the ship, produced by cavitation. From several publications which had appeared in recent years it was known that the forces and pressures could become very big. If this was not observed with the ship under discussion, although the cavitation (see Fig. 10) had grown to a remarkable proportion, it was probably due to the aspects mentioned (large number of blades and propeller hull clearances).

The question arose as to why the trend to propellers with a large number of blades did not become more obviously evident especially since a notable loss of efficiency was not expected with 6 blades. The wish for large propeller-hull clearances, however, was leading to undue impairment of the propeller diameter and the efficiency. It was probably still an open question which choice of this parameter would be the optimum one.

In conclusion Professor Grim wished to remark upon Fig. 11: This figure showed that the system was overcritical with regard to the transverse vibration of the propeller shaft of 1st mode and the excitation of 6th order. It further showed that the resonance, which, with regard to the transverse vibration of 2nd mode should be given in a horizontal direction, could obviously not be observed.

MR. K.V. TAYLOR, F.I.Mar.E., wrote that these ships, built by Blohm + Voss, and similar ships built by Flenderwerke for the same owner, represented advanced engineering concepts in ship powering, and while some of like power had been built in Japan, there was little information on their performance in service.

Lloyd's Register of Shipping, as the classifying authority, was of the opinion that, in view of the considerable research and development carried out on these ships - particularly in the form of model wake experiments and propeller cavitation tests - it would have been advantageous and prudent, in view of the uncertainties that then existed in respect to possible vibration levels from propeller excitation, to have undertaken a comprehensive investigation on one of each of the ships built by the two yards.

Such an investigation would have concerned pressure measurements in way of the propellers in association with comprehensive vibration measurements in the hull, and also the shafting system.

Unfortunately, sponsoring the project through Lloyd's Register, the owners, the builders and the propeller manufacturers proved difficult, a major factor no doubt being the economic climate, and an excellent opportunity was lost to obtain data from these ships. Certainly, the Society would have become involved had problems arisen during trials, but it should be appreciated that it was not only information from troublesome ships which was needed for design, but data from successful ones. After all, the one aspect was what they wanted to achieve while the other was what they had to avoid.

Had these investigations been carried out, much more information would have become available for use in the design of future higher powered ships.

As regards specific points in the paper, there were several he would like to raise. Firstly, while Table III of the paper indicated the generosity of the propeller-hull clearances, he would point out that those recommended by Lloyd's Register were minimum values and should be exceeded wherever possible. However, it was recognized that it was more important that the aft end lines gave a good wake (as in the case of this design) rather than complywith a few specific dimensions which did not always provide a vibration trouble-free environment. Nevertheless, the latter were a good guide where powers were more conservative, that is, small compared with these ships.

Secondly, the location of the superstructure well away from the source of excitation should ensure that the risk of vibration in the accommodation was small, unless a main hull-superstructure resonance occurred at service revolutions, and if the propeller blade excitation was high. The predicted value of a resonance at 9.4 Hz could have given some concern, bearing in mind its closeness to 10 Hz, the service propeller blade frequency.

In general, it was a sound policy to have such resonances clear and above the operational range of first-order propeller excitation though, in this instance, due to low propeller excitation force, it did not cause undue vibration. An examination of this superstructure's supporting arrangements, as referred to in the paper, suggested that the fundamental frequencies could be lower than might have been anticipated based on general experience of similar structures.

While not wishing to condemn outright the prediction method used by the authors for the superstructure resonance, he would be very guarded in its use for an aft-end installation where the penalty of an error in preduction should be much more serious and any close proximity to a resonance flank would result in vibration problems.

Thirdly, the authors had referred to vertical hull vibration, but it was not clear as to the method used for the prediction technique, i.e., "40 discrete discs" or the reference to "the first two natural frequencies at 7.5 and 13.5 Hz". These obviously did not refer to the 2 and 3 node modes, and he asked if the modal forms were checked in order to enable comparison with the predicted data ?

Apart from the above minor criticisms, the designers were to be congratulated on their overall approach. In the past, the difference between good and bad vibration characteristics of many types of ship had resulted more from good fortune than from foresight. However, in recent years, there had been an increasing tendency to consider vibration aspects to some extent in the design stage. The example illustrated in this paper described a design philosophy, including vibration from its initial stage, and their approach was considered largely responsible for the good performance of these ships. This was particularly evident in the research and effort put into optimizing the aft-body hull form to give a favourable wake distribution, and thus minimizing excitation.

Other designers would do well to note the benefits achieved by carrying out such studies at a stage in the design cycle when sufficient time was available to investigate hull form modifications in model tanks, and by applying the results to the benefit of the full-scale design.

MR. P.W. AYLING, B.Sc., C.Eng., FRINA., wrote that he would confine his remarks to the vibration aspects recorded in this excellent paper. The authors were to be congratulated on and admired for the meticulous attention paid to wake induced vibratory forces; there was a lesson here for all to learn. However, apart from incorporating generous hull clearances and making every attempt to reduce the magnification effects due to cavitation (and some would argue that these measures were sufficient), the authors did not appear to him to have given the same attention to the other important component, the hull pressure forces.

The manner in which the wake and pressure forces combine to give a total excitation from the propeller was complex and to the best of his knowledge, a complete understanding and solution had so far defied all attempts. Until this had been achieved, caution would have to be exercised in drawing too many conclusions from one example, however excellent that might be.

While repeating his admiration of their effort, would the authors care to indicate the degree of confidence they had in their abilities to achieve similarly low levels of propeller excited vibration in future designs ?

MR. J.N. EDGAR, M.I.Mar.E., had been closely concerned with the British design of propellers for the Maersk 'A' class containerships described in the paper, and he could bear witness to the intense care and attention to detail by both the shipowner and shipbuilders.

The concept of involvement of shipowner, shipbuilder and propeller manufacturer, together with valuable help and advice from the experimental establishment (in this instance HSVA) was an invaluable one. It was often given lip service but often alas not sufficiently implemented. A clear aim of obtaining the best possible operating environment for the propeller was one which had been clearly demonstrated in the paper. Prevention was certainly better than a partial cure. It was regrettable though that in other less enlightened instances hindsight would sometimes discover what a little foresight could have avoided. Where no attempt had been made to consider the proper inflow into the propeller, the designer had perforce to do the best he could. This was usually sufficient, but sometimes troubles could arise, such as vibration and cavitation erosion.

In emphasizing the desirability of improving the flow into the propeller, the authors were endorsing the contention that the increasingly discussed topic of 'propeller excited vibration" should more properly be described as "wake induced" vibration, and cavitation. One could not after all blame a car's wheels for bouncing it over a ploughed field. It was the environment which was the culprit; so why look to the propeller when the ship vibrated.

The analogy was in fact rather exaggerated since it was possible to design a propeller with the intention of minimizing the undesirable effects of working in a mixed wake. This could involve choice of blade section chord lengths, choice of centre line camber and camber chord ratio, pitch distribution etc., all of which can be summarized as choice of blade loading distribution. Other factors such as skew and rake also enter into consideration. A proper examination did, however, require knowledge of the hull form and wake pattern, together with, ideally, the use of the cavitation tunnel or depressurised tank as a design instrument.

Cavitation tunnel tests could include simulation of tangential as well as axial wake components. These should at best be obtained by use of a dummy model hull forward of the propeller. The method of using wake grids to simulate axial components and of increasing or decreasing the propeller rotational speed to simulate tangential effects seemed to be a compromise which should be treated with some care. The specified adjustment to rev/min could only apply to one particular blade radius at one angular position in its revolution. This meant that the rest of the blade both chordally and radially could be operating with an inappropriate loading and cavitation number. The observed cavitation phenomena over the whole blade would therefore not be truly representative. At best the simulation of tangential effects by adjusting rev/min could only be acceptable for studying very local effects. It should also be ensured that no tangential or rotational flow effects existed within the tunnel itself.

The propeller designs evolved for the Maersk ships were prepared by two reputable manufacturers, one in

England and one in Germany. The evolution resulted in two not dissimilar designs as would be seen from the paper. Apparent differences in the shape of the cavitation patterns could be attributed to differences in blade skew. The extent of the sheet cavitation was also similar. Although some differences were apparent at some angular positions it should be borne in mind that the size of the observed phenomena was not the only criterion; stability of the sheet and its thickness were other factors which should be noted.

With regard to the development of future newbuildings based on the subject ships, did the authors envisage conducting any full scale examination of propeller or wake induced effects ? Such investigations involving visual and photographic examination of the propeller through viewing ports in the hull, and the measurement of pressures on the hull above the propeller would be a valuable contribution to the sparse fund of knowledge on correlation between model and full scale results.

Also looking to the future, the authors contemplated an increase of power up to 50,000 s.h.p. (36,770 kW) on the single screw. It was conceivable that even higher powers could be accommodated although limitations of draught might lead to the need for higher rev/min with a consequent fundamental loss in efficiency.

It was possible that some thought might yet be given to the contra-rotating propeller configuration This would have some obvious advantages such as: i) a gain in efficiency by recovery of rotational slipstream losses; ii) the use of lower rev/min leading to higher fundamental efficiency; iii) two propellers sharing the load without sacrificing the wake gain effect of the single-screw layout and iv) reduction of cavitation and vibration consequent on the reduced propeller loadings.

Would the authors care to comment on this suggestion, which seemed to fulfil the criterion called for on innovation or a change in technology as stated in the paper. The capability for the hydrodynamic design of the contra-rotating propeller already existed; the main difficulty seemed to be one of engineering but doubtless this could be overcome if the demand arose.

Finally, whilst it was realized that propeller excited, or more correctly wake induced, effects were causing problems was it not outside the realms of possibility that the response of the ship's structure itself could be diminished by, for example, the localized use of materials with a higher intrinsic damping capacity and also extrinsically by building into the structure itself a capacity for yielding to the imposed forces.

AUTHORS' REPLY

Feeling that much more consideration had been given to this paper than expected, the authors thanked all participants in the discussion for their valuable and inspiring contributions.

In reply to Captain Dilling, the yard would first like to express its great appreciation of the full co-operation of the owner. At all times it was extensive and fruitful.

If the ships had been a success, this was mainly due to the persons representing the owner. They had always concentrated their efforts on the essential points, if not academically, in a most professional manner.

It was a pleasure to hear that the owner acknowledged the efforts of the yard, not only to fulfil the contract, but to reach the best result possible. What should be mentioned was that the yard was not led by false ambition but by the perception that a vast amount of money and even professional reputations were at stake. Due to the technical risk with high-powered, single-screw ships, everything possible had to be done to help avoid vibration problems. It was the fear of running into such problems which had given rise to considerable investigations - and finally success.

As shown in the paper, the ships had exceeded the contract speed considerably during the trials. If the ships in loaded condition had not attained a given speed, this meant that they must have been considerably effected by the environmental conditions.

The vessels belonged to a class of fast running ships; their resistance curve had an extremely low gradient which meant that a small increase in power caused a large gain of speed; but against that, a rather small increase in resistance could cause a considerable loss in speed (Fig. 26). Additionally, the height of the deck load was an important factor. With containers in four tiers (10 m) on deck, the windage area of these ships was equal to that of the biggest sailing ships. Consequently additional resistance components were induced by the drift angle and the rudder angle necessary to compensate for the yawing moment. Finally, it should be considered that the resistance increase caused a loss in propeller efficiency due to a shift of the propeller operating point. It was a fortunate decision to install a steam turbine instead of a diesel motor, since otherwise the problems related to the motor characteristics would increase.

The authors thought that the question of speed losses under service conditions with fast container-ships needed detailed investigation. Therefore, they had started a research programme supported by the Government of the Federal Republic of Germany which would provide results within the year.

Another aspect mentioned by Captain Dilling was the steel weight. Widespread and well-founded experience stated that "heavy" ships would rarely have vibration problems. This, however, was not generally true for fast single-screw vessels. It was important to insert the additional steel in the right places. If this was done - and it had been carried out here - it would probably be one of the best ways to overcome vibration problems.

Mr. Rapo had drawn attention to some high-powered, single-screw containerships already in service for some years. Indeed, these might be called a new generation of containerships.

Mentioning the low block coefficient, Mr. Rapo had stressed the most important aspect in ship design, namely to choose the right main data. The issue of the battle for the economic ship and the solution of its technical problems - headed by the vibration problem - was determined in the initial design stage.

Searching for a suitable ship form, the old Taylor diagrams were very useful in the present case since they exhibited a distinct minimum of the residual resistance for $C_P = 0.54$, corresponding to a block coefficient of about $C_B = 0.5$. Together with the aspect of high form stability necessary for the operational economy of containerships, the designer was forced to choose a low L/B ratio, and frames of markedly V-shaped character, respectively. Both had detrimental influence on the formation of a beneficial wake pattern. That problem, however, had been solved by adopting a special stern bulb that was discussed in more detail below.

In this connection, thanks should be expressed to Mr. Lockart and to Mr. Rapo, of Lloyd's Register of Shipping, in London. Before signing the contract, the question of low block coefficient had been discussed in detail with them and the owner. Contrary opinions gave birth to new ideas and thus the foundation was laid for today's improved ship designs from the yard.

Mr. Garrett had introduced several questions. The first referred to the frequency measurements during the trials. These were carried out using electrical pick-ups and, therefore, the records contained all frequencies in the normal measuring range. The records, however, were only made on paper, so that comprehensive evaluation was impossible. Inspection of every record showed that the blade rate frequency clearly predominated, hence only this frequency was evaluated.

The second question referred to propeller design aspects. The authors could not answer this question because the propeller design was in the hands of the propeller manufacturers. However, they had given relevant information during this discussion which was considered to be illustrative enough. The open water performance of both propellers was nearly identical.

For calculation of the hydrodynamic propeller forces, the HSVA had used a three-dimensional steady propeller lifting-surface theory. Details of this theory were given in Ref. (1). An abstract of this paper in English was published in Ref. (2). The nominal model wake pattern was applied in the calculations.

The last question referred to the propeller rev/min: At the owner's request, a low shaft rate of about 100 rev/min had been considered from the beginning. The only aspect then was to increase the propulsive efficiency.

Mr. Sinclair, in drawing attention to the favourable flow conditions towards the propeller, and withdrawing it from the importance of good propeller design, could not remove the authors' conviction that a good deal of the success was due to the capability and experience of both of the propeller manufacturers. It should be noted that the tank predicted ship speed instantly increased by half a knot when using just the initial propeller designs of both producers as compared to the formerly used stock propeller.

Mentioning the restricted value of the tank prognosis, dependent on the small size of the propeller model, Mr. Sinclair certainly had in mind that it was, in particular, the two ships fitted with his company's propellers which had attained considerably higher trial speeds than the tank tests predicted. This might indeed be attributed to some existing uncertainties accompanied with model/full-scale correlation of fast ships.

The authors, however, could not join in a general criticism of propeller model testing. It was only too true that the model experiment was the best and most reliable method of full-scale prediction available. There were some well known drawbacks involved

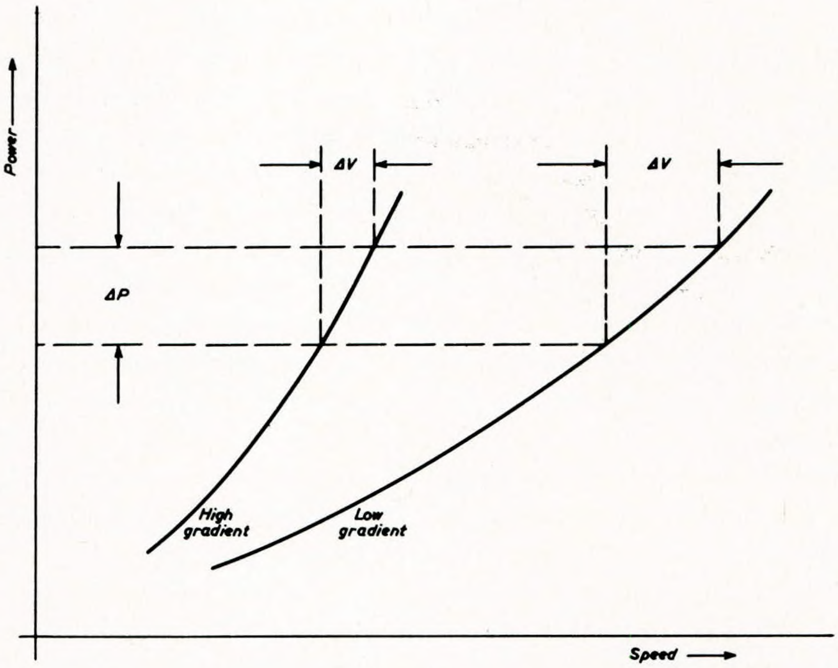


Fig. 26 — Effect of the gradient of the power versus speed curve on speed loss due to a resistance increase

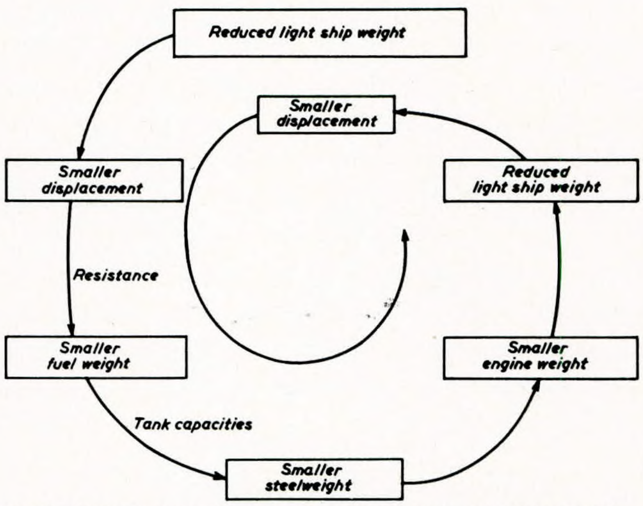


Fig. 27 — Spiral showing gradual improvement of a ship design induced by reduced light ship weight

in model testing, but these were far from sufficient to reject model test results in general. Therefore, the authors considered that the most objective method for the comparison of two different propellers was to test them under equal conditions in the same cavitation tunnel. Since the models, additionally, were produced by the same Institute and to the same scale, there should be no objections to having obtained comparable results.

Concerning the question of single or twin-screw arrangement: the propeller manufacturer and the shipyard would probably take this to have different meanings. Undoubtedly the single-screw ship had less power requirements due to better utilization of the wake and the absence of struts. Savings of 10 per cent to 15 per cent were generally taken into account by the yard.

Further advantages were as follows:

- The weight of the power plant was less; the capacity of the bunker could be reduced due to lower fuel consumption. Consequently, by keeping the weight of the payload constant, the block coefficient could be reduced (leading especially to finer afterbody lines) which in turn would lead to further savings in weight and installed power. This could be represented by a spiral as shown in Fig. 27.
- Improvement of the manoeuvring characteristics when compared with a twin-screw ship, having one rudder;
- Smaller investment for ship and engine.

On the other hand the twin-screw ship, although much more expensive, was affected by a much smaller technical risk. Generally it would be easier to avoid vibration problems.

The criterion used in Figs. 1 and 17 of the paper was not intended purely to give an indication of the quality of the ship lines or of the propulsive efficiency. The authors agreed with Mr. Sinclair that, for this purpose, the Admiralty constant would serve best. Here, however, a relation was sought which could be used to assess the whole complexity of a ship design. That meant that the ship's ability to transport a certain payload, within a certain time, along a certain distance, had to be compared with a typical factor representing the major part of the costs for this effort. These costs could, from a design point of view, be represented by, say, the mass of fuel required or - as finally chosen - by the engine power required to fulfil the task. It should be mentioned that the criterion chosen also considered stability, since the amount of payload (containers) was greatly affected by the vertical centre of gravity and, hence, by GM, or the righting arm. Thus it seemed that the economy of a ship design could be assessed to a great extent.

Mr. Eames had referred to the new B + V stern bulb, and the authors offered a brief description of its development. In the initial design stage the ship's afterbody and the involved hydrodynamic problems were treated as a "black box". It was completely open as to how and whether it would be possible to produce a sufficiently good wake field by using wide V-shaped afterbody frames. Such lines provided high form stability but, on the other hand, had a negative effect on the inflow towards the twelve o'clock position of the propeller.

Different stern arrangements, such as the Grim-tube, the A.G. Weser-bulb, and several other stern bulbs were tested in the model tank in order to gain substantial knowledge on possible solutions.

Temporarily the idea of applying a stern bulb was dropped, but then it became evident that the shafting and the stern tube required a considerable amount of space, and rigidity of the supporting construction, respectively. Thus the stern bulb came in again. Yet the production of the hull was already going on, and an alteration of the afterbody lines had to be as limited as possible. Thus a small bulb was developed which, among other things, was expected to deflect the normally upwards directed flow towards the propeller more horizontally.

After successfully completing the Maersk ship test programme, the yard continued tests to search for further improvements by applying the new stern bulb on other ships. Though this programme had not yet come to an end, it had been possible to present some results in Fig. 15 of the paper. Unfortunately the drawing of the wake distribution of HSVA hull form 2632 became available only after the print make-up of the paper, thus it was given here in Fig. 28. Model 2632 had a remarkably low ratio L/B = 5.3 and extremely V-shaped afterbody frames yielding a minimum KM/B = 0.475. The reduction of the wake peak was obvious in comparison with the wake fields shown in Fig. 14 of the paper.

It would seem that the new stern bulb, now being a B + V patent, could be recommended for all types of fast single-screw ships which, due to V-shaped afterbody frames, were suffering from an unfavourable wake distribution.

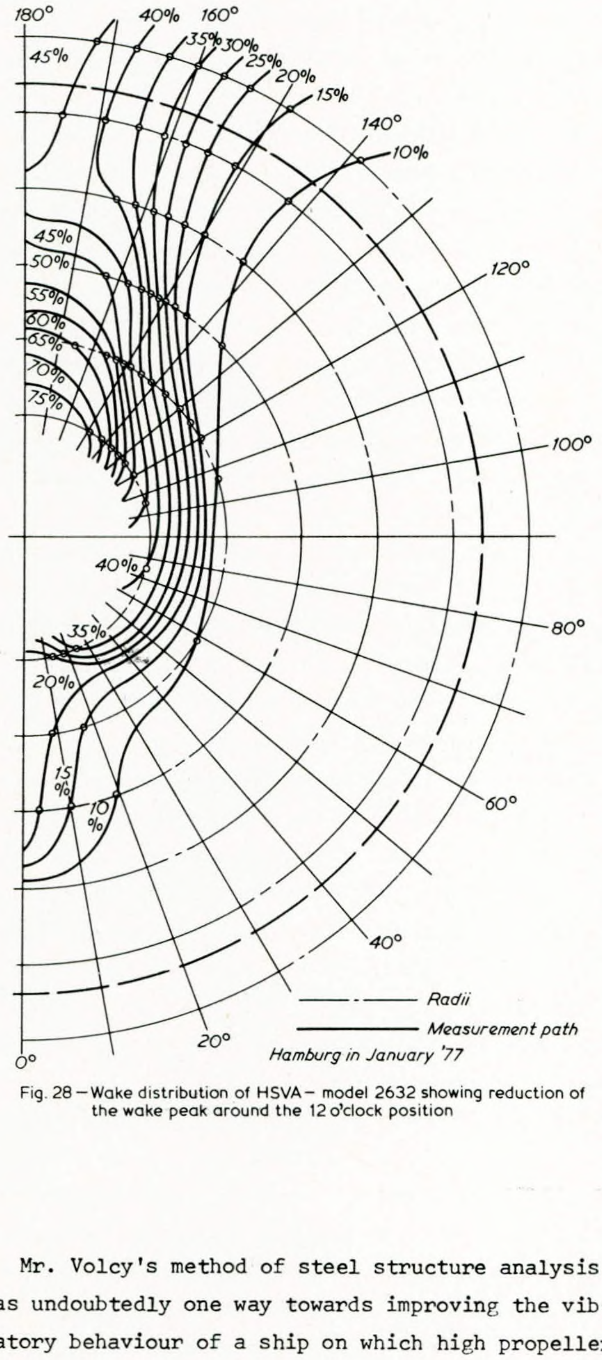


Fig. 28 — Wake distribution of HSVA – model 2632 showing reduction of the wake peak around the 12 o'clock position

Mr. Volcy's method of steel structure analysis was undoubtedly one way towards improving the vibratory behaviour of a ship on which high propeller forces were to be expected. The reported success was admirable and encouraging. However, since the method of detuning resonators was also widely applied on the Maersk ships and was well-known practice in most shipyards, it would be better not to call it an "unconventional approach".

The exposition of Dr. Meyne's propeller design concept deserved the greatest interest. For the success of this concept it should be remembered that there were no signs of erosion on the propeller blades after one year in service.

Mr. Kappel had gone back into the history of his company's investigations to find the best possible ship for the intended container service. Being without practical experience with containerships, the owner had made a decision for the single-screw arrangement as a solution offering the lowest operational costs; however, this bore a high technical risk. The yard, on the other side, was well experienced in designing and building containerships and had tried for many years to convince owners of the advantages of single-screw ships. The yard had in mind the lower building costs of this concept offering advantages in the competition with other shipyards quoting for twin-screw ships. So the two parties had amalgamated for their mutual benefit.

As Mr. Kappel had pointed out the choice between diesel engine and steam turbine should not purely be based on considerations about the fuel rate, but the whole complexity of the installation should be considered. So in certain cases the considerably higher weight of a diesel plant could turn the economy in favour of a steam plant.

As for K_x and K_{yz}, it was not advisable to draw too many conclusions from these coefficients. In no case should the K_x and K_{yz} values of two ships be compared if they had been calculated by different authors. This, however, held good for the Blohm + Voss and Flender results, respectively. The method used to obtain the Blohm + Voss results produced roughly twice the values obtained by the method used for the Flender ships.

It was agreed that over-critical design might have the drawback that resonance due to higher order excitation could not always be avoided near the nominal number of revolutions. This, however, was mostly harmless since the strength of the higher order exciting forces decreased rapidly with increasing frequency

The optimism that Mr. Kappel exhibited with regard to even higher powered single-screw ships might result from the success with his company's first containerships. However, as the essence of experience, the authors wanted to summarize that the design of high-powered, single-screw ships should always be treated as a special task requiring more time and money for intensive investigations than was generally supposed.

Mr. Carlton had said that he missed the results of manoeuvring and seakeeping tests. These were not given in the paper for reasons of brevity, i.e. in order to concentrate on the more significant problems of speed and vibrations.

Seakeeping tests had been performed in the HSVA at 11.19 m draught in irregular head and stern seas at four sea states. The results had shown excellent seakeeping behaviour. There were very small motions in head seas up to a sea state according to BN 6, power unreduced. The same held for stern waves up to about BN 8. It had been attempted to achieve good seakeeping behaviour of this design by slender fore and afterbody frames that were only moderately flared.

The speed loss in head seas was characterized by about half a knot at BN 4 1/2 to 5, growing rapidly further on. In stern seas a loss of half a knot would occur at BN 9.

According to the HSVA statistics, the manoeuvring test results were within the normal range for good ships, in spite of the low L/B ratio. Fig. 29 showed a turning circle obtained during one of the trial trips.

Concerning the nuclei content of the tank water during cavitation tests there was only little to say for shipyard personnel as this was a problem for the model tanks. Since it was an open question to date as to how high the nuclei content of seawater might be, it was a matter of philosophy which nuclei content should be maintained during cavitation tests. Whilst on this subject, the HSVA, where the cavitation tests had been conducted, rendered the tank-water to a standard condition of nuclei content before each test. According to cavitation observations of constantly running propellers this condition could be maintained for hours. Since full-scale observations of cavitating propellers showed good agreement with the corresponding model observations, there apparently existed an accordance between model and full scale conditions.

Professor Grim, among others, had drawn attention to the propeller/hull clearances and the number of propeller blades which, in his opinion, were of significant influence on the level of vibrations. Large propeller/hull clearances did not provide in themselves the warranty of small vibrations. Normally, they caused large angles at the waterline endings and these should, of course, be avoided. Thus, there was an interaction of different influences, all of which had to be observed.

Mr. Taylor had expressed his disappointment about just dropping the joint intentions, of owner, shipyard, propeller manufacturers, and classification society, of performing full-scale measurements near the main vibration exciters. At least with the first ship it was a shortage of time which prevented respective measurements. Afterwards the interest of the majority of the partners decreased as a consequence of the fine performance of the ship.

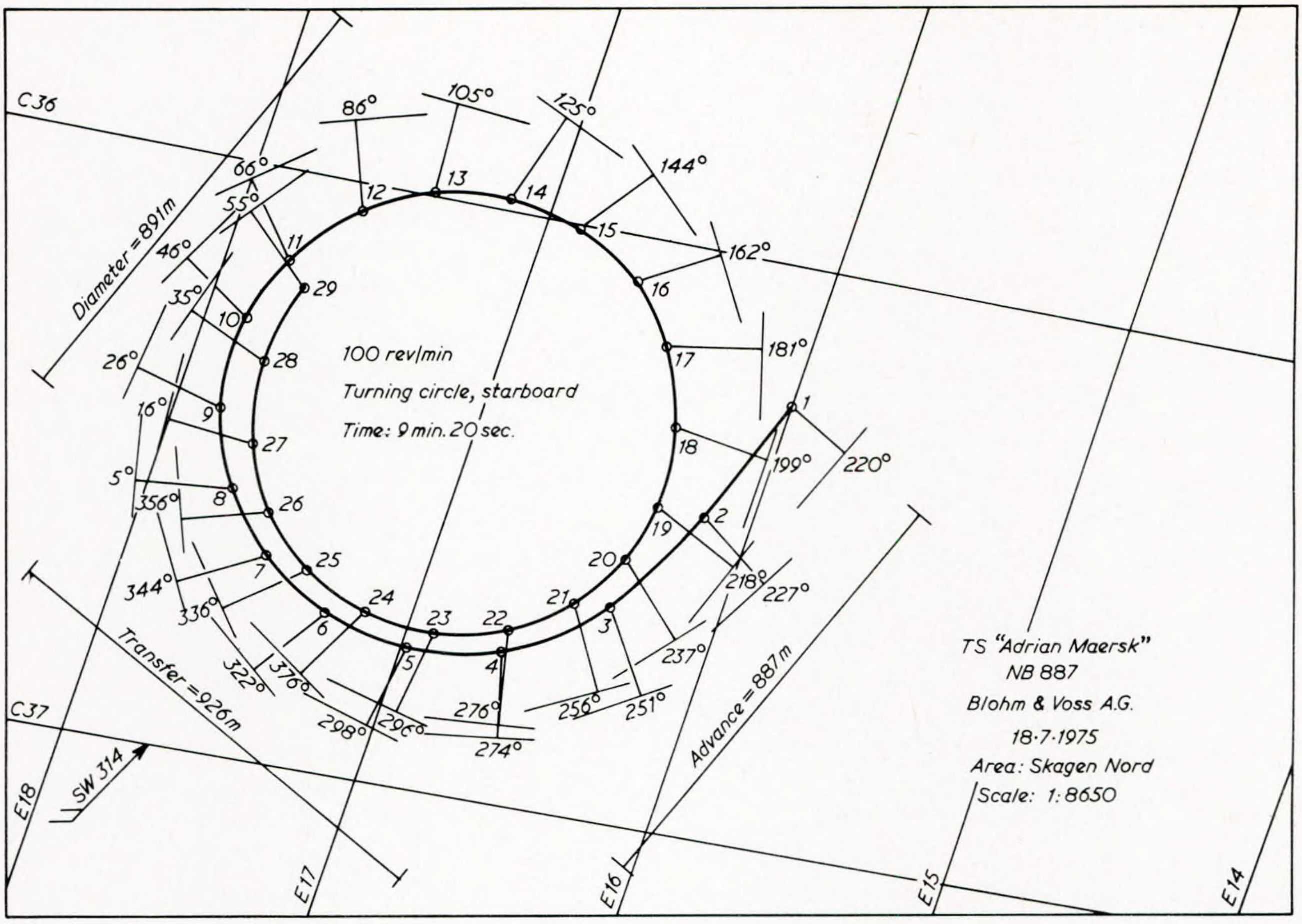
The authors could easily agree with Mr. Taylor's first two points, with only a few amendments. Concerning the propeller/hull clearances recommended by the classification society it was just practical experience that recommendations were very often taken as a rule.

Could the addendum be made that good propeller inflow was more desirable, together with some hints on how to achieve it ?

Regarding the overcritical design strategy the authors were still convinced that its results were equally as tolerable as they were low in cost. Shifting the natural frequency to higher values always meant building-in additional steel weight. Excitation with a frequency higher than the main exciting frequency (e.g. propeller blade frequency) generally meant smaller exciting forces.

The vertical hull vibrations (third point) were calculated, according to the Gümbel-Csupor method, by replacing the ship hull with a system of mass discs and springs in series totalling 40 units. By denoting the natural frequencies the authors had made a mistake. The correct values of the first two natural frequencies were obtained by dividing the given values by 6, resulting in 1.1 and 2.3 Hz for the calculated, and 1.3 and 2.3 Hz for the measured values. These were likely to represent the 2 and 3 node modes, respectively. The modal form was only checked at one single control point and a very satisfactory agreement with the prognosis was found.

The comments that Mr. Ayling had made on the hull pressure forces indicated that knowledge in this field was still developing, and it was just in the very initial phase when the design of the Maersk ships was started. Results of model measurements in cavitation tunnels were then said to be uncertain. Furthermore, the time needed for the preparation of respective tests was long in comparison with the time available in the ship design stage. Another point was the tremendous costs caused by such tests. Consequently owner and shipyard agreed to concentrate on other items such as the improvement of the axial wake distribution. Both, of course, had nourished the hope that a fine wake field, causing only a small extent of cavitation on the propeller blades, would also be favourable with respect to the hull pressure forces. After all, some points related to this problem were mentioned in the paper, and this discussion



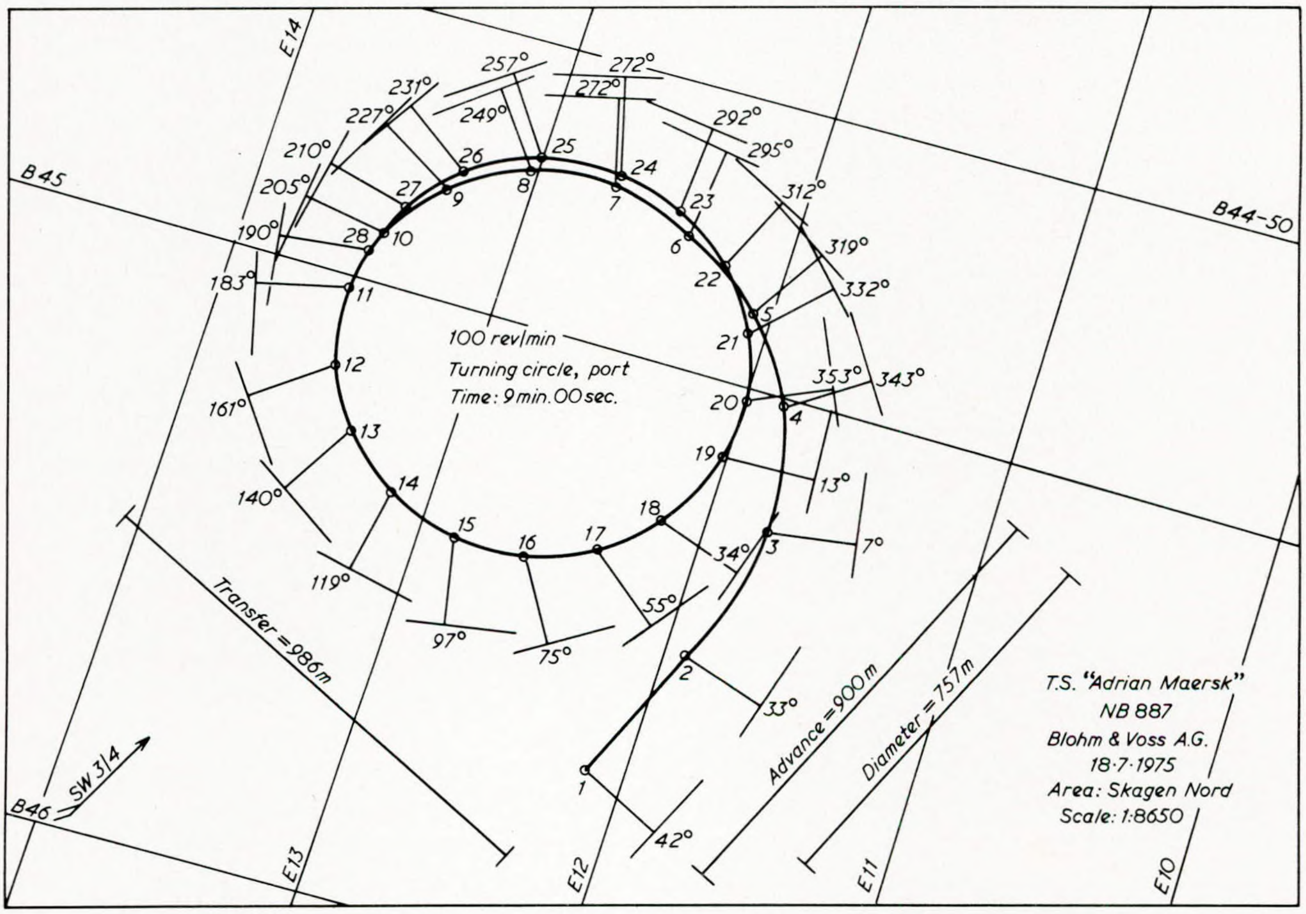


Fig. 29— Port and starboard turning circles measured on the trial ship of one of the Maersk container ships

had shown that some care was taken although less than with respect to the shaft forces.

That a dummy afterbody model mounted in the cavitation tunnel was more suitable to simulate the spatial velocity field in the propeller plane, as stated by Mr. Edgar, was readily agreed. There were, however, certain limitations of size in a number of cavitation tunnels and, here, the method described in the paper must be applied. A large improvement was expected by performing cavitation tests in a low pressure circulating water tunnel, having the whole ship model in front of the propeller. Moreover, the authors wished to express their firm conviction that further progress in developing reliable full-scale wake models was needed to eliminate the scale effect imposed by using the model wake for special propeller investigations. Full-scale measurements were also needed but, as was well known, these depended largely on the readiness of the shipowners to provide a ship.

The contra-rotating propeller configuration could be a feasible solution, particularly for ships sailing at about 26 kn and more, i.e. under conditions traditionally reserved for twin-screw vessels. The possible improvement for a typical high speed containership was shown on Fig. 30 (compared with Fig. 17 of the paper). The comparison was based on the single-screw ship indicated "1976". The dotted line "a" referred to the same ship with twin-screw arrangement, line "b" to the ship with contra-rotating propellers. Indeed there was a certain superiority characterized by line "b". The estimated technical risk, however, was likewise much higher than for the other two arrangements.

Relying on the steady progress in the understanding of problems like cavitation, vibration excitation, or hull structure analysis, the authors would not hesitate to do the next step towards the 50,000 h.p. single-screw containership. Partly they felt themselves to be supported by their own experience, but to a larger extent, however, by the international collaboration of experts exercised in connection with the Maersk ships. This collaboration had proved most effective and could not be sufficiently appreciated.

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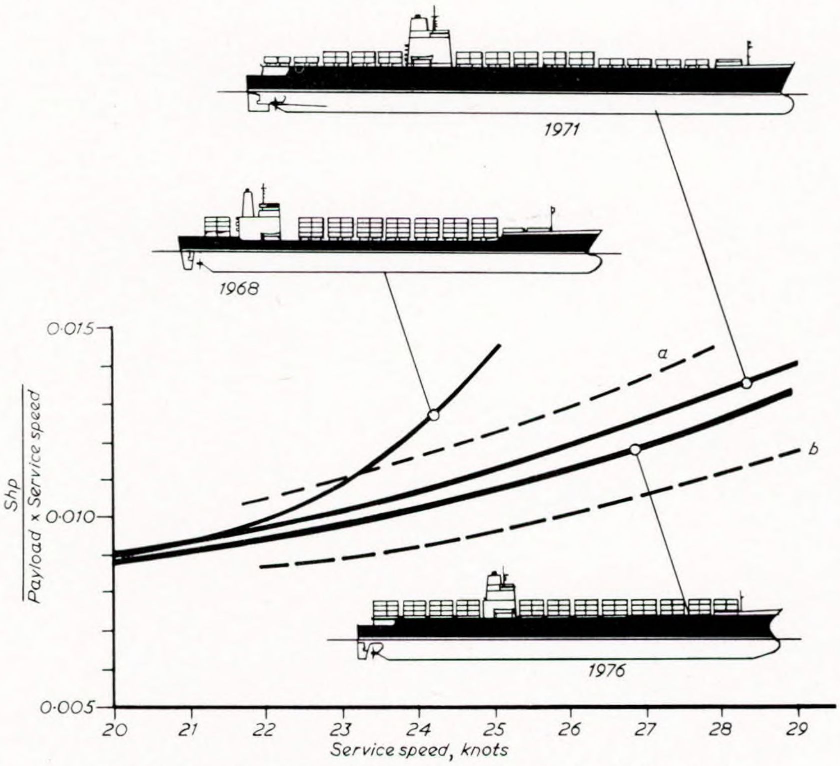

Fig.30 – Dotted line *b* gives possible improvement for ship denoted '1976' due to contra-rotating propellers. Line *a* shows the effect of a twin-screw arrangement (compare Fig. 17 of the paper)