THE PRACTICAL APPLICATION OF COMPUTERS IN MARINE ENGINEERING

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This paper gives examples of the steady state design problems for which digital computers are most suitable, and of the dynamic performance and control problems for which analogue computers are most useful. It describes very briefly the principles upon which they work. The results obtained from a computer when analysing the transient conditions that occur when ships are manœuvred is given in some detail and comparisons are made with ship trial results. The cases investigated cover large and small warships where shaft reversal is achieved by reversing turbines, reversing gears, controllable pitch propellers and electric drive; they also cover merchant ship shaft reversal by direct reversing Diesel engines, gear driven medium speed Diesels with friction clutches, and controllable pitch propellers. Arguments are put forward to show that the best results are obtained by using the maximum amount of astern power available as quickly as possible, rather than by running the shaft slowly ahead or holding it stopped, as is sometimes suggested as the best means.

INTRODUCTION

The paper is intended to give marine engineers a broad picture of the part that computers can now play in the theatre of marine engineering design. The use of computers has grown so rapidly of late that it impinges on practically every aspect of the industry from the initial design, through production, both in the factory and the shipyard, to the analysis of performance at sea and indirectly in all the attendant aspects of book-keeping. It is the use in design that will be considered here.

There are two basic types of computer, the digital and the analogue. A few years ago it would have been possible clearly to define the types of problem for which each was suitable and the other unsuitable. Latterly, however, there has been a tendency for the two to overlap, the digital, which was basically an extremely industrious arithmetician, can now solve differential equations, albeit by trial and error methods and the pure mathematician, the analogue, which has always been adept at the calculus, now uses non-linear tables of figures which cannot be expressed as mathematical formulae. This tendency to overlap is recognized in the development of the hybrid computer, incorporating features from both types.

The distinction is still great enough for it to be convenient to discuss the two types separately and the problems that they are used to solve and this paper is written in three parts, the first concerned with digital computers and typical problems, the second with analogue computers and typical problems, and the third with a subject that demands the use of computers for satisfactory solutions and is of special interest to marine engineers, namely, manœuvring of various ships with different types of machinery giving computed and trial figures for the transient values of speed, torque and thrust.

DIGITAL COMPUTERS

General

The present generation of digital computers has evolved from the mechanical calculators and counting machines, the basis of which is centuries old, but although the electronic digital computer utilizes the basic operating principles to the full it replaces the slow mechanical moving parts by the speed of electrical impulses. The removal of the inertia of moving parts allows computation at very great speeds.

Electronic components and circuits have one simple obvious characteristic in common. Each may be either on or off, in other words each can exist in two states; a number can be represented by a two state device if we write the numbers on a binary scale.

For example the number 237 would be represented by eight bits as follows:

2^7	2^6	2^5	2^4	2^3	2^2	2^1	2^0
1	1	1	0	1	1	0	1
On	On	On	Off	On	On	Off	On

Addition, subtraction, multiplication and division, are all done by addition. Addition is straightforward and multiplication is by multi-addition. Subtraction is done by adding for example 2^8 − the number to be subtracted (say 79) thus:

2^8	2^7	2^6	2^5	2^4	2^3	2^2	2^1	2^0	
	1	1	1	0	1	1	0	1	
									= 237
	1	0	1	1	0	0	0	1	
									= 177 = 256−79

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Adding									
gives	1	1	0	0	1	1	1	1	0

$= 256 + 158$

The 2^8 has to be ignored thus giving $237 - 79 = 158$.

Division is done by successive subtractions; 237 is to be divided by 79. After one subtraction the number is

	1	0	0	1	1	1	1	0
Adding again	1	0	1	1	0	0	0	1
	0	1	0	0	1	1	1	1
Adding for the	1	0	1	1	0	0	0	1
third time	0	0	0	0	0	0	0	0

Any representation of external information within the computer will be equated to an integral number of bits whether this information be numerical or character in form. The actual representation may be varied, with some reservations, since functions, arithmetic process and character handling facilities are normally made automatic and are based on a definite interpretation of bit patterns which will give rise to an optimum representation system for the machine. It might be concluded that because representation of information within these machines has been reduced to a simple code which is so cumbersome that even the simplest operation involves a disproportionate number of steps when compared with the normal human approach to the same operation. This disadvantage has been completely offset by the phenomenal speed at which the machine can handle these bit patterns. An average, but by no means the fastest, speed for modern computers is for the machine to be able to assess 2 000 000 six-bit patterns per second, and be capable of adding the equivalent of 125 000 seven-digit numbers in the same time. The requirement for

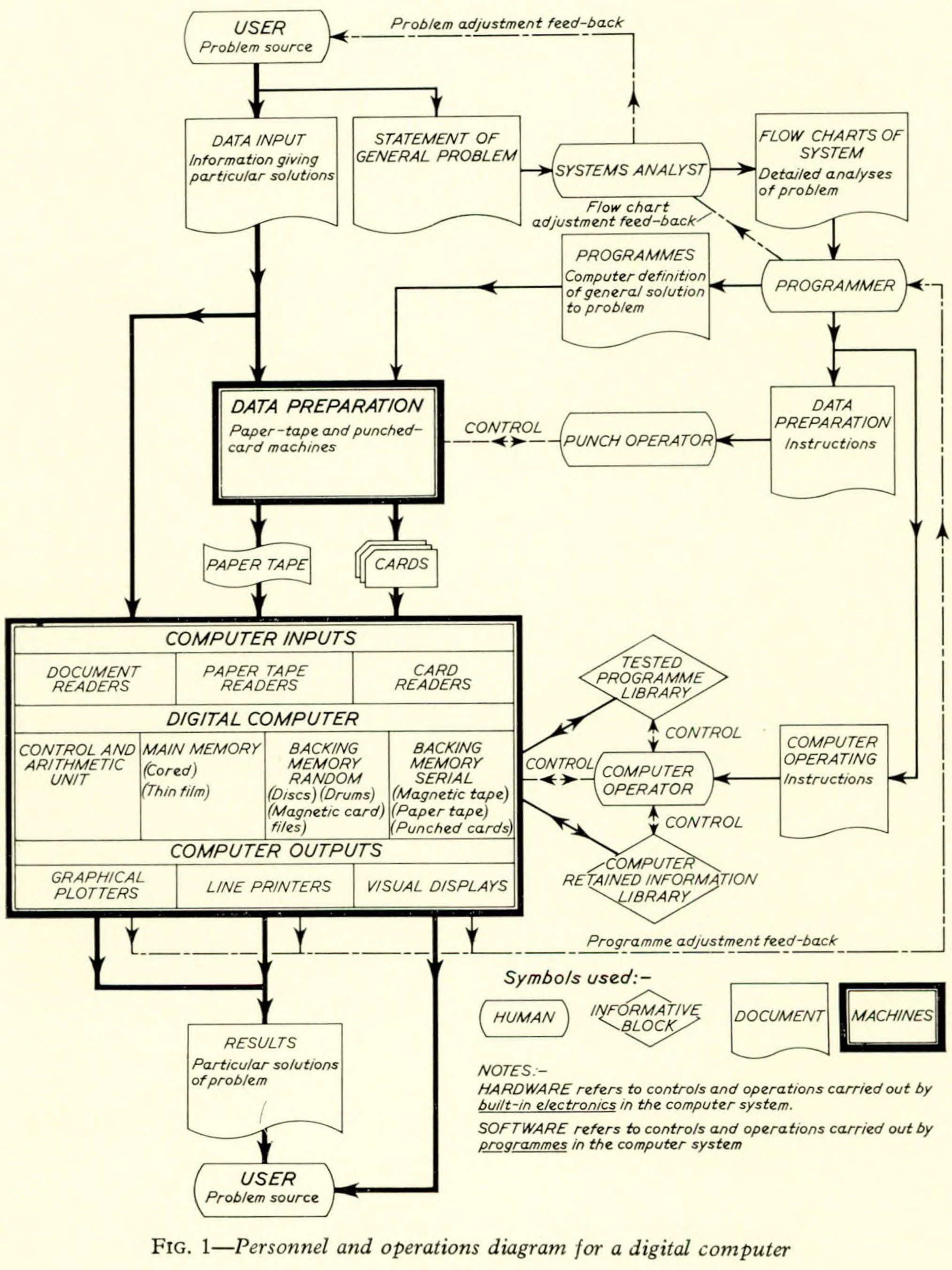

FIG. 1—*Personnel and operations diagram for a digital computer*

speed has become so pressing that even distances at the speed of light have become significant and microminiaturization is beneficial.

To take advantage of this potentiality, it is necessary to provide two things, the programme and a suitable interface between the user and the machine (see Fig. 1). The interface is normally provided as a mixture of hardware and software (see Fig. 1) by the manufacturer and its function is to control the running of the installation efficiently with the minimum of human intervention and reduce the skill and effort required to produce an operable programme. This interface, as the installation becomes more complex, is assuming more and more importance and the time is fast approaching where the actual speed of the installation will be of secondary importance to the interface provided. This can be easily appreciated if we consider the reaction time of a human to visual stimuli which is around 0·5 s to press a given button and in this time it is possible for the machine to have carried out some 60 000 operations. If the human predetermines the pressing of the button and the machine does it itself, the whole operation will be speeded up and the saving in time utilized productively.

Some remarkable advances have been made in this aspect of computer design in the last few years, e.g. the automatic operating systems which are supplied with the latest machines available especially the large multi-programme installations, and the sophistication of high level languages like Fortran, Cobol, and Algol. The increasing capital costs of large digital computers are justified by the greater speeds at which they operate, and the speed of the interface is a significant feature in these.

Reference is frequently made to the remarkable memory storage that can be included in a digital computer some 7×10^8 "words" of on line random access being now commonly used. It is this feature that renders them so useful for Information Retrieval. It is perhaps salutary to remember that the human being, who was quoted above as being so dilatory in pressing a button, has a memory of some 280×10^{18} "bits" according to Adler[1].

Pipe Stressing

It has been realized for many years that steam pipes are liable to low cycle fatigue damage and joint leakage due to the effects of thermal expansion and contraction. Because of this, a pipework designer must, in addition to designing a pipework system to perform its normal duties of conveying steam from one point to another, analyse the system as a load bearing structure. This analysis is quite complex even in the case of the simplest pipe. An ordinary pipe fixed at both ends may require up to a week of hand calculation to establish the stresses existing under thermal expansion. One additional degree of complication, such as one branch of a simple pipe, increases the time required for hand calculation to nearly a month, and additional branches make the time required for hand calculation so long as to be economically impossible. The effects of this load of hand calculation, imposed on the pipework designer, were naturally profound. He invariably designed pipework in such a way as to reduce the amount of calculation required to a minimum and once he had successfully carried out a stress calculation on a pipe, that pipe became a vested interest which the designer would defend against modification or alteration. Naturally, this did not lead to pipe systems being optimized, for weight, space, pressure drop or cost.

In common with many computer applications, computer pipe stressing has not reduced the work of the designer, rather it has increased it, but it has made it many times more effective in securing good and economical design.

Marine Shafting Alignment

During the evolution of a shafting design, the shafting dimensions are determined on stress calculations, and bearing positions are tentatively fixed on layout consideration and on the requirement to keep shafting as flexible as possible within the safety limits of whirling. Computer programmes for whirling permit refinement of the permissible span lengths considerably in advance of what could be done with the simple approximate formulae.

The shafting alignment computer programme will then give the following:

a) Bearing reactions with all bearings on line of sight.
b) Table of influence coefficients. This is a table showing for each bearing, the effect of a displacement (usually 0·010 in) at each of the other bearings. This will show up badly spaced bearings, since bearings which are too close together have large influence coefficients, indicating that the arrangement is sensitive to misalignment.
c) Where an unsatisfactory load distribution is obtained with all bearings on the line of sight, the bearing reaction can be adjusted by raising and lowering bearings by small amounts. Using the influence coefficients the bearing offsets can be readily calculated to give an acceptable distribution of load over the bearings. The relative position of the engine or gearcase flange to the forward shafting flange can be defined in terms of relative height and opening at the flanges at top and bottom.
d) When the shafting design is settled, including any offsets at bearings, as described in (c), a complete set of shear force, bending moment, and deflexion diagrams can be produced.

Steam Turbine Design

It is found necessary during the evolution of certain installations, to establish the optimum steam turbine for main propulsion duties in terms of number of cylinders, performance, weight and overall bulk.

It was considered that a computer solution would meet the requirements best and, consequently, a number of computer programmes were written which vary in complexity from relatively simple routines for calculation of preliminary estimates of steam consumption, gland quantities, etc. to more comprehensive programmes providing complete thermodynamic designs giving turbine staging, blade and nozzle heights, angles, mean diameters, speed of rotation, efficiency, non-bled steam consumption, condenser heat load, gland steam quantities, last row centrifugal stress, turbine weight, etc.

The concept of a comprehensive turbine programme for a computer is basically simple. It entails setting down in logical sequence, the operations carried out manually in designing a steam turbine. The number of unknowns exceed the number of equations thus making a trial and error process inevitable, with iterative loops at many points in the calculation to correct assumptions. Roughly the process is as follows.

The overall adiabatic heat drop is obtained from the known values of inlet and exhaust steam conditions and is divided proportionally to the required power split if the machine is cross-compounded. An assumed value of turbine efficiency gives the exhaust steam condition and the known turbine power gives the steam flow. The exhaust end of the turbine is now designed. The speed or rotation is calculated from a last blade centrifugal stress criterion. If a Curtis stage is required, this is now designed and the wheel case conditions are determined. The gland leakage calculations are made and their effect taken into account where appropriate. The number of stages required is calculated from the known values of inlet and exhaust mean diameter and available heat drop. For constant U/C_0, the heat drop in each stage is made proportional to $Dm^2/\Sigma Dm^2$. The diagram efficiency for each stage is now calculated from the velocity triangles allowance being made for friction, windage, wetness, spillage and leakage. The power developed and non-bled steam rate are calculated and compared to the values assumed at the outset. If agreement is not within the required limits, an iterative loop is entered until acceptable agreement is reached.

Gas Turbine Performance

To establish quickly the performance of known gas turbines in any given hull, a programme has been prepared in which the usual "no-loss" engine performance data for selected ambient conditions (usually 14·7 lb/in^2, 15° C) in the form of curves are transferred to computer data tapes.

The other input data required are:

i) ambient temperature and pressure;
ii) intake and exhaust duct pressure losses;
iii) gearing efficiency;
iv) power rev/min relationship.

Limits on fuel flow, compressor rev/min, gas generator turbine entry temperature and power turbine entry temperature are also given as input data.

Full particulars of the engine performance over the power range required are printed as output data and these particulars include:

1) gas generator turbine entry temperatures;
2) power turbine entry temperature;
3) power turbine exhaust temperature;
4) air mass flow;
5) fuel flow;
6) specific fuel consumption (lb/shp-h);
7) shaft horsepower;
8) power turbine output torque;
9) power turbine speed;
10) compressor speed.

The programme can be used for single or multiple gas turbines on each shaft and in the case of multiple engines, the engines can be of different types and outputs if desired. Thus COGOG (combined cruising gas turbine or main gas turbine) and COGAG (cruising gas turbine and main gas turbine) installations can be dealt with.

The programme is being extended to include the data for a number of Diesel engines and will be available to cope with CODOG (combined cruising Diesel or main gas turbine) and CODAG (combined cruising Diesel and main gas turbine) installations.

Engine Flexible Mounting Systems

It is well known that the reduction of underwater noise levels is of great operational importance in naval construction. An obvious means of attenuating the transmission of vibration from machinery into the sea is to mount noisy machinery on flexible mountings. A particular problem concerns a new anti-vibration mounting system which "constant positions" the mounted item with respect to its seating with great accuracy and, at the same time, provides very high attenuation of vibration.

To obtain the high sensitivity in the positioning control circuit necessary to counteract rolling, pitching and torque forces, the elements of the system must be non-linear. Even with a passive rubber element type of mount, calculations of the relative movement of a mounted item under rolling conditions requires the solution of a set of six differential equations. If it is required to size the elements of the active mounting system to ensure that there is an adequate stability margin to avoid hunting under all conditions and to calculate the movements under sea-way conditions, taking due account of damping, a computer is clearly essential.

Where main machinery is mounted, it is necessary to examine the movements of the machinery under the considerable excitation forces from the propeller. This so called "blade rate" excitation inevitably coincides at some speed with one or other of the coupled natural frequencies of the mounted system. In such cases it becomes necessary to analyse the response of the complete shafting system and the escalation in the number of equations to be solved is considerable. Both axial and torsional excitation have to be considered and may indeed require solution simultaneously due to the coupling between axial and torsional response. A sample case, considering only a single main machinery mounted complex will require the solution of about 25 simultaneous differential equations and solutions, to give amplitudes of movement at selected points, have to be provided for a wide range of frequencies because of the relatively rapid variation of response with frequency. The solution of this problem requires at least one hour's running time of an I.C.T. 1903 computer.

Other Applications

Only a brief reference can be made to a number of other problems for which digital computer solutions prove useful and economical such as:

Boiler Design

Programmes are available to carry out the heat transfer calculations for projected boiler designs.

Heat Balances for Steam Cycles

A number of programmes are available to permit rapid and accurate heat balances to be calculated. The programmes cater for:

a) turbo-driven auxiliaries (condensing and back pressure);
b) motor-driven auxiliaries;
c) engine-driven auxiliaries;
d) regenerative feed heating up to six stages;
e) steam and gas combustion air heating;
f) re-heat cycles.

Some of the programmes are arranged for full power optimization of the cycle and others are arranged for part load heat balances. The programme print outs provide all the data required to permit the flow diagram to be drawn.

Gearing Design

Programmes are available to produce project gearing designs conforming to specified criteria. Inverse routines are also available which, when supplied with the leading particulars of a maker's design which has been submitted for evaluation, calculate the criteria on which the design has been based for comparison with the specified criteria.

Apart from the gear element design covered above an important aspect of the evaluation of makers' proposed designs prior to manufacture, or after some trouble has arisen in service, is the determination of magnitude and direction of the forces on gearing journal bearings and hence the selection of optimum oil inlet and thermocouple positions.

Bearing Performance

Programmes are available to estimate lubricating oil flow, power loss, oil outlet temperature, operative viscosity and oil flow thickness for journal bearings. In addition, the programme is capable of providing similar information for thrust bearings and estimating gearing sprayer oil quantities.

Scoop Design and Performance

The basic design method of Hewins and Reilly has been programmed. In merchant work where the scoop position may be very far aft, it is necessary to be able to verify whether scoop performance will be affected by separation of the boundary layer. A technique has been developed whereby the velocity gradient at the proposed scoop position is determined by an analogue method. This information enables the pair of simultaneous differential equations for evaluation of the "flow separation" criteria to be solved using a digital programme.

Condenser and Heat Exchanger Performance

A programme is available for the calculation of physical particulars and also for the derivation of heat transfer criteria from the physical particulars of a given heat exchanger.

THE ANALOGUE COMPUTER

General

Industrial research, development and design has demanded methods of solving complex problems practically since the industrial revolution. However, from the early 1930s the demand increased rapidly as the aircraft industry developed. The demands for non-destructive testing of airframes, engines, controls and pilots produced numerous analytical methods to obtain transient stress loads, vibration characteristics, temperature loads, and control stability, etc. before an expensive machine was sent forth to brave the elements. Development of computers provided the means for one or a small team of scientists and engineers to solve complex problems in a realistic time scale.

The ability of simple electronic circuits to provide a method of summation, integration and multiplication by constant factors continuously, which is directly analogous to how variables behave in a physical system, provided the early analogues used to represent the dynamic behaviour of a system. The behaviour of primary parts of a system can be deduced from basic scientific laws and these can in turn be re-written as mathematical equations. The substitution of electrical analogues for each of the operations or integrations provides the basis for an electronic analogue computer.

How a Problem is Solved

Fig. 2 shows the method of solving a problem using an analogue computer. From the physical system being studied the problem is defined—this may be a verbal description of an incident which has arisen during a test or a design problem requiring analysis before the design is finalized.

The problem statement is then formulated mathematically as a series of algebraic and differential equations. These equations are then scaled in time and in magnitude.

The time scaling is the means whereby the computer output may be run exactly on the same time scale as physical events in a physical system or at any multiple or fraction of real time. The equations are amplitude-scaled by assigning values to each of the variables slightly larger than they can possibly reach so that the computer output is obtained directly as a fraction of the chosen maximum value. This process is directly analogous to choosing the scale when plotting test results on a graph.

From the scaled equations the programming information for the computer is derived. This comprises three sections,

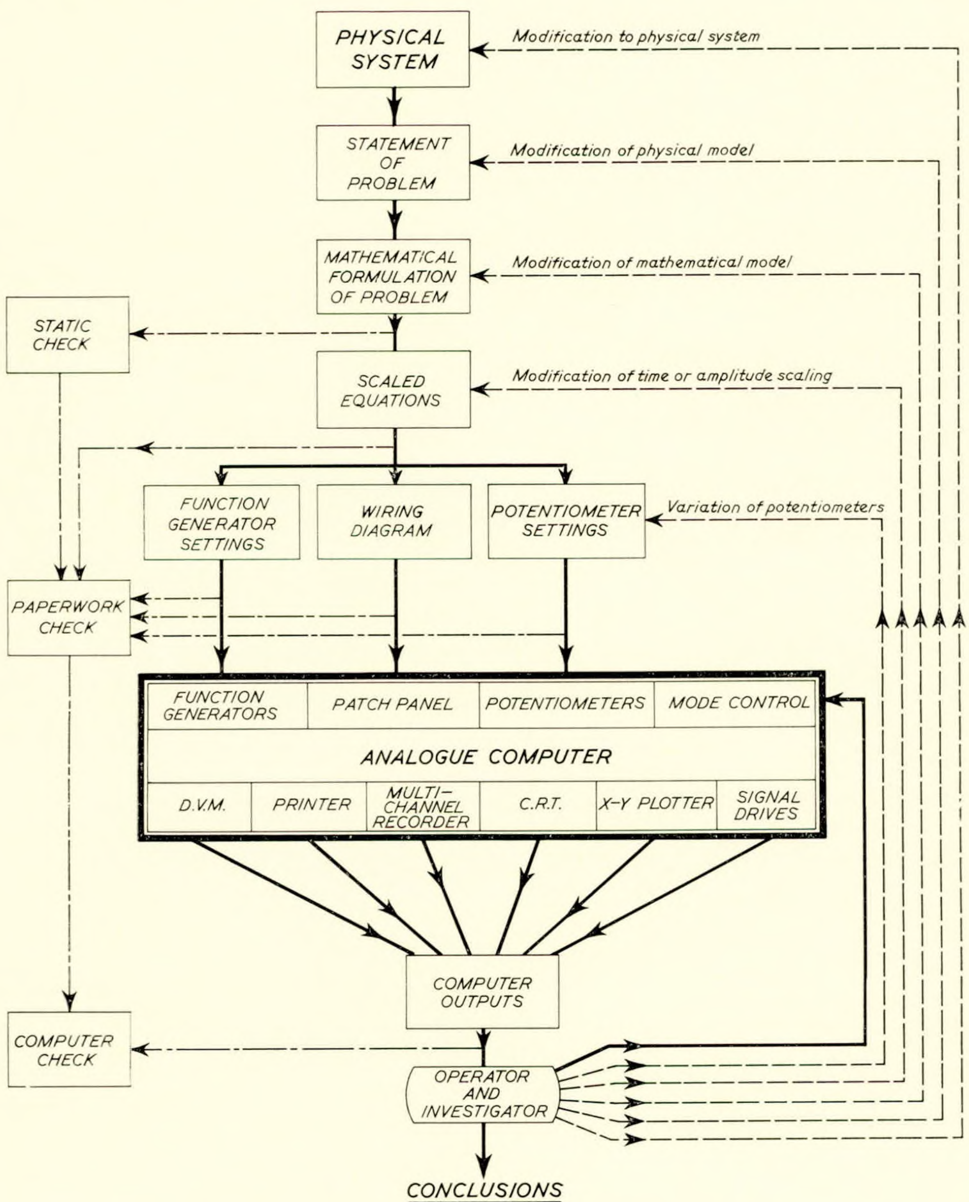

FIG. 2—*Operation diagram for an analogue computer*

potentiometer settings, a patch panel wiring diagram and function generator settings which correspond to data for the specific problem, and are the constant coefficients of the scaled equations. The patch panel wiring diagram provides the means whereby the computational blocks in the computer are interconnected to form a control programme. The function generator settings correspond to specific characteristics of the physical system such as heat transfer rates as functions of liquid flow, or valve discharge quantities as functions of valve lift.

For example, a simplified equation used for ship manoeuvring might be:

$$\frac{dN}{dt} = \frac{Q_E - Q_p}{K}$$

Extreme values are assessed as follows:

$$N_{max} = 5 \text{ rev/s}$$
$$Q_{Emax} = 10^6 \text{ lb/ft}$$
$$Q_{Pmax} = 10^6 \text{ lb/ft}$$
and K is 5×10^5 lb/ft s^2

Scaled values are denoted as $(N) = \frac{N}{N_{max}}$ etc. Thus amplitude scaling gives:

$$\frac{d(N)}{dt} = \frac{5\,(5 \times 10^5)}{10^6}\,[(Q_E)-(Q_P)]$$

$$= 0{\cdot}400\,(Q_E) - {\cdot}400\,(Q_P)$$

For time scaling to speed up the simulation the computer time τ is made equal to 1/10 of real time t, hence dt becomes 10τ and $\frac{d(N)}{d\tau} = 4{\cdot}00\,(Q_E) - 4{\cdot}00(Q_P)$.

The computer diagram is shown below, including an initial condition for N of 3 rev/s when $t = 0$, and remembering that the integrator reverses the sign.

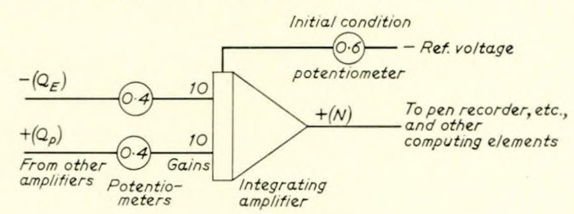

FIG. 3

Before going to the computer with the produced documentation a check routine is implemented. This is known as a combined static check and paper work check. Values are assumed for the variables which are expressed as differentials and from the remaining algebraic equations values are worked out for the other variables. The purpose of this is to ensure that no mistakes have been made in the implementation of the paper work following the mathematical formulation of the problem.

When the potentiometers have been set, the function generators set up, the patch panel wired and inserted into the computer, the computer outputs units are checked using the static check. This ensures that no errors have been made in setting up or in wiring the patch panel, and that the components of the computer are operational. This means the problem formulated in the mathematical equations is being solved by the computer.

The available outputs from the computer are either cross-records of variables (X-Y plots), simultaneous multigraphical recordings of variables against time, cathode ray tube outputs, digital values, or output drive signals which may be used to motivate simulated instrument panels or hardware.

Problems most suitable for Analogue Computation

The vast majority of problems solved using analogue computers are dynamic with physical time as the independent variable. Steady state parameters are always of interest to designers; however, if these are the primary or only interest, then it would be more economic to solve them using numerical methods. It is the insight to transient phenomena obtained by using an analogue computer that is invaluable, as the graphical outputs appear exactly as they would from recording instruments on a machinery test.

In analogue computation there is virtually no interface between the engineer or scientist who has the problem and the computer output. Consequently, it is primarily important that the designer himself uses or stands by the machine to gain the full insight available.

The analogue computer is prominently used by control engineering groups as a necessary stage in the design and evolution of control systems. Control theory is now many years in advance of its application, particularly so in the marine field and the advancement in applications of new techniques arises as the systems to be controlled are better understood.

The following examples are typical analogue problems and have been chosen to illustrate where intangible pay-offs have been realized from the investigations.

Complete Propulsion Plant Dynamics

Two studies have been carried out to evaluate the overall dynamic performance of the steam propulsion plant of guided missile destroyers. The initial stage in carrying out the first study was to divide the plant into its major sub-systems such as boiler pressure control, combustion control, boiler level control, superheater steam outlet temperature control, de-aerator control, auxiliary exhaust range pressure control, condenser vacuum, condenser level control, steam turbine performance and propeller and ship response.

The mathematical model derived to represent the plant and its control comprised eighty simultaneous equations providing a continuous simultaneous output of eighty variables. The controller gains and set up data were used from a plant which was at sea and the results obtained from the simulation for various changes in steam turbine throttle position were compared with trials results for the same system inputs. The simulation results agreed remarkably well with the trial records, indeed in many cases the validity of trial records was checked from the simulation records as the measuring transducers on the trial were imposing their own characteristics in some of the very rapidly changing variables.

The second overall study was carried out on a somewhat similar system before design completion of the propulsion plant and consequently had a slightly different overall objective from the first study. The data obtained from the simulation are to be used for system design purposes and to provide a handbook for setting up, testing, tuning and investigating fault conditions arising from credible system failures. For example, it is possible to study accident conditions and how to overcome them on the computer where conducting similar trials on a plant would be catastrophic. Results from the first study clearly indicated violent cross coupling between some systems and conversely showed where the total propulsion plant could be split into three major sections without errors arising from changing boundary conditions. The splitting of the overall study into parts allowed greater detail to be included in the models for a given computer capacity. The first study required over two hundred amplifiers. The second study was carried out on a single console in three separate parts overall, each section being thus manageable. The first study is described in detail in a paper given in 1966[2].

Dynamic Performance and Control of an Auxiliary Exhaust Range

One item shown to be difficult to control in the first simulation, and which confirmed this during the trials was the auxiliary exhaust range pressure. The pressure control system comprises a sensor, the output of which is compared to a fixed set point in a two-term controller, the output driving a split range valve positioner stroking a steam make-up and a steam rejection valve. The make up and rejection valves were sized to cope with the steady state demands of the system over the whole of the power range. The dynamic behaviour of the system was investigated closely during the second simulation and the cause of the lack of control established. The major input of steam to the exhaust range is from the turbo driven forced draught blowers, the major offtake being the steam to the de-aerator which is determined by the feed flow to the boiler. The time response rate of the blowers is much higher than the feed flow response rate, consequently a considerable difference between steam input and output results during rapid manœuvring. The problem can be severely accentuated by the level programme used for the boiler drum water level. A small programmed level change produced large variations in feed flow demand with consequent fluctuations in steam offtake. A large programmed level change produced a much smaller transient demand on the feed system. Overall, the problem was overcome by re-sizing the valves and carefully matching the controller settings on the boiler water level and exhaust range pressure controls.

Condenser Level Control

The transient demands of a feed system are accentuated where the attitude of the ship may vary transiently to a considerable amount. A primary requirement for a main feed pump which is not designed to cavitate is that the inlet pressure to the pump be maintained above its nett positive suction head value. The condensate extraction pump supplies the main feed pump and under large angles of heel and trim that pump may cavitate. Consequently, in order to ensure that the input pressure is maintained, a surge tank is included in the system. The system studied is shown in Fig. 4.

Level control is achieved by taking a signal from the sump level and using a proportional controller to recirculate the extraction pump flow to the sump if the level is low, or feeding the main feed line when the level is high.

The main objects of simulating the dynamic characteristics of this extraction system were to determine the required surge tank volume, assess the time during which cavitation would be taking place in the extraction pump, determine the discharge pressure of the extraction pump during cavitation, and determine the cooling water flow to the condenser air ejectors and glands vapour condenser supplied from the extraction pump discharge.

Fig. 4 also shows the transient results obtained from the simulation for a steady power level with the system inclined to 40° causing the extraction pump to cavitate.

The simulation of the extraction pump during cavitation is of interest. If the inlet head is above the nett positive suction head corresponding to the flow through the pump at that instant, the pump discharge flow and head are determined by the normal operating characteristic of the pump. If the inlet head falls below the nett positive suction head value the inlet head determines the pump flow and the discharge head of the pump is determined by the system resistance characteristic.

Continuous computation of the pump discharge characteristics was obtained using implicit function techniques on the analogue computer. This technique provides the means of continuously up-dating the solution of simultaneous equations to match the discharge conditions of the pump to the system.

The required analogue computer capacity to do this study was small, being well within the capability of two twenty-amplifier computers. The input data on the sump characteristics of level as functions of total water quantity in the sump were obtained from model tests.

Diesel Electric Propulsion Units

A Diesel electric propulsion unit has been studied. The study was used to assess the value of using resistance braking on the electric induction/synchronous propulsion motor so as to increase the deceleration rate of the ship particularly during crash manœuvres from high speed, to evaluate the basic multi-Diesel governing requirements and to assess the overall control sequence timing to ensure safe operation of the unit.

The effect on stopping a ship using resistance braking is referred to in the section of the paper dealing with ship manœuvring. The type of transient record obtained from the analogue simulation for start-up of the propulsion motor is of interest. Typical results for the fluctuating load torque and

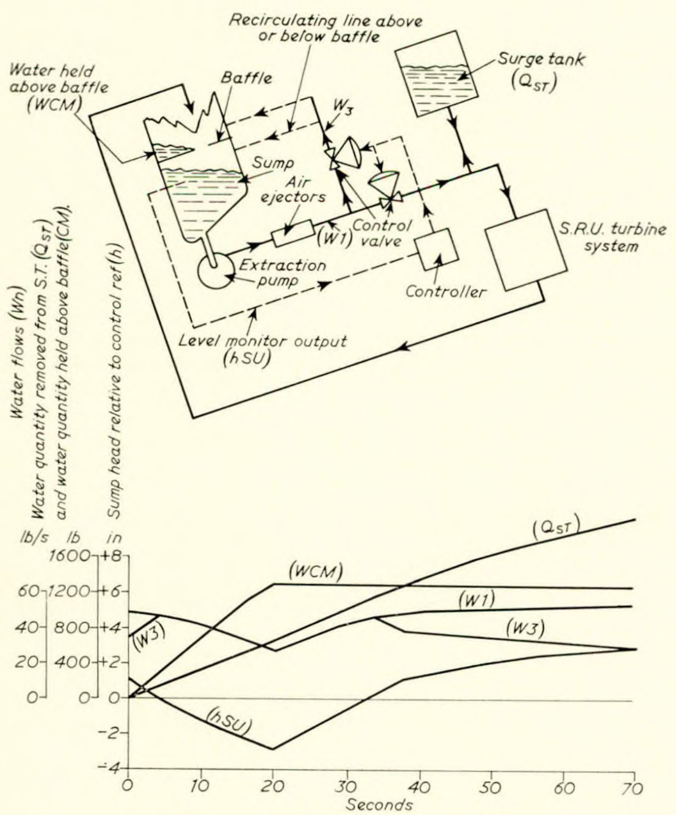

FIG. 4—*Transient performance of condensate level control system*

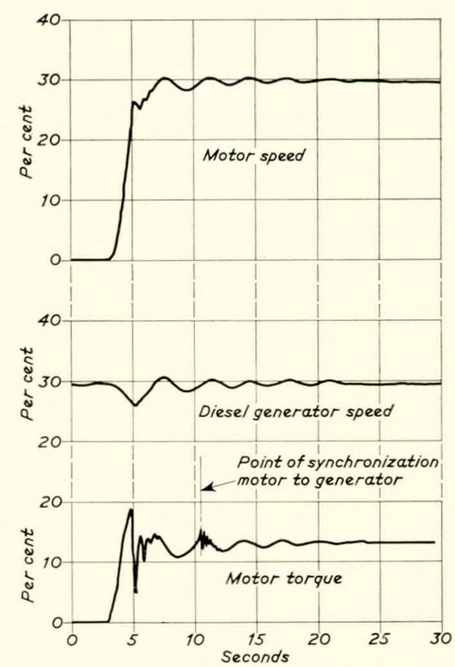

FIG. 5—*Transient motor and generator response following motor start*

motor speed and Diesel speed are shown in Fig. 5. On connexion of the motor to the generator, the motor torque builds up as functions of the motor field excitation, the slip frequency between the motor speed and the equivalent Diesel generators speed and the absolute frequency of the Diesel generator supply. These factors are evidently very interdependent.

The load torque builds up as the excitation increases and the slip between the supply and motor frequency reduces. When the slip reduces to approximately seven per cent the torque-slip characteristic for the motor running as an induction motor changes and rapidly decreases as the slip decreases, causing the large oscillatory torque values.

When the slip frequency is reduced to less than three per cent, the motor is synchronized to the supply. The second high frequency torque oscillation clearly indicates the point of synchronization.

Variable control parameters investigated during the study included: the rate of increase in excitation; the heat generated in the motor as a result of reducing the excitation and consequently allowing the motor to run at high slip frequency for a considerable time, before synchronization can be achieved; the design requirements and necessary inter-relations between the Diesel governor and the excitation control rate; and in addition the ship stopping investigation.

The whole study included simulation of the Diesel engines, the generator, motor, propeller, forward motion of the hull, together with the associated operational control system. The simulation was fairly large requiring approximately a 140-amplifier computer.

Non-dynamic Investigations

One such problem was the analysis of the tension on a trawl winch, and the design of a trawl wire to cope with the tension load, where a sampling drag was being towed to depths of up to 8000 m.

Typical questions arising are:

1) Could the winch load be reduced by increasing the trawl speed to recover the rope?
2) How much rope must be paid out to attain a given trawl depth at a given trawl speed?
3) What is the shape taken up by the cable?

Typical results obtained from the analogue computer analysis are given in Fig. 6.

The problem was analysed by starting from the end of the cable to which the drag is attached. The cable was taken as being flexible, so the angle taken up by the cable at any point on it must be determined by the nett summation of horizontal and vertical face components caused by hydrodynamic drag and rope weight respectively. An assumed diameter/length previously determined function for the diameter can be used. Where the diameter required was continuously determined the rope tension is maintained as a fixed function of the rope diameter and material properties.

A prime mathematical analysis of such a problem is given by Kullenbörg[3]. It is interesting to note the complexity of the solution obtained. The problem was solved on one small twenty-amplifier analogue computer within two hours of starting to patch the problem on the machine.

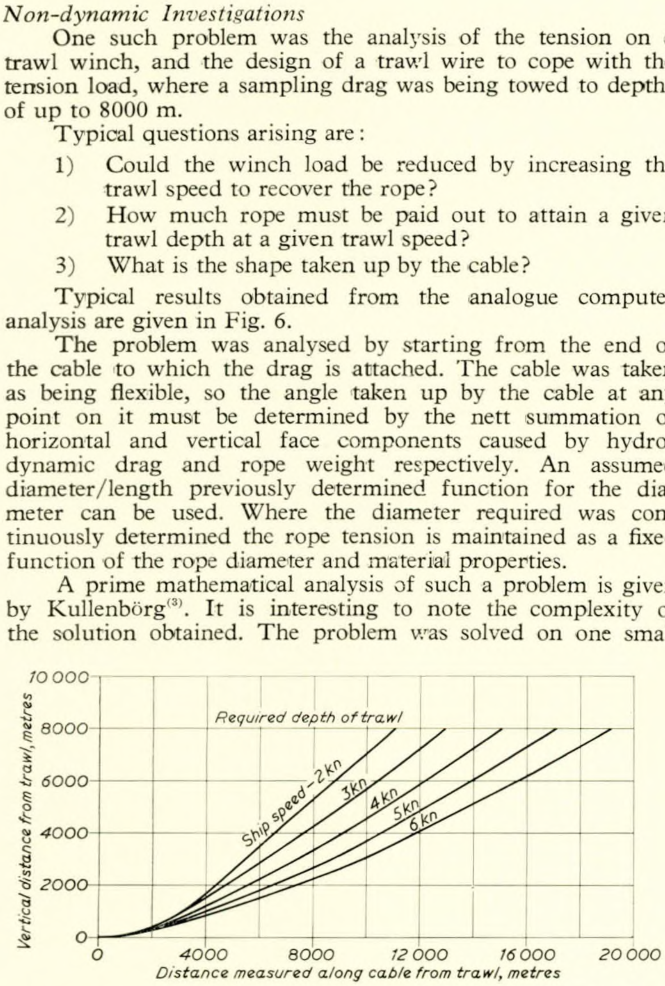

FIG. 6—*Required trawl cable lengths for ranges of ship speed and trawling depths*

Overall Analogue Computer Usage and the Way Ahead

Of late the number, size and complexity of the problems posed to the analogue computer have increased as the value of the techniques is realized, as dynamic performance demands increase, and as the amount of automatic control has increased.

Current large scale investigations now in progress include, evaluation of mechanical shock transmitted to ships' hulls and machinery items; design of passive roll stabilizing units for trawlers' transient performance of various propulsion units including Diesel and gas turbine units driving controllable pitch propellers.

The requirement for insight to the physical problems which are initially ill-defined, the ability to produce continuous output of several variables and the ease and immediate direct access to the problem solution appeal to the design engineer. Each problem is assessed to determine whether the solution could be most satisfactorily reached using the analogue or the digital computers. In many cases when the basic insight to a problem has been gained and the best mathematical model established using the analogue computer, it is feasible to transfer the problem to the digital computer for further runs of results for a well defined problem.

To date, one of the most serious deficiencies in analogue computation has been the amount of time to set the problem up assuming that the patch panel has been de-patched, etc. This has been particularly shown when the problem has to be patched on-line for each problem solved.

The way ahead is clear. When the analogue computer facilities become overloaded the general machine requirements to meet the increased work-load demands are:

a) the machine must be safe for operation on an "open-shop" system by engineers who are not computer specialists;
b) storage of programmes must be available;
c) rapid transfer from one problem to another must be possible without tedious intervention of massive manual checking;
d) the output from the machine must be as clear, extensive, accurate and selectable as possible.

Many problems lie on the boundary where neither analogue nor digital computer solutions are readily available, and a combination of the two to produce a new scientific problem solving machine has created the "hybrid" computer. The limitations of the analogue computer are: fixed precision i.e. no floating point, no memory; poor arithmetic capability; these are the strong points of the digital computer. The weakest point of the digital computer operation is the difficulty of performing continuous integration—the analogue computer's greatest attribute. The new generation of hybrid computers combines the previous attributes of both machines.

SHIP MANŒUVRING COMPUTATIONS

Although a large number of well-instrumented trials have been carried out in various ships to test their manœuvring characteristics, full analysis of the results was not possible without computers. Some of the reported trials[4] are given in the list of references but the authors have probably accumulated useful ideas from many other papers. Good empirical rules were developed and used to define the requirements for astern power, before the general use of computers, but without full analysis certain areas remained largely a matter of opinion.

This is particularly true of the controversy which exists as to whether a ship can be stopped more rapidly by applying all the astern power available as quickly as possible, or by holding the propeller at an optimum low speed or even stopped for a period before applying the full astern power. In all cases examined by the authors, the ship is stopped more quickly by applying the maximum astern power available as soon as possible and it is hoped to demonstrate this clearly.

Ship manœuvring calculations can be performed on either analogue or digital computers, the former being preferable when exploring the potential of a relatively novel propulsion means and the latter when accuracy is required in analysing existing well defined systems in detail. A hybrid computer would probably give the best of both worlds for these problems. The results given have been computed on a digital computer but the accuracy, as always, depends on the validity of the assumptions and engine and propeller characteristics used. By far the greatest part of the task lies in accumulating the correct data. Even so, some of the assumptions may be challengeable.

The Basic Equations, Assumptions and Data

The ship motion is defined by two differential equations:

$$\text{Rate of change of ship speed } \frac{dV}{dt} = \frac{T_P(1 - td) - R}{M} \qquad (1)$$

Where, V = forward speed of the ship
T_P = forward thrust on the propellers
td = thrust deduction
R = resistance to the ship's motion
M = mass of the ship and its entrained water

$$\text{Rate of change of shaft speed } \frac{dn}{dt} = \frac{Q_E - Q_P - Q_F}{2\pi\ I} \qquad (2)$$

Where, n = shaft speed
Q_E = torque produced by the engine
Q_P = torque at the propeller
Q_F = friction torque in the transmission, and changes sign when n changes sign
I = moment of inertia of engine, transmission and propeller (including entrained water); when these have to be dealt with separately, initials are used as suffixes.

T_P and Q_P are found from the propeller characteristics and are functions of Va, n, P and D where Va is the speed of advance of the propeller taking into account the wake factor, P is the propeller pitch and D the propeller diameter.

The assumptions made for thrust deductions are:

For n positive and P/D positive $\quad td = 0{\cdot}1$
For n zero or P/D zero $\quad td = 0{\cdot}0$
For n negative or P/D *negative* $\quad td = 0{\cdot}25$

$$\frac{nP}{n \text{ full astern } P \text{ full astern}}$$

When the propellers are rotating astern or there is negative pitch, water is thrown forward, the rate of change of momentum governing the astern thrust. As this water strikes the hull its forward momentum is reduced and a thrust is generated that reduces the effective water thrust. When n or P/D is zero this effect is probably absent, whereas under the full astern conditions a thrust reduction factor of 0·25 seems to fit the trial results, and a linear change has been assumed between these two conditions.

The ship's resistance R is a function of ship's speed and is found from the resistance/ship's-speed curve. When going astern, the astern resistance is assumed to be 1·3 times the ahead resistance at the same speed ahead.

M includes the total mass of the ship + eight per cent of this for entrained water. Hooft and Van Manen[5] suggest that this percentage should vary with ship's and shaft speed at very low speeds but this variation has been ignored here.

For a Steam Turbine

Q_E the ahead engine torque is taken as Q_E maximum ahead times

$$\left(\frac{\text{percentage steam}}{100 \text{ per cent}}\right)\left(2{\cdot}0 - \frac{n \text{ ahead}}{n \text{ maximum ahead}}\right) \qquad (3)$$

i.e. the turbine stalled torque is twice the design torque at full ahead speed.

Q_E the astern engine torque is taken as Q_E maximum astern times

$$\left(\frac{\text{percentage steam}}{100 \text{ per cent}}\right)\left(1{\cdot}5 - \frac{n \text{ astern}}{2n \text{ maximum astern}}\right) \qquad (4)$$

A torque of this magnitude was measured during the shore trials of Y.E.A.D. 1 turbines when the shaft was held stationary.

The much reduced number of rows of blades in the astern turbine accounts for the stationary factor of 1·5 in lieu of 2·0.

Q_F the friction torque is taken as four per cent of Q_E at full power ahead and to vary as the square of the shaft speed; this probably exaggerates the dependence on speed but friction torque is relatively unimportant. When the shaft is stopped the friction torque is very much greater due to stiction, but, since zero speed is passed through rapidly while oil films still exist, the stalled condition does not arise in the examples given in this paper, so that stiction can be ignored.

The total moment of inertia, I, includes an addition of 25 per cent to the propeller inertia for entrained water. Inertias and torques are all referred to the propeller shaft speed.

Propeller Characteristics

The values of propeller thrust and propeller torque are given by the equations:

thrust $T = K_T \rho D^2 Va^2$, torque $Q = K_Q \rho D^3 Va^2$

For manœuvring calculations, it is necessary to have values of K_T and K_Q both for J and I/J where J is $\frac{Va}{nD}$ since values are required for conditions where the ship is stopped, i.e. Va is zero and when the shaft is stopped. At this point nD is zero, so both J and I/J reach infinity during a manœuvre. Stephanson[6] has shown how to overcome this difficulty by combining the curves for J and I/J in single curves against a value of J^1 where $J^1 = \dfrac{Va}{\sqrt{(Va^2 + n^2 D^2)}}$ and

the thrust coefficient $K^1T = \dfrac{T_p}{\rho D^2(Va^2 + n^2D^2)}$

the torque coefficient $K^1Q = \dfrac{Q_P}{\rho D^3(Va^2 + n^2D^2)}$

Va the speed of advance $= V\,(1 - \text{wake factor})$

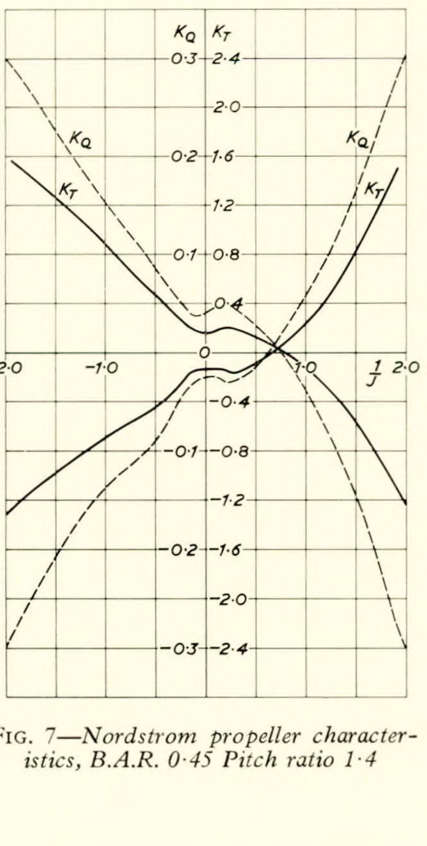

FIG. 7—*Nordstrom propeller characteristics, B.A.R. 0·45 Pitch ratio 1·4*

Fig. 7 shows K_T and K_Q plotted against I/J from Nordstrom curves for a four-bladed propeller with blade area ratio 0·45. P/D = 1·4.

Fig. 8 shows K^1_T and K^1_Q plotted against J^1 for Nordstrom four-bladed blade area ratio 0·45 P/D = 1·4.

Fig. 9 shows K^1_T and K^1_Q plotted against J^1 derived from trial results carried out in H.M.S. *Savage* where the three-bladed propeller had a blade area ratio of 0·75.

Interpolations between Figs. 8 and 9 have been used in this paper for the values of K^1_T for propellers of other blade area ratios, and similarly for $K_Q{}^1$.

In the K_T curve in Fig. 7 it can be seen that the negative value of K_T reaches a maximum when I/J = about 0·2. It then reaches a minimum when I/J = about zero before rising again for negative values of I/J which occur when the propeller shaft is rotating astern and the ship is still moving ahead. It is this turning back of the K_T curve at I/J = about 0·2 which led to the contention that the most effective way to stop a ship is to reduce the shaft speed but to keep it running ahead so that $\frac{nD}{Va}$ remains at about the value of 0·2. The fallacy here, lies in the fact that the thrust depends not only on this coefficient but also on the value of $\rho D^2 Va^2$ and Va is a diminishing quantity (illustrated later in Fig. 15).

For controllable pitch propellers, families of curves of the type shown in Fig. 8 and Fig. 9 are used over the range of pitch diameter ratios. The values of K^1_T when P/D is negative tend to be less than they would be for a fixed pitch propeller of the same P/D ratio, because, when turned into the astern running position, the reverse pitch near the boss is much less than that near the tip. The reduction of K^1_T due to this cause may be as much as 40 per cent.

Comparison of Trial Results with Computed Figures

In 1950-53, extensive propeller trials were carried out by the Admiralty in the destroyer H.M.S. *Savage*; one of the objects was to discover what transient thrusts and torques

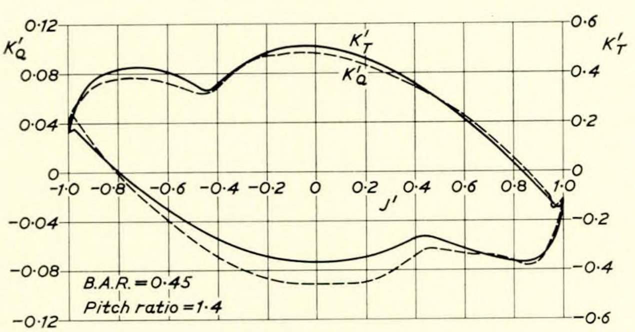

FIG. 8—*Nordstrom propeller characteristics*

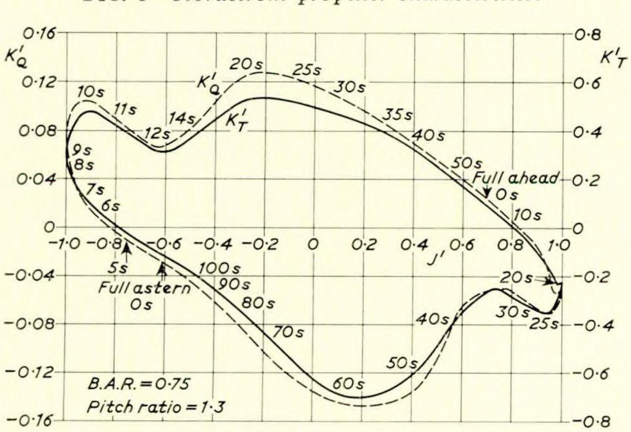

FIG. 9—*H.M.S.* Savage—*Propeller characteristics—B.A.R. = 0·75; pitch ratio = 1·3*

TABLE I

Ship	Displacement tons	Shp shaft	No. of shafts	rev/min	Propeller diameter ft.	Pitch ft.	P/D	Blade area ratio	Speed knots	Wake factor	Fixed propeller thrust tons	Astern shaft speed rev/min	Prime mover
H.M.S. *Savage*	2370	20 000	2	312	10·5	13·6	1·293	0·75	31·3	0·1	107	—216	—
Large fast ship										0·095	532	— 68	—
H.M.S. *Redpole*	1590	2150	2	300	7·5	8·5	1·133	0·342	18·4	0·1	30·0	per cent —214	—
s.v. *British Bombardier*	66 200	15 000	1	105	22	17·2	0·78	0·61	15·86	0·35	115	—100	—
Frigate Type:													
Reversible engine	1536	15 000	1	220	12	15·6	1·3	0·75	27·8	0·1	55	—157	5630
Reversible gears	1536	15 000	1	220	12	15·6	1·3	0·75	27·8	0·1	55	—157	5630
C.P. propeller	1536	15 000	1	220	12	16	1·33	0·698	27·8	0·1	55	—	5630
Electric drive	1536	15 000	1	216	12	15·6	1·3	0·75	27·8	0·1	55	—	5630
Merchant ship (Sulzer 6RD 76 F.P.P.)	22 300	7100	1	104	18·5	14·1	0·76	0·545	14·75	0·33	54	—104	104
Merchant ship (2 Ruston 8 AO's C.P.P.)	22 300	6420	1	82	21	18·7	0·89	0·45	14·75	0·33	54	— 82	450
Merchant ship (2 Ruston 8 AO's Clutch)	22 300	6420	1	82	21	18·3	0·87	0·45	14·75	0·33	54	— 82	450

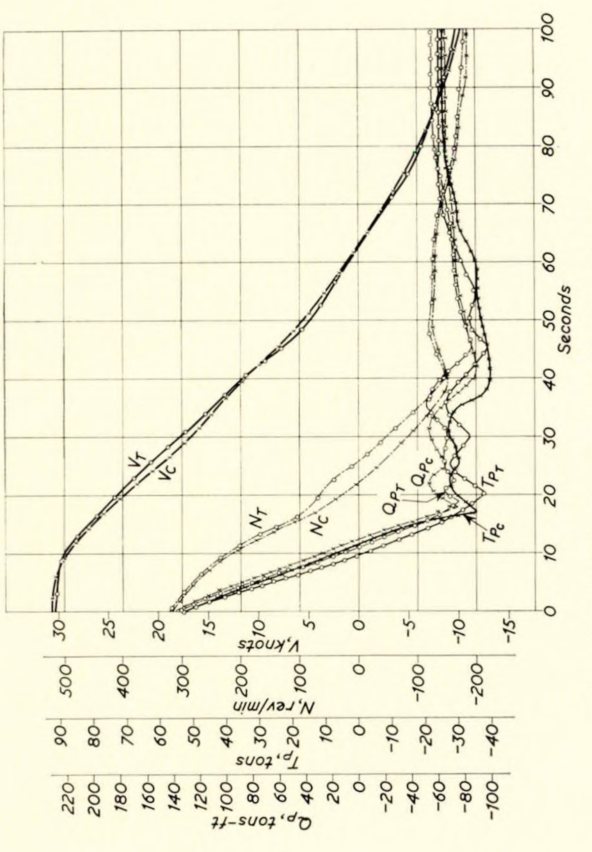

FIG. 10—*H.M.S.* Savage—*Full ahead to full astern*

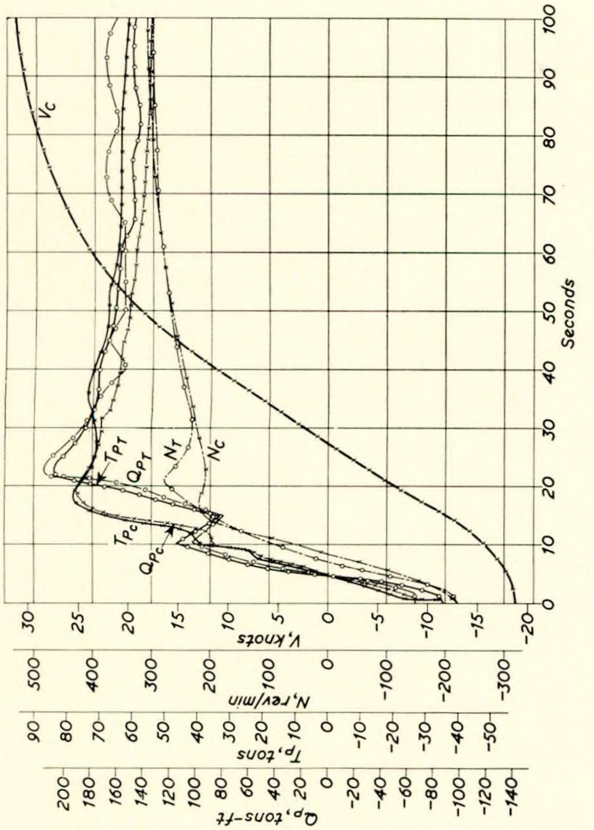

FIG. 11—*H.M.S.* Savage—*Full astern to full ahead*

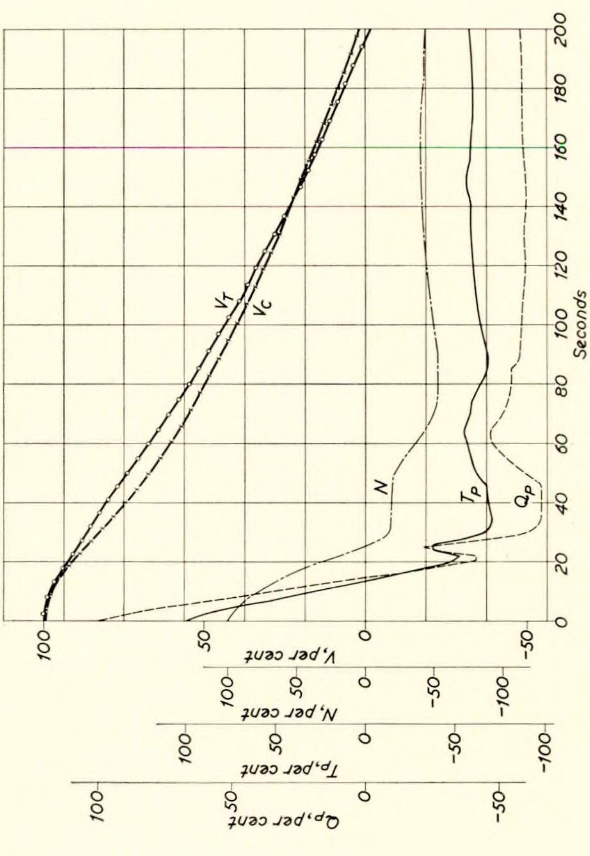

FIG. 12—*Large fast ship—Full ahead to full astern*

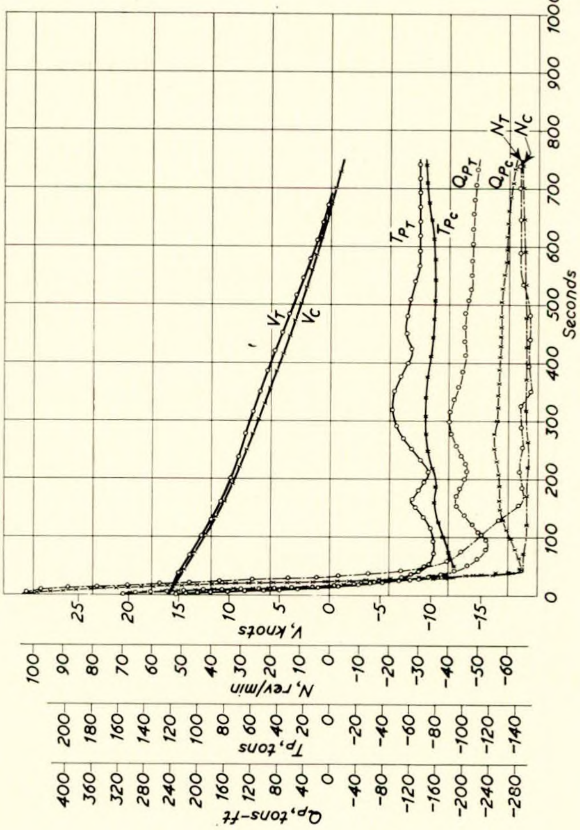

FIG. 13—British Bombadier—*Full ahead to full astern*

TABLE II

Ship	Trial or computed	Initial speed, knots	Astern power		Ship stopping time in secs.	Ship stopping distance in ft.	Shaft stopping or zero pitch time in secs.	Maximum astern torque: maximum ahead torque, per cent	Maximum astern thrust: maximum ahead thrust, per cent	Maximum astern displacement Shp/tons	4/5 Initial speed, knots × stopping time seconds/ stopping distance, ft.	Fig.
			Shp	Percentage of full ahead								
H.M.S. *Savage*	T	31·3	13 300	33	62·5	1740	26	63	73	5·6	0·90	9
H.M.S. *Savage*	C	31·3	13 300	33	62·5	1720	22	63	74	5·6	0·92	9
H.M.S. *Savage*	T	18·9	−40 000	100	—	—	7·5	157	155	(16·8)	—	10
H.M.S. *Savage*	C	18·9	−40 000	100	27·5	535	8·5	139	142	(16·8)	—	10
Large Fast Ship	T	100%	—	30	195	4940				1·06	—	11
Large Fast Ship	C	100%		30	205	4720	25	66	70	1·06	—	11
Large Fast Ship	T	38%	—	12	199	1890				0·43	—	—
Large Fast Ship	C	38%		12	200	1950	12	28	32	0·43	—	—
H.M.S. *Redpole*	T	18·4	1700	40	112	1690				1·07	0·98	—
H.M.S. *Redpole*	C	18·4	1700	40	110	1540	26	63	51	1·07	1·05	—
s.v. *British Bombardier*	T	15·9	10 000	67	690	8300	35	76	68	0·15	1·06	12
s.v. *British Bombardier*	C	15·9	10 000	67	675	7630	25	93	84	0·15	1·12	12
Steam Frigate:												
Reversible turb.	C	27·8	5000	33	70	1610	21	63	70	3·25	0·97	13
Reversible gears	C	27·8	5000	33	70	1620	17·5	66	78	3·25	0·96	14
C.P. Propeller	C	27·8	10 500	70	60	1360	20	109	96	6·8	0·98	15
Electric drive	C	27·8	10 500	70	59	1610	23	109	120	6·8	0·83	16
Diesel merchant ship:												
Direct reversing	C	14·75	7100	100	395	4880	86	98	95	0·31	0·96	17
C.P. propeller	C	14·75	6420	100	394	4295	10	90	121	0·29	1·08	18
With friction clutches	C	14·75	6420	100	340	3843	11	107	125	0·29	1·03	19

occurred during rapid manœuvres. Some results from these trials were published in 1953[7] but without the assistance of computers, it was not found possible at that time to relate these results to the propeller characteristics. Relevant propulsion particulars of H.M.S. *Savage* are shown in Table I.

These data have recently been used in a Fortran programme on an ICT 1903 computer, using the Runge-Kutta-Gill method of integration to determine on a time base for various manœuvres the following:

i) ship's speed;
ii) shaft rev/min;
iii) thrust;
iv) torque.

Comparisons have been made with a number of trial records and Fig. 9 shows the computed and the trial results for a manœuvre full speed ahead to full power astern, the throttles being operated simultaneously, the opening and closing time for both the ahead and astern throttles being 16 s.

Fig. 10 shows the results for a manœuvre full speed astern to full power ahead the throttles being operated simultaneously, but taking 12 s.

Not unnaturally, the agreeement between the theoretical and practical results is good because the K^1_T and K^1_Q values in Fig. 8 were partly derived from these trial results. However, values of K^1_T and K^1_Q interpolated from Fig. 8 and Fig. 9 give good results for stopping distance and time when compared with trial results in a number of other ships, namely a large fast ship, the sloop H.M.S. *Redpole,* and the bulk carrier *British Bombardier,* the propulsion particulars of which are also shown in Table I. Table II gives computed and trial results.

Large Fast Ship

Measurements of ship speed against time were taken during the trials but no transient values of torque and thrust. Fig. 11 shows the computed and trial results for a full astern manœuvre from full speed and in Table II figures for this and for a half power astern manœuvre from a lower speed are given. The stopping times were respectively 195 and 199 seconds in the trials and 205 and 200 seconds computed; stopping distances were 4940 ft and 1890 ft in the trials and 4720 ft and 1950 ft computed.

Computed values of the maximum transient thrusts and torques for the full astern manœuvring were 70 per cent and 66 per cent of the steady full ahead values. These values represent 160 per cent and 150 per cent of the design full astern values.

H.M.S. Redpole

Computed and trial stopping times for a full astern manœuvre from 18·4 knots ahead were 110 seconds and 112 seconds respectively, with corresponding stopping distances of 1540 and 1690 feet.

Computed values of the maximum transient thrusts and torques were 51 per cent and 63 per cent of the steady full astern values.

S.V. British Bombardier

The B.S.R.A. report[8] gives a full account of the stopping trials carried out in 1966 in s.v. *British Bombardier.*

Fig. 13 shows computed results compared with the trial results. There is good agreement in the speed time curves and fairly good agreement in the computed and trial results for thrust and shaft rev/min but the measured results for torque are generally lower than the computed figures. Examination of the records, particularly the trial shaft speed, suggests that astern steam was not applied as quickly as assumed in the computer programme and that this has reduced the maximum astern torque and thrust but this would not account for the later discrepancy in torques.

Interesting points were the tendency to yaw off a straight course when stopping and the corresponding reduction in distance travelled, and the abortive attempt to keep the propeller revolving slowly ahead, no doubt made to test the validity of the postulation that it was advantageous to stay on the I/J value of 0·2 (Fig. 7). In the event the resisting propeller torque (and no doubt the astern thrust) was so low at this point that the surplus engine torque very quickly reversed the shafts into the negative I/J region where the astern torque (and thrust) could build up to resist the turbine astern torque. This was an excellent demonstration of the fallacy of the postulation.

Computed and trial stopping times for a full astern manœuvre from 15·8 knots were 675 s and 690 s respectively, and stopping distances were 7630 ft and 8300 ft.

Comparison of Different Means of Reversing

The results of the comparison of computation and trial results are considered to be sufficiently good to warrant using the computer programme for making comparison of the effectiveness of different reversing means in a given ship. The reversing means studied were:

a) reversing engines;
b) reversing gears;
c) controllable pitch propeller;
d) electric drive.

For the study a single shaft steam-driven frigate was selected. As far as possible the same assumptions are made in each case for the performance of engines and propellers, except where practical considerations would make the assumptions ridiculous, for example, the astern turbine is not given an equal performance with the ahead turbine since this would obviously lead to an unacceptably large astern turbine. Similarly the reversing gears are designed for this acceptable amount of astern power, although there is sufficient steam available and reversing gears would be feasible, though costlier, for considerably more astern power.

The data common to (a), (b), (c) and (d) are included in Table I.

a) *Reversing Steam Turbine*

The astern turbine is assumed to have been designed to give one third of the full ahead shp i.e. 5000 shp at 157 rev/min giving a maximum astern speed of 15·4 knots. Time to shut ahead throttle and to open astern throttle simultaneously was taken as 20 s. Fig. 14 shows the complete results:

time to stop shaft	=	21 s
ship stopping time	=	70 s
stopping distance	=	1610 ft
maximum astern propeller torque	=	96 ton ft 63 per cent full ahead torque
maximum astern propeller thrust	=	37 tons 70 per cent full ahead thrust.

b) *Reversing Gears*

The reversing gear-box is assumed to contain arrangements for driving ahead through an inverted synchro-self-shifting clutch and, when it is required to manœuvre, for the drive to be transferred in the first place to an ahead fluid coupling. This can then be emptied while the astern fluid coupling which drives through another train of gears that includes an idler is being filled. The astern train is assumed to be designed to give 5000 shp at 157 rev/min, the fluid couplings running at 3000 rev/min.

The assumed operating cycle is:

1) reduce turbine steam to 33 per cent in ten seconds and maintain at this;
2) simultaneously operate clutch and transfer to ahead fluid coupling in ten seconds;
3) time to empty ahead fluid coupling and fill astern—ten seconds.

Additional equations and data curves are necessary to describe the actions of the fluid coupling. Equation (1) remains the same but equation (2) becomes:

$$\frac{dn}{dt} = \frac{Q_{CH} - Q_{CS} - Q_P - Q_{F2}}{2\pi\ I_2}$$

and an additional equation is required:

Rate of change of engine speed $\frac{dn_E}{dt} = \frac{Q_E - Q_{CH} - Q_{CS} - Q_{FI}}{2\pi\ I_1}$

Q_{CH} and Q_{CS}, the torques in the ahead and astern fluid couplings respectively, are functions of the amount of filling, the slip and the square of the input speed of the coupling, and torque characteristics ascertained by trial have to be fed into the computer. The maximum values of these torque characteristics tend to occur at 100 per cent slip and are about six times the torque at the same speed under normal running conditions with about 4 per cent slip. Q_{F1} and Q_{F2} are the friction torques up and downstream of the fluid couplings, and have each been taken as two per cent at full power.

Fig. 15 shows the computer results. The plotted values of Q_E, Q_{CH} and Q_{CS} shown in Fig. 15 are the torques in the coupling rather than those referred to the propeller shaft.

Time to stop shaft	=	17½ s
Ship stopping time	=	70 s
Ship stopping distance	=	1620 ft
Maximum astern propeller torque	=	103 ton ft, 66 per cent full ahead torque
Maximum astern propeller thrust	=	42 tons, 78 per cent full ahead thrust

The design of the fluid couplings has to be such that:

1) it can withstand the maximum transmitted torque;
2) it can withstand the maximum centripetal forces at its maximum running speed with the appropriate filling;
3) it will not overheat during reversal of the engines.

An estimate of the temperature rise in the oil flowing through the coupling can be obtained from the computer programme assessing the heat produced by the product of coupling torque and slip and apportioning the heat to the metal parts and to the oil flow characteristics under different slip conditions. The manœuvre shown in Fig. 15 indicated a minimum flow of 360 gal/min with an oil temperature rise of 100 deg F (59 deg C).

As soon as the propeller starts to revolve astern, useful work is done by the coupling and the amount of heat given to the oil rapidly reduces.

Also shown in Fig. 15 is the effect of holding the propeller stopped as soon as it reaches this condition. The astern thrust and torque curves resulting from this manœuvre quickly fall below those obtained if the propeller is made to revolve astern and there is a corresponding reduction in ship's deceleration. This fall-off is due to the falling ship's speed and torque and thrust fall off as the square of this speed.

c) Controllable Pitch Propeller

The equations used for determining the manœuvring performance with controllable pitch propellers are the same as those for the reversing turbine, but different K_T^1 and K_Q^1 curves have to be used as the pitch is changed and steam is always admitted to the ahead turbine so that equation (3) is used throughout. There is therefore greater astern torque available than with either the reversing turbine or the reversing gears both designed for an astern power of 5000 shp at 157 rev/min.

To obtain optimum performance from the controllable pitch propeller, it is necessary to determine satisfactory values for the time delay and rate of operation of the pitch changing mechanism and for the operation of the throttle. To prevent overspeeding of the shaft, the steam to the turbine must initially be decreased at the maximum rate, so that reduction of the pitch can safely be started as soon as possible. Assuming that throttle opening to the turbine can be reduced at a rate of seven per cent per second (approximately that assumed for the reversing engine), it was found that pitch reduction from the maximum ahead P/D to the maximum astern P/D in 27 seconds could commence five seconds after the steam commenced reducing. Without this delay overspeeding might occur.

The greatest tendency to overspeed occurs when the P/D ratio is slightly positive and the steam flow to the turbine must be kept very small until a negative P/D ratio is obtained; as the negative pitch builds up, the steam flow can be similarly built up. There is however a limit, as excessive torque would occur if full steam is applied to the turbine before the ship has built up way astern. Limiting the build up to 70 per cent steam limited this tendency to overtorque satisfactorily. Fig. 16 illustrates the complete results from what were considered to be about the optimum conditions, namely, steam flow immediately reduced at a rate of seven per cent per second down to ten per cent flow and retained at this figure until the pitch became negative; it was then increased at the same rate up to 70 per cent flow and retained there.

The range of P/D is from 1·33 ahead to 1·10 astern and the full movement was assumed to take 27 seconds. The results obtained were:

Time to start negative pitch	=	20 s
Ship stopping time	=	60 s
Ship stopping distance	=	1360 ft
Maximum astern propeller torque	=	165 tons ft 109 per cent of full ahead

It can be seen that a very small delay in applying the full 70 per cent steam would have kept the propeller torque below 100 per cent of full ahead:

maximum astern propeller thrust	=	56 tons 96 per cent of full ahead

Electric Drive

The type of electric drive suitable for this type of ship is a steam turbo-generator running at 5630 rev/min at full power (similar to (a), (b) and (c) above) with a synchronous motor developing 15 000 shp at 216 rev/min. The procedure for reversing[9] is to reduce the steam flow at the maximum permissible rate until it is safe to disconnect the generator and then to apply a braking effect to the motor by connecting the terminals across a resistor until the shaft speed is reduced sufficiently for the motor to be used as an induction motor with the generator field reversed. When the shaft has been reversed and the slip between generator and motor is sufficiently reduced, the generator can be connected for synchronous astern rotation of the motor.

Equation(1) for ship's speed can still be used but different equations are needed for engine and shaft speed during the various phases of the manœuvre.

Phase (1): During the initial stages while the motor is still acting as a synchronous motor, the generator and motor are constrained to revolve at the same speed:

Rate of change of speed of motor and generator (all referred to propeller speed)

$$\frac{dn_G}{dt} = \frac{Q_E - Q_P - Q_{F1} - Q_{F2}}{2\pi(I_{M+P} + I_{E+G})}$$

where Q_E is determined from equation (3).

This phase is continued until steam has been sufficiently reduced for disconnexion of the motor from the generator to be safe (about 18 s).

Phase (2): When the motor is disconnected from the generator, braking torque is applied to it by connecting the motor terminals across a resistance. From characteristics of the braking torque obtained with different resistances, an optimum resistance of 0·5 ohms was selected and values of the braking curve for this can be fed into the computation.

The applicable equations during this phase are:

$$\frac{dn_G}{dt} = \frac{Q_E - Q_{F1}}{2\pi\ (I_{E+G})}$$

Q_E depending on steam flow as in Equation (3)

Q_{F1} being taken as two per cent at full ahead and varying as n_G^2

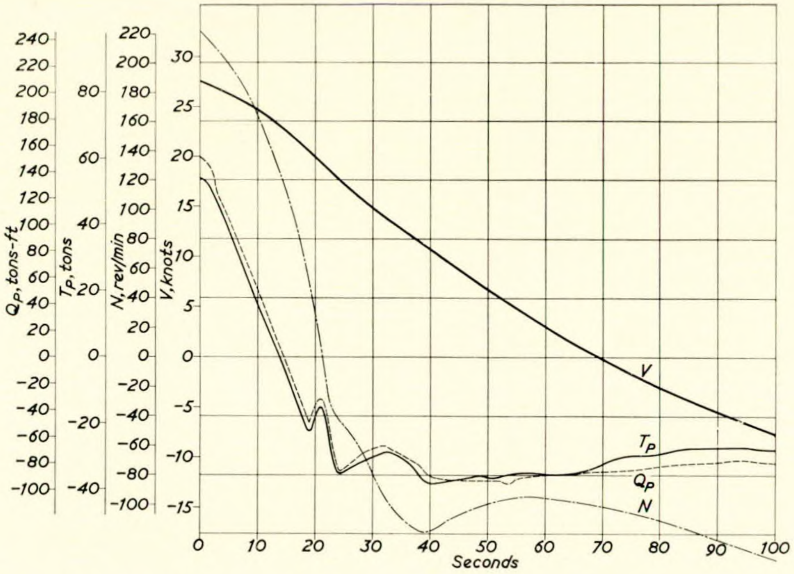

FIG. 14—*Reversing steam turbine—Full ahead to full astern*

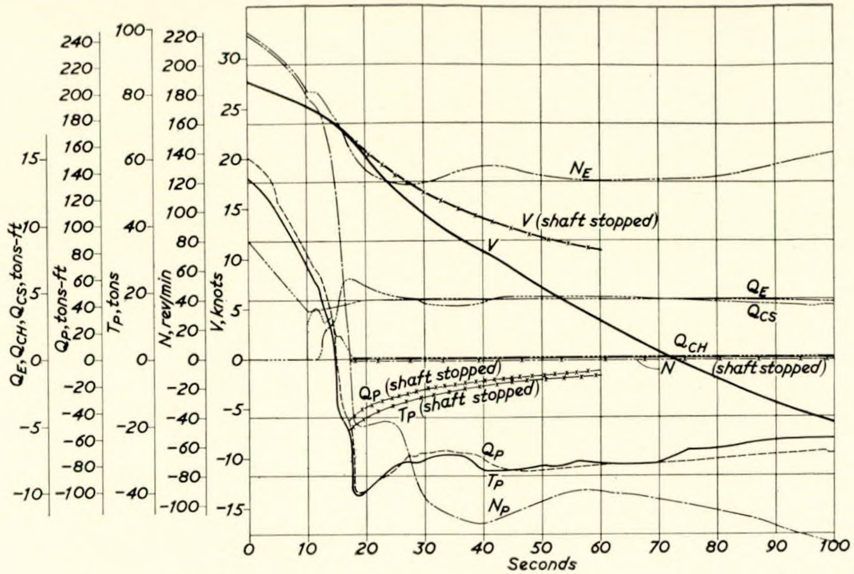

FIG. 15—*Reversing gears—Full ahead to full astern*

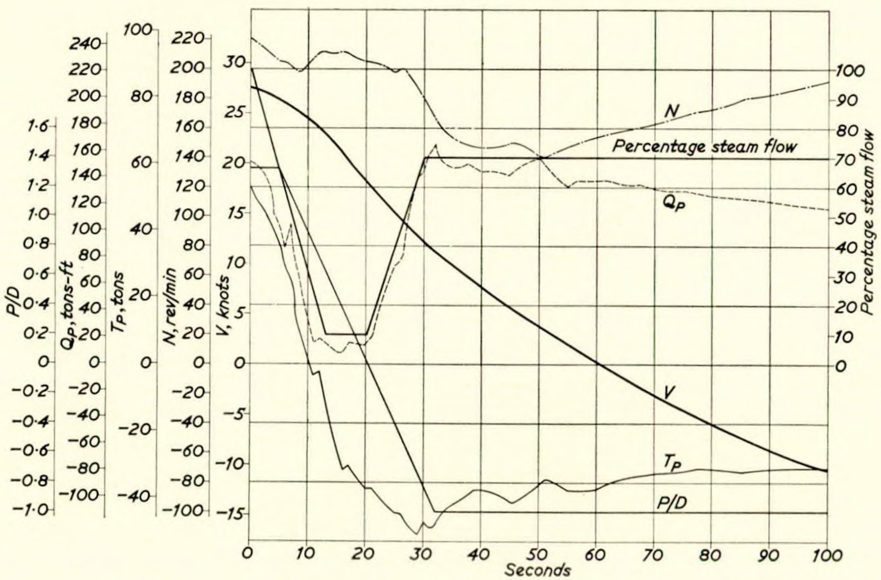

FIG. 16—*Controllable pitch propeller—Full ahead to full astern*

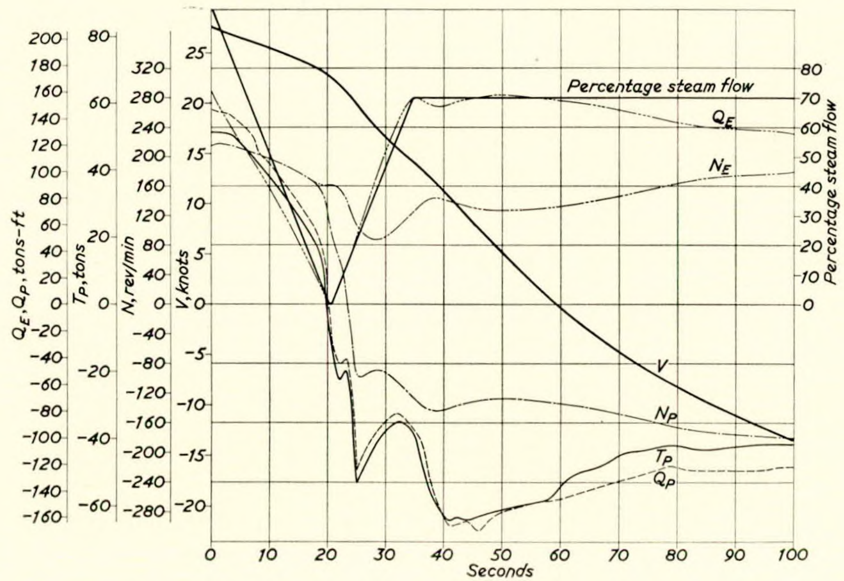

FIG. 17—*Electric drive—Full ahead to full astern*

$$\frac{dn_M}{dt} = \frac{-Q_B - Q_P - Q_{F2}}{2\pi\,(I_{M+P})}$$

Q_{F2} being taken as two per cent at full speed and varying as n_M^2.

This phase is continued until the propeller speed and torque are sufficiently low for the induction motor to be brought into operation. This has been taken as a negative propeller torque of 27 tons ft and occurs about three seconds after application of the braking torque.

Phase (3): The generator now drives the motor through the induction reversing coils and the braking resistance is disconnected. The steam flow to the turbine is increased at the standard rate of 7 per cent per second.

The applicable equations become:

$$\frac{dn_G}{dt} = \frac{Q_E - Q_M - Q_{F1}}{2\pi\,(I_{E+G})}$$

$$\frac{dn_M}{dt} = \frac{-Q_M - Q_P - Q_{F2}}{2\pi\,(I_{M+P})}$$

This phase is continued until the slip between generator and motor is reduced to three per cent when change-over from inductions to synchronous running can be safely carried out. This reduction in slip is achieved by a simultaneous reduction in generator speed and a build-up of astern motor speed.

Phase (4): The generator now drives the motor synchronously in reverse and the equation becomes

$$\frac{dn_G}{dt} = \frac{Q_E + Q_P - Q_{F2}}{2(I_{E+G} + I_{M+P})}$$

If the 100 per cent steam flow were admitted to the turbine, the maximum propeller astern torques and thrusts would exceed 150 per cent of the maximum design ahead torque and thrust which would obviously be excessive, and in the computations shown in Fig. 17 the steam flow has been limited to 70 per cent as was done for the controllable pitch propeller in Fig. 16. Even so, the maximum propeller thrust is some 25 per cent above the design ahead thrust and in practice the steam flow should not be allowed to reach the full 70 per cent until about 60 s after initiation of the manœuvre; this would also keep the maximum torque within 100 per cent of the full ahead torque. This would give a small increase to the stopping times and distances.

The results shown in Fig. 17 give the following figures:

Time to stop shaft	= 23 s
Ship stopping time	= 59 s
Stopping distance	= 1610 ft
Maximum astern propeller torque	= 165 tons ft 109 per cent full ahead torque
Maximum astern propeller thrust	= 65 tons 120 per cent full ahead thrust

The stopping distance seems disproportionately large compared with the stopping time but is brought about by the absence of effective braking during Phase (1).

Computations of Different Means of Reversing for Diesel-driven Merchant Ships

Computations have been made of the effectiveness of reversing Diesel-driven merchant ships, by:

a) direct reversing engine;
b) controllable pitch propeller;
c) reversing engines using friction clutches.

As discussed later, it is unsafe to make direct comparisons between these owing to the uncertainty of some of the input data.

a) Direct Reversing Engine

For direct reversing, a 6RD 76 Sulzer engine fitted in a ship of 22 300 tons displacement has been used in the computation. Propulsion particulars are shown in Table I.

Equations (1) and (2) apply but Q_E differs from Equation (3). Q_F has been taken as two per cent of full ahead torque and to vary as n^2.

It is assumed that when the order for "astern" is given, fuel is immediately shut off from the engine, Q_E becoming negative, i.e. a braking torque equal to ten per cent of the full power ahead torque reducing linearly with speed to five per cent of the full ahead torque at zero speed.

In the absence of any additional braking on the shaft, the engine slows down due to this Q_E effect whilst Q_P becomes negative and tends to keep the shaft speed up due to the windmilling effect of the water.

For engine safety reasons it is considered necessary to wait until the shaft speed has fallen to about 40 rev/min before applying the reversing air. When this is done, it is estimated that it takes four seconds for the engine to reverse to 40 rev/min. Thereafter it is assumed that the engine torque in reverse can be built up in six seconds from 50 per cent of full ahead torque at 40 rev/min to full ahead torque at 80 rev/min at which torque it remains until full astern rev/min are reached.

Fig. 18 shows the computed results for this manœuvre.

Delay time before reversing engine	= 84 s
Time to stop shaft	= 86 s
Ship stopping time	= 395 s
Ship stopping distance	= 4880 ft
Maximum astern propeller torque	= 98 per cent full ahead
Maximum astern propeller thrust	= 95 per cent full ahead

Diesel engine manufacturers claim that this delay before reversing is unnecessary and quote trial results in a ship with a similar engine, where the engine was running astern at 50 rev/min 22 s after the order "astern".

b) Controllable Pitch Propeller

For this study, the same ship has been used but two Ruston AO eight-cylinder engines driving a controllable pitch propeller through gearing have been assumed and advantage has been taken of the gearing to reduce the propeller speed to 82 rev/min and obtain a better propulsive efficiency. The relevant particulars are therefore:

Displacement 22 300 tons	6420 rev/min at 82 rev/min	Engine speed 450 rev/min
Propeller diameter 21 ft	Pitch ratio 0·89-0·70	Blade area ratio 0·45

Full power torque 183 tons ft
Full power thrust 54 tons
Speed 14·75 knots

Wake and thrust deduction factors as for the direct reversing engine.

Equations (1) and (2) apply and Q_F has been taken as four per cent of full ahead torque and to vary as n^2. When the order for "astern" is given, fuel is immediately shut off from the engine and the braking torque is again assumed to be equal to 10 per cent of the full ahead torque reducing linearly with speed to five per cent of the full ahead torque at zero speed. Simultaneously, the propeller pitch is reduced at a constant rate taking 18 seconds from 0·89 pitch ratio ahead to 0·7 astern.

As soon as the pitch ratio becomes negative it is possible to start increasing the fuel supply to the engine and it is assumed that it takes seven seconds to build up the full b.m.e.p. Thereafter the engine runs with maximum b.m.e.p. at the speed that the propeller torque will permit.

Fig. 19 shows the computed results for this manœuvre.

Time to reach zero pitch	= 10 s
Ship stopping time	= 394 s
Ship stopping distance	= 4295 ft
Maximum astern propeller torque	= 165 ton ft 90 per cent of full ahead torque
Maximum astern propeller thrust	= 65 tons 121 per cent of full ahead thrust

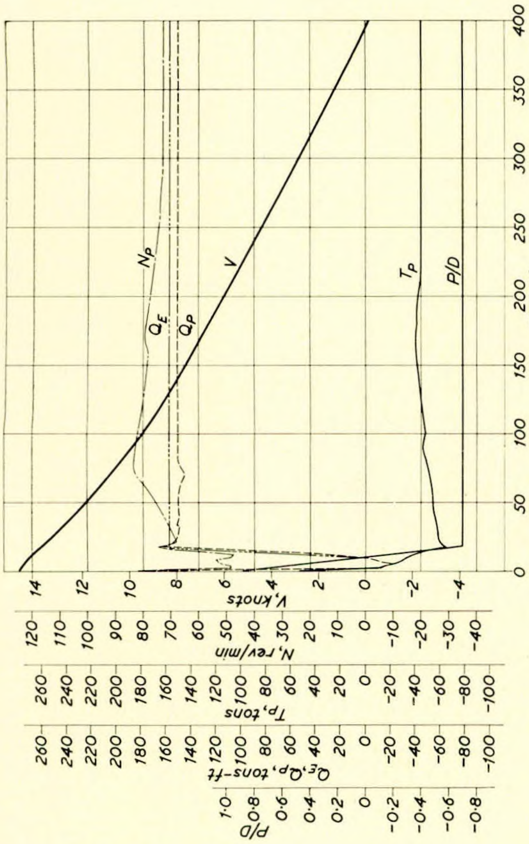

FIG. 19—22 300 ton merchant ship—Controllable pitch propeller—Full ahead to full astern

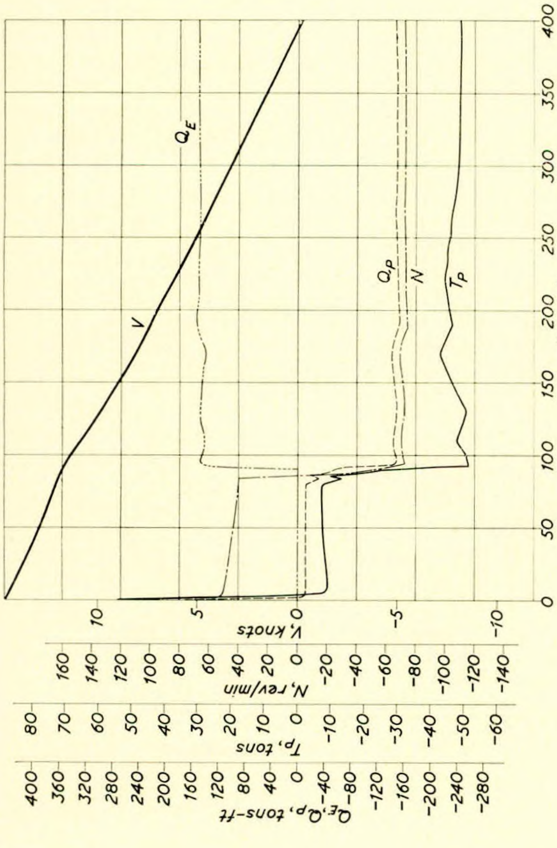

FIG. 18—22 300 ton merchant ship—Direct reversing Diesel—Full ahead to full astern

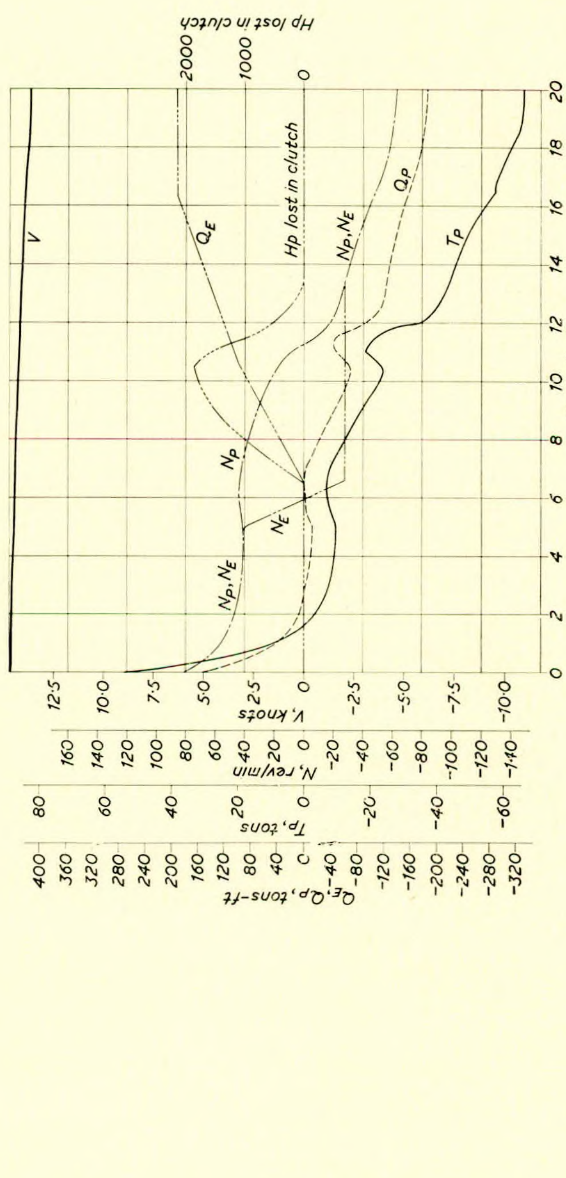

FIG. 20—22 300 ton merchant ship—Reversing with friction clutches—Full ahead to full astern

When the pitch is passing through the pitch ratios from 0·5 to 0·2 ahead there is a tendency for the windmilling effect to increase the rev/min, but the maximum speed reached is well below the full speed of the engine.

Comparing this case with the direct reversing engine, it can be seen that in the initial stages, the controllable pitch propeller enables considerably greater useful braking of the ship than the fixed pitch propeller because of the delay time before the latter engine can be reversed. At a later stage the greater astern thrust of the fixed pitch propeller compared with the controllable pitch propeller (except for a momentary peak) compensates for this.

c) Reversing Engines Using Friction Clutches

Similar engines to those used with controllable pitch propellers have been assumed and are arranged to drive the gearing through air controlled friction clutches.

Equations (1) and (2) apply at the outset, and on the "astern" order fuel is shut off from both engines and Q_E becomes a braking torque as with the direct reversing engine varying linearly with speed from ten per cent of full power ahead torque at full speed to five per cent zero speed. A delay of three seconds is allowed before commencing to release the clutches to permit the engine to slow down while propeller slip vanishes. Clutch release is assumed to take two seconds.

When the clutches are free the equations become:

$$\frac{dn}{dt} = \frac{-Q_P - Q_{F2}}{2\pi\, I_{P+G+C}}$$

and on application of reversing air

$$\frac{dn_E}{dt} = \frac{-Q_E - Q_{F1}}{2\pi I_{E+C}}$$

Q_E is taken as $\frac{80}{150}$ Q_E at full power since the reversing air can build up a b.m.e.p. of 80 lb/in^2 compared with the full ahead b.m.e.p. of 150 lb/in^2.

When the engine reaches idling speed astern and is firing, it is governed to maintain this speed while the clutches are re-engaged. It is assumed that while re-engagement is taking place, the torque transmitted by the clutch builds up linearly at a rate of 12½ per cent full ahead torque per second. Thereafter when the clutch is fully engaged, the engine speed is governed to increase from idling towards full speed and the torque transmitted by the engine increases to 100 per cent full ahead torque in ten seconds from idling when disengaged.

Computed results for the first 20 seconds are shown in Fig. 20 and the full computation gives:

Time for shaft to reverse	= 11 s
Ship stopping time	= 340 s
Ship stopping distance	= 3843 ft
Maximum astern propeller torque	= 107 per cent of full ahead torque
Maximum astern propeller thrust	= 125 per cent of full ahead thrust

The heat generated in the clutch is the product of clutch slip and the torque generated in the clutch and the horsepower being absorbed in the clutch during engagement is shown in Fig. 20 and this reaches a maximum of 1860 shp. An exceptable figure for hp/in^2 of clutch surface is 1·5 hp/in^2 and an area of 1250 in^2 can easily be accommodated in the clutch design.

Discussion of Results

When drawing conclusions from a series of results such as those on manœuvring calculations, there is a danger of believing that one has proved something that in fact one has only fed in as an assumption. The authors hope they have avoided this pitfall.

The stopping times and distances are determined almost entirely by the initial speed and the astern-power/displacement ratio, the mechanical means for applying the astern power making little difference. From Table II it can be seen that although the initial speeds differ, the order of merit of stopping times is the same as that of the power/weight ratio, e.g.

Ship	*Speed, knots*	*Stopping time*	*Power/weight ratio*
H.M.S. *Savage*	31·2	62·5 s	5·6
H.M.S. *Redpole*	18·4	112 s	1·07
Large fast ship		195 s	1·06
22 300 ton merchant ship	14·75	395 s	0·31
s.v. *British Bombardier*	15·9	690 s	0·15

The stopping distances in feet are roughly 4/5 times initial speed in knots times stopping time in seconds.

In all the reversing means which involve slipping, i.e. friction clutches, magnetic and hydraulic couplings, electric drive and reversing epicyclic gears, the main problem is the heat generated in the shipping mechanism. The problem devolves around the maximum torque that this can safely develop while slipping and the speed at which this can be applied; the larger both these are, the less heat will be generated. In the extreme, no heat at all could be generated but only at the expense of infinite torque. The design problem lies in deciding what degree of overtorque is permissible. While the prime mover and the propeller are revolving in opposite directions both are feeding energy into the clutch; once the propeller has been reversed, it absorbs some of the energy from the prime mover and the clutch conditions are quickly alleviated. With a controllable pitch propeller any heat is generated in the sea, the ideal heat sink.

3) The controllable pitch propeller starts with the initial advantage that all the ahead power is theoretically available for reversal in the time taken to reverse the pitch, this is limited in practice by the need to prevent overspeeding when passing through small ahead pitches and by the desirability of limiting overtorque. For the latter reason, the maximum astern pitch is usually limited to about two-thirds of the maximum ahead pitch. In this study it has appeared that the danger of overspeeding can be exaggerated. There is only a small range of pitches where the propeller tends to accelerate and, if suitably programmed controls are arranged to prevent the prime mover from adding to this acceleration, no overspeed should occur. This applies for Diesel drive, steam turbine drive and particularly with gas turbine drive where the windage losses of the power turbine which is not working in a vacuum exercise a large restraining effect.

4) When reversing, the possibility exists of exceeding the 100 per cent ahead torque conditions in the transmission system and the propeller. With reversing steam turbines this could arise either from the engine characteristic, see Equations (3) and (4), when the steam is applied and the propeller is still revolving in the opposite direction or, at a later stage, when the propeller characteristic imposes a large torque under the conditions of shaft rev/min and ship's speed existing at a given moment. In practice, the astern turbine in naval vessels is usually designed for a power producing a maximum steady astern torque of less than 50 per cent of the full ahead design torque and equation (4) suggests that the engine torque is unlikely ever to exceed double this value. In cross-compound designs where the astern stages are in one turbine only, the gearing may have overtorques up to nearly 200 per cent. When manœuvring from full astern to full ahead overtorquing of the gearing, shafting and propellers is even more probable as Equation (3) shows, due to the factor of 2·0 rather than 1·5 and the availability of steam for full ahead power. This is however an unrealistic manœuvre and in view of this danger, instructions were issued after trials in H.M.S. *Savage* to delay the application of more than 50 per cent ahead steam until 60 seconds had elapsed from the start of the manœuvre.

In Figs. 10, 11 etc. the torques shown are those computed at the propeller or measured in the intermediate shafting, whereas the engine torques referred to in the preceding paragraph are substantially greater and account for the sharp deceleration or acceleration of the turbines, gearing and shafting. Maximum engine torques computed were:

H.M.S. *Savage* ahead to astern: Maximum astern torque 171 per cent design astern torque
H.M.S. *Savage* astern to ahead: Maximum ahead torque 143 per cent design ahead torque

In Fig. 10 maxima of torque and thrust occur both before the shaft has stopped revolving ahead at 17-20 s and again a higher maximum at about 40-45 s when appreciable propeller speed has built up astern. These are due to the propeller characteristics, the first occurring when J^1 is about 0·95 with n positive and the second when J^1 is about 0·6 with n negative (see Fig. 9). During the latter, it can be seen from Fig. 10 that the propeller torque exceeds the engine torque causing the propeller to slow down until at about 60 s the propeller torque has reduced sufficiently for the engine torque to accelerate it again. These maxima can also be identified in Figs 12 and 14 with reversing steam turbines. The first peak is inconspicuous with reversing gears, controllable pitch propellers and electric drive, but all have the second larger maximum accompanied by a temporary falling off of propeller speed.

Of the three maxima of torque referred to above, the first, the maximum engine and gearing torque, is governed only by the quantity of steam and the forward speed of the turbines, the third is governed by the quantity of astern power used at substantial ahead ship's speeds and it is to prevent excessive torques in this condition that the limitation to 70 per cent steam was used with electric drive and controllable pitch propellers. The second maximum, that occurring while the shaft is still revolving ahead, occurs at a definite relationship of shaft speed to ship speed (i.e. a particular value of J^1) and therefore depends for its magnitude on the ship's speed when shaft reversal is about to occur. Delay in applying the full astern power will reduce this magnitude but in steam or gas turbine-driven ships this is a relatively unimportant maximum, and is merely a quirk of the propeller characteristic. In Diesel-driven ships however it may assume importance during reversal while starting air is being used for braking and reversing.

5) The assumptions made in considering the Diesel manœuvring are rather too superficial to permit of any very firm conclusions. In particular the assumption of an effective b.m.ep. being provided by the reversing and starting air regardless of the engine speed is probably oversimplified and the possibility should be considered of overpressure and relief valve lifting if reversing is carried out at too high a speed. However, as far as the propeller characteristic is concerned it is likely that the engine builder is correct in that engine reversal can take place without the long delay that many Diesel operators allow. In Fig. 18, it can be seen that the magnitude of the peak that occurs just before the shaft stops is very small, and a much larger value of Va could be afforded without embarrassing the chance of reversal. Incidentally, the large wake factor seems to facilitate reversal since Va is appreciably smaller than the ship's speed. The other requirement concerning the propeller characteristic is that the starting air must be capable of overcoming the propeller torque at zero J^1 up to idling speed but this does not seem to present much difficulty.

It is sometimes said that after reversal the rate of increase of engine speed has to be restrained to prevent cavitation accompanied by runaway of the engine and a diminution of thrust occurring. The propeller characteristics give no indication that such a condition could exist and the authors would be interested to see any trial records of transient thrust and torque substantiating this view.

6) The authors' view that the most effective way of stopping a ship is to use the maximum astern power as soon as possible is supported by:

i) the shape of the propeller characteristics;
ii) the trials in *British Bombardier;*
iii) the computation of the effect of stopped shaft in Fig. 15.

ACKNOWLEDGEMENTS

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Discussion

COMMANDER W. J. R. THOMAS, R.N. (Member), congratulated the authors on dealing with complex problems in a clear way.

He did not think it important that he should be an expert because the paper seemed to be intended for people like himself, marine engineers who did not realize exactly what the computer could do for them.

Having grown up before computers were generally available for use on everyday problems, he admitted that he was envious of the youngsters who, like Aubrey Goodwin, could now enjoy playing with those fascinating toys. They were in a position to solve problems which had been absolutely closed to the speaker. Marine engineering had always been fun and the computer was adding to it. He thought that Commander Goodwin would agree with that. Would the authors care to say how one should choose between employing a digital computer expert diverted into marine engineering, or a marine engineer who had learned to love the digital computer?

Once having admitted his ignorance, he felt he could cheerfully say how delighted he was to find, coyly concealed as a note to Fig. 1, definitions of "hardware" and "software". Jargon was now accepted at such a rate that people like himself soon found themselves unable to ask, without embarrassment, what the terms meant.

An obvious question was: "What help and advice is available to individual firms who wish to make use of computer facilities?" Were the programmes described in the paper generally available, or was a list available of those and others which were in existence? With the country in its present state, he felt it would be unfortunate if identical programmes were being developed by different authorities simultaneously. Also he wondered whether the authors saw any future for extending the use of computers beyond the straightforward calculation. Did they see any future for drawing machinery layouts in computer controlled draughting machines, calculating the centre of gravity and weight, etc., at the same time.

There were those who condemned anything from a computer as untrustworthy and those who regarded its output as having authority far beyond its feeble masters. Could the authors say what sort of accuracy was achieved in, say, the pipe stress calculation, and whether the computer, by virtue of its faster work, was now making more accurate and valid calculations possible, as a result of refinement of calculation methods which had hitherto been impossible?

Commander Thomas' present staff were having some success at getting their work on to computer tape because of their thorough understanding of their calculations. That understanding had been dearly bought at the expense of previously making laborious hand calculations. Did the authors feel that there was any risk of the next generation growing up with a thorough understanding of the computer, but lacking that basic understanding and feel for marine engineering which he was sure had stemmed in the past from intimate contact with the sums?

Some years ago he was concerned in the prototype guided missile destroyer boiler trials at the Admiralty Experimental Station at Hasler. There he came into association with the analogue computer, and he testified that it was much cheaper to try burning out the boiler on the computer than it was to do it in practice! One could try, on the computer, a manoeuvring rate, which might get one into trouble, rather than experience it on the actual boiler.

The Ministry of Defence, since those days, had been making an increasing use of computers in the marine field. In fact, Commander Thomas thought it was only a matter of time before the newspapers accused them of having not only more men but more computers ashore than they had men at sea. Typical uses were for spare gear accounting, defect analysis, usage analysis and, not least, in his own field of new machinery installation work.

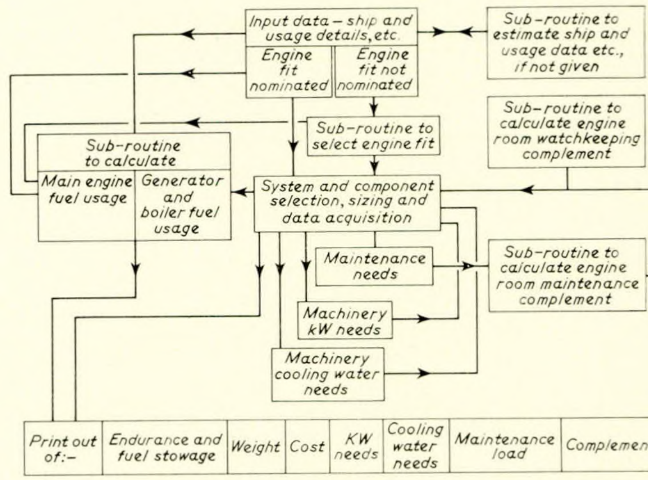


FIG. 21—*Machinery installation computer programme*

Fig. 21 showed diagrammatically a programme which was being devised jointly by the Yarrow-Admiralty Research Department and also the Ministry of Defence (Navy), Bath, with the aim of providing, rapidly, life cycle costs for machinery options for new ship designs. It might be thought that that was a very ambitious programme but the overall problem at Y-A.R.D. was near working properly. Large parts of it were working and the sub-routines were now in existence. Y-A.R.D. had been largely concerned with the overall programme while his staff had been concerned in providing information and sub-routines.

The input data came from the staff requirements. If the computer was not told what it needed to know, it made an estimate in the sub-routine at the top right. The engine fit could either be nominated or selected by the computer. Suitable systems were selected to go with the required engine fit, which allowed calculation of the propulsion fuel needs. The auxiliary fuel needs were calculated from the system data and hence the total fuel situation was printed out. The maintenance needs were assessed for each machinery item and system, and hence the manpower needs could be summated. The electrical and cooling water needs were refined in a reiterative process. Ultimately there was a printout of all the salient features of the entire machinery installation.

Both Y-A.R.D. and the Ministry of Defence at Bath were aware of the old computer tag, "garbage in, garbage out", and much of the effort of the staff of Commander Thomas was devoted to checking the data they were feeding in, while Y-A.R.D. concentrated on the overall computer programme. They were finding that not the least valuable aspect of computing was the discipline it imposed. One had to define the problem, the method of calculation and all the assumptions to be made with great care. They were committing things to paper for all to see and that was something which they had not had to do often in the past.

On reading through the paper he saw no mention of computers going wrong. He always had a pretty shrewd idea when one of his staff had hangovers and could channel the sufferer's work into areas not likely to be too disastrous if a mistake were made. How did one know when the computer was likely to produce doubtful sums?

Commander Thomas said he was far from competent to criticize the authors' many examples of problems that they had investigated on their computers. He wanted to say that the authors showed what a fascinating job the computer programmer had. It was one long succession of different problems and elegant solutions, and he even got paid for it!

In conclusion Commander Thomas said he had never actually tried maintaining his nD/Va equal to 0·2 when he had seen "full astern" rung on his engine room telegraph. He had just opened the astern throttle and shut the ahead as fast as the good Lord would let him. He was most relieved to have confirmation, not only from the authors but also from an ICT 1903 computer, that he was dead right.

MR. D. D. STEPHEN was mainly concerned with the electrical side of the paper. In the paper there was reference to an investigation into engine flexible mounting systems and the analysis of a complete shafting system. This form of exercise had been undertaken by "hand" many times in the past. With regard to its usage, for example, on turbo-electric drives, had any advantage been taken in the investigation to consider the de-coupling effect which electro-magnetic drives could have on a shaft system, either using magnetic couplings or motor generator loops? Such drives, by simplifying the simultaneous equations, were obviously a desirable thing as this permitted easier optimization of the parameters involved and he felt it would be interesting to know if the programme was able to prove that point.

Later, reference was made to Diesel electric propulsion units. That section was of particular interest to him as in a paper presented in 1962 he had suggested that one of the great advantages of electric propulsion was the amount of accurate data, both steady state and transient, which could be obtained from it very easily and directly as compared with purely mechanical systems. It was rather disappointing, however, that the comparison of alternative forms of drive, as, for example, on the reversing characteristics, gave results which were virtually inconclusive when referred to a uniform basis. The idea that if one put more power into a vessel one would probably stop it quicker was well appreciated, and that had certainly been proved by the authors in their introduction. However, the small differences in behaviour given in the section "Comparison of Different Means of Reversing", could well be outside the accuracies used in the assumptions made in the separate investigation. Thus all that could usefully be deduced was that everything was of the same order except for the differences which they still had not managed to include accurately in the programme.

Considering only the electrical system, there appeared to be two difficult points on which the validity of the results depended. Those were the form of simulation used for the electrical machines, and also their control circuits, if automatic, and the criteria used to determine the operating sequence when that was not automatic. The operating sequence was a very difficult thing to define as it depended on the operators, and it was not easy to predict precisely what they were going to do. This could have a direct effect on manoeuvring times since if the operator took 0·5s, 2s or 5s for an operation then this was a direct time delay.

There were two methods for studying the behaviour of electrical machines as part of a larger system; either to inject the characteristics exhibited by the machines under the specific operating conditions, or to incorporate into the programme a circuit or set of equations which were capable of generating those characteristics when the specific operating conditions were imposed on the circuit or equations, i.e., direct simulation. In many cases of machine simulation in some power system studies, the operating characteristics had been used to derive a form of simulation which was purely mathematical—it represented the characteristic curves with reasonable accuracy over some part of the working range. Such a simulation was usually not related in any physical or theoretical way to the machine, however, and had been shown to be inaccurate and unsatisfactory for problems outside the simple range of variables from which the simulation was derived. That was fairly understandable when one thought about the problem but was not always appreciated by users of the form of simulation. If one took the characteristics which were valid over a restricted range and represented those by mathematical equations, the equations might be used over a much wider range, but that did not mean to say that they then represented the machine over the whole range. Trouble had been experienced in the past by using such equations outside the range of validity.

The speaker felt it would be interesting to hear what basis for machine performance was adopted by the authors, and what limits of accuracy of simulation or representation were obtained in that way. The behaviour of automatic voltage regulators and speed governors also exerted a considerable influence on machine behaviour, and some information regarding those characteristics used in the programme would be useful and of great interest to machine engineers.

Investigations by electrical machine design staff, using calculated characteristic propulsion problems, had shown that the phenomena often regarded as in the class of "transient" were involved in simple manoeuvring problems and could drastically affect the normal concept of such operations in service. Mr. Stephen said he probably differed with Commander Goodwin in the use of the word "transient". When used with an electrical machine the term "transient" was normally confined to a phenomenon which was over in a matter of seconds, or less, and not what one would call rapidly changing parameters during manoeuvres lasting for two or three minutes. In the process of synchronizing, the true transients in the machine concept, did appear. That in effect meant that the machine exhibited very dissimilar characteristics to what it did under "variable" conditions. The process of synchronizing in the paper had referred to a simple three per cent slip criterion, but in practice the prime-mover speed fluctuation which could reduce the instantaneous slip in practice, meant that satisfactory service could be obtained when synchronizing was initiated at a greater nominal slip than three per cent. Was there any specific reason for selecting three per cent rather than a higher figure which would reduce reversal time?

Such investigations had shown that many of the classical "machine simulations" did not give accurate representation outside the very narrow range of operating conditions for which they were derived originally. This had been shown by the calculations performed for the *Canberra* propulsion units over a wide range of frequency, flux, number of machines in combination, etc.

Under "Electric Drive", Phase (3) the phrase "induction reversing coils" was not clear and he would appreciate some explanation.

It appeared that imposed limits on change of steam flow adversely affected the performance of the electrical system. What criteria were used to determine the Phase (1) rate of reduction of steam flow and the Phase (4) rate of increase? Presumably the propeller "stall" was the most significant factor and not the torque transmission, since the electrical loop could probably deal with very much more torque than was determined by the steam limits imposed.

In conclusion, he said that the paper should be pursued in conjunction with any service experience of marine propulsion systems which could be obtained. The wide range of other problems tackled merely emphasized the necessity of using computers in all complicated problems if a satisfactory result was required within a reasonably short time with the employment of a minimum number of engineers.

MR. K. E. LEA (Associate Member), said that his company had been giving considerable attention to ship manoeuvring computations but their prime concern had been the protection of the machinery under those conditions of manoeuvring from full ahead to full astern with the ship travelling at service speed. In general, their computation methods and equations were similar to those given by the authors and conclusion (six) was one with which his company was in full agreement.

Experience had shown that careful attention must be paid to the transient values of torque and thrust if the design was to be adequate for those manoeuvres. Therefore, the use of the interpolation methods for evaluating K_T,K_Q, and J for a merchant vessel based, in one extreme, on naval propeller data and, in the other, on Nordstrom series, which were not particularly representative of the majority of propellers used in this country, could lead to errors in the detection of transient values of torque and thrust. That could contribute to the discrepancies between measured and calculated values of torque and thrust shown clearly in Fig. 13. Reference to that figure showed that although that discrepancy existed there was good agreement in the calculation of vessel retardation

It could, therefore, be agreed that the authors' postulations were adequate for the performance of any vessel, but it could be questioned if their methods would lead to a satisfactory design of machinery. To try and eliminate some of those errors, his company had commissioned a series of model propeller trials, representative of the type of propellers used in vessels which could be powered by their product.

Typical transient values of torque were shown in Fig. 22 for a 26 000 ton displacement vessel propelled by a single fixed pitch propeller. The power was provided by two Mirrlees National KV-16 Major medium speed engines through a twin input single output reduction gear-box. Manoeuvring was facilitated by the use of a friction clutch placed between the engine and input gear pinion shaft.

The shaded area represented the normal torque speed characteristic of the engines referred to the propeller speed. It was clear that from the vessel's full ahead service speed of 18 knots, the engines would stall as the propeller torque overcame the output of the output torque unless an overload condition was allowable. Machine reversal time was in the order of 30 seconds and, therefore, the ship would still be travelling at a speed of about 17 kn and the argument still applied. Overloads were not usually allowable and a time delay had to be installed in the control system to prevent engagement before the engines could fulfill the torque requirements, the calculated value in that case was $16\frac{1}{2}$ kn, if the idling

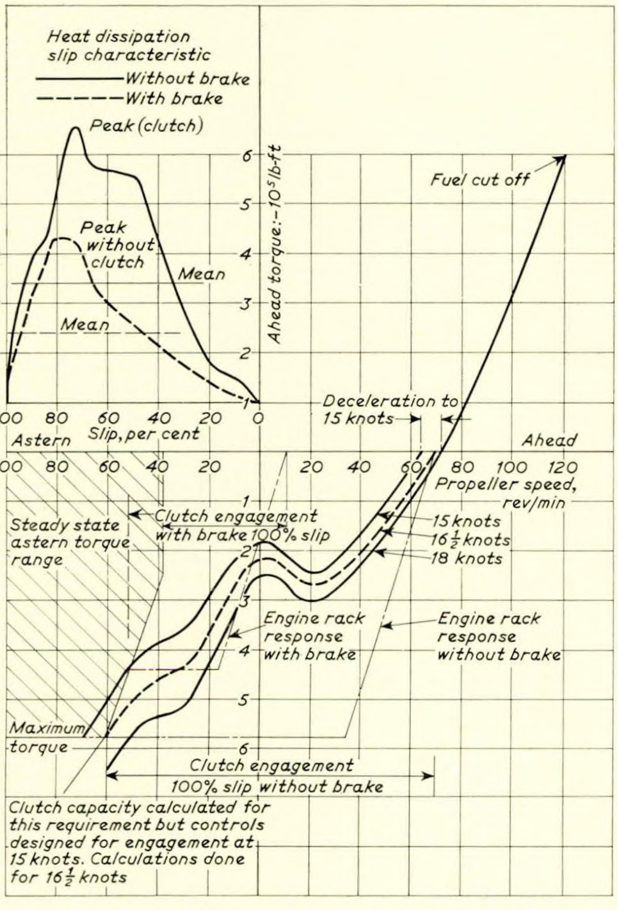


FIG. 22—*Torque-propeller speed characteristic of 26 000 tons displacement during full astern manoeuvre 13 160 shp provided by two KV-16 Major engines*

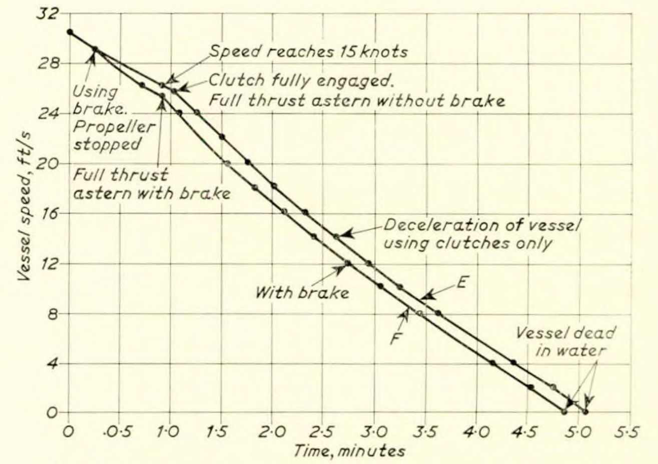


FIG. 23—*Performance of vessel driven by two KV-16 Major engines 13 160 shp at 18 knots*

speed of the engine was raised to half speed, thus taking advantage of the maximum torque availability. That was at variance with the authors' written remarks that normal idling speeds were sufficient to meet that requirement. Therefore, in selecting a clutch, the design would have to be adequate to meet theoretical requirements at $16\frac{1}{2}$ kn. The clutch heat dissipation-slip characteristic was shown in Fig. 1 for a system if fitted with clutches only, peaking at some 2800 hp/clutch with a mean of 1200 hp. On that plot there was also shown the heat dissipation-slip requirement if a brake were also fitted to each pinion shaft. That could alleviate the clutch heat dissipation but led to a more expensive machinery installation and control system without a significant improvement in vessel performance.

The comparison of vessel performance was shown in Fig. 23 where clutch engagement was not permitted until a retardation to 15 kn had been achieved. Even so, for the size of vessel, it was felt that the time to bring it dead in the water, of five minutes, was one which could be considered acceptable. The performance of the vessel when fitted with shaft brakes was only slightly improved by virtue of the extra retardation available through the propeller being brought to rest. That was not, however, a significant contribution to the overall performance.

When considering the design capacity of the clutch, the following points should be studied:

a) that the clutch is capable of transmitting the maximum torque of the engine;
b) that it was capable of dissipating the slip horsepower which was usually given in terms of:
   (i) peak horsepower/unit area of clutch surface;
   (ii) mean horsepower/unit area of clutch surface;
c) that it was capable of sustaining those horsepower ratings for a finite length of time, in the region of ten seconds for the mean and three seconds for the peak rating.

In general, the clutch that just satisfied (a) did not always satisfy (b) and (c). It was, therefore, felt that the quoted figure of 1·5 hp/in^2 required more explanation with respect to peak or mean ratings, type of material, etc. His experience showed that for organic clutch materials the ratings were: mean: 0·5 hp/in^2; peak: 0·8/0·9 hp/in^2.

Finally when considering medium speed Diesel propulsion through a reduction gear-box, and controllable pitch propellers, the clutch criterion given in (a) was sufficient, but the rate of change of pitch had to be controlled to give adequate time for the engine to meet the changing requirements, allowing for turbocharger lag, etc.

MR. G. A. ABRAHAM said the previous speakers had covered the majority of points excepting two which he wanted to raise. One related to the use of computers and the other, a more specific question, related to stopping of a ship.

The paper presented many different ways in which computers could be used for solving problems. There was a use, of which the equipment manufacturer could take advantage, which had not been brought out. An earlier speaker previously had made a small reference to the blowing up of a boiler which could be avoided by studying the system on a computer. A complex piece of equipment, such as a ship, presented many control problems which had to be solved. A significant advantage could be obtained if the hardware—and the speaker did not mean "hardware" as referred to at the bottom of Fig. 1 of the paper, but physical hardware—which is operated in conjunction with a computer simulation before applying it to the ship.

An example might be given of a propulsion engine control system. In the early design phases of the engine control the computer would simulate not only the engine but also the proposed control system as well.

Using that as a study, in the same way as the paper had suggested, control techniques and criteria could be established, and applied into the hardware of the equipment as it was manufactured.

As the hardware became available it could be substituted for the appropriate sections of the computer and the whole system could be checked out in complete safety in the comfort of a laboratory in preference to a swaying ship, and the hazards of an exploding boiler. Did the authors consider that such techniques could be usefully applied in the marine field?

His second point referred to the means of stopping a ship with a variable pitch propeller. Everybody had apparently agreed that the best way to stop a ship was to apply maximum power in reverse. Looking at the graphs on page 15 of the paper it was apparent that the method of controlling the engine speed permitted the engine speed, and therefore propeller speed, to fall away quite considerably during some of the manoeuvres, particularly with the variable pitch propeller. If the propeller was to have a better speed control on it, through presumably, the engine speed governor, would it not be possible to apply the pitch mechanism from ahead to reverse without any reduction in engine speed?

MR. B. H. SLATTER first raised a point about the use of computers. Commander Goodwin in his introduction had explained that in using analogue computers there was then a need to be able to set up problems more rapidly. It had always seemed to him that setting up a complex programme on an analogue computer was a lengthy process. Perhaps even worse, it was subject to human errors which were not easily found. Could the authors elaborate a little on how they foresaw the overcoming of that difficulty?

Turning to the ship manoeuvring calculations, the response of propellers or thrust accepted in ships during manoeuvring seemed often unnecessarily slow and, when he argued that with some of his marine friends, Mr. Slatter was told that the response of the ship was very slow and there was no point in faster machinery response. This seemed quite untrue. The operators had to anticipate the lag of the ship without having to cope with excessive lag in machinery response also. Alternatively, it was argued that if one put sudden loads into a propeller in unusual circumstances, the propeller would merely dig a hole in the water and not behave in a rational manner. The calculations which the authors had carried out, seemed to have exploded that completely because the excellence of the agreement of the practical and computer results would not have been achieved unless the propeller had always behaved in a continuous and predictable manner.

He was particularly interested in the use of controllable pitch propellers which seemed to improve the chances of good control response in a ship, and the use of those devices reminded him of an exercise some years ago in getting propeller turbine engines flying in aircraft. The problems had a great deal in common. The only real difference was that in aircraft it all happened quicker. As soon as propeller turbines were flying they had to make them land and take off from aircraft carriers. One would appreciate that they had to get the thrust off with considerable precision to give the pilot a chance to get the aircraft down in the right spot on the deck, and if he missed the arrester wire there had to be the facility to get back full thrust rapidly to give the machine flying speed before it ran out of deck. In those days they did not have computers and did manual step-by-step calculations. It would have been useful to do the work with computers, but, because one was involved in considerable manual labour, one took great pains with the assumptions before embarking on the calculations. By the time one finished the calculations one had an extremely good understanding of the mechanism of the problem. Mr. Slatter wondered whether nowadays, with everything done on computers, they did not miss those points.

A particular aspect of that problem was the high drag which the propeller could develop in zero pitch. The drag was a function of the speeds of the propeller and the aircraft. He was particularly interested that, from the curves in the paper, it seemed that much the same thing applied with a ship's controllable pitch propeller. In fact the drag shown on the curves when the propellers were in zero pitch was not far short of the drag obtained when full astern power was used, although that only applied while the ship had considerable forward speed.

In trying to go from ahead to astern with a controllable pitch propeller it seemed there were two limitations. One was that one might tend to overspeed the engine as the propeller went through the fine pitch range, although the paper suggested that this was unlikely to be a serious problem.

The other limitation was, if one went into full astern pitch straightaway with a high forward speed on, one might seriously over-torque the system. Mr. Slatter wondered whether there might be a case for bringing the propeller back to zero pitch, if necessary maintaining the rev/min near maximum, as that manoeuvre would not be difficult with a turbine governor fitted, and selecting astern pitch and power simultaneously when the speed of the ship had dropped far enough to enable one to do that without any danger of excessive torque. He would like to hear the views of the authors on that suggestion.

MR. J. NEUMANN, B.Sc. (Associate Member), remarked on the preparations necessary before using a computer. For example, the paper referred to steam turbine and thermodynamic cycle programmes, and before they could be used one had to computerize the properties of steam and water, i.e., to store in the computer all the data on sensible, latent and total heat and corresponding pressures, temperatures and entropies that were normally found in Steam Tables. Once this basic work was done it could be used not only in the programme for which it was specifically developed, but also in other related programmes. It was important to wonder whether a piece of work would be useful in the future.

Sometimes a programme was written to solve a single problem and it was found to be so flexible that it could be used with only minor changes for solving a whole family of problems. An example of that was quoted in R. F. Rimmer's contribution to the discussion of T. B. Hutchison's paper on reheat plant*. By contrast, other programmes could solve only a narrow range of problems, and in contemplating the development of a new programme it was important to optimize the degree to which it should be made general. That could sometimes be done by balancing the reduced programming and running costs of a special narrow-band programme, against the higher costs, but wider applicability, of a more general programme.

The paper referred to computer languages such as Fortran, Cobol and Algol. Could the authors provide a clear but brief explanation, of the nature and use of those languages?

Commander Thomas had already referred to "garbage in—garbage out". Computers were basically stupid machines and would do what they were told without asking questions unless, of course, the input data were outside any limits that might have been fed in. It was of paramount importance to check that the

*30 000-s.h.p. Unitized Reheat Steam Installation by T. B. Hutchison *Trans. I.Mar.E.*, 1966.

input data were correct. Furthermore, people tended to accept the computer printout as correct almost by divine right, and he stressed that it was right and necessary to challenge the results if they did not appear reasonable. Perhaps somebody forgot to convert from lb to kg or from m/h to ft/s. The computer would not necessarily know.

Mr. Abrahams had referred to use of hardware—actual bits of ironmongery—in conjunction with the analogue computer. Fig. 2 showed various types of output from the analogue computer and it was possible to make one of these outputs a series of gauges graduated in lb/in^2, rev/min, degrees F etc. as convenient for the system being simulated. A combination of such outputs with the use of real bits of equipment would result in a realistic simulation having a valuable "feel". Could the authors give some idea of the increase in cost of that sort of simulation as compared with feeding in the equations and seeing what came out on the XY plotter or multi-channel recorder?

Mr. J. Bilton (Member), felt that the authors had shown with several good examples how ship performance could be evaluated, and various schemes compared in a way which would have necessitated a formidable mathematical task without computer aid.

They had stated that the greatest part of the task lay in accumulating the correct data to feed-in, since it was self-evident that the accuracy of the results depended on the validity of the assumptions made.

In the comparative methods of reversing a single shaft steam driven frigate, the assumptions made in each case were different, for certain reasons which were quoted. It was interesting to see however, that despite very high astern propeller torques and thrusts for the c.p. propeller and electric drive, as compared with the reversible turbine and reversible gears, the stopping distance was only 15 per cent less in the case of the c.p. propeller, whereas the performance of the electric drive was the same as the reversing propeller proposals.

Under "Discussion of Results", the conclusion was drawn that the stopping times and distances were determined almost entirely by the initial speed and the astern power/displacement ratio, the mechanical means for applying astern power making little difference. That difference seemed quite significant however, when comparing the results of the c.p. propeller and electric drive as the stopping time for the latter was no better than the reversible turbine or gears, despite an astern power/displacement ratio of 6·8 as against 3·25.

Although a large astern turbine might be unacceptable, it would appear relatively easy to transmit more astern power through reversing gears than the 66 per cent allowed by the authors in that comparison. Most of the wheels would already be sized for 100 per cent ahead power.

Fluid couplings of a suitable type had already been made for a similar application involving gas turbines which would allow the reversing torque to be raised to about 75 per cent of ahead torque and a correspondingly greater flow through the fluid coupling could be arranged.

Whether those large astern powers were really essential was another matter, but if they were, then, as the authors inferred, the task was to check every link in the chain before feeding the data into the computer, so that the answer given out could be achieved in fact. The important fact that these studies could be made and various limitations pinpointed would prove an invaluable aid to designers and manufacturers.

Mr. M. B. F. Ranken (Member), said that Commander Goodwin had tried to show what types of problem could be tackled with the help of computers. He felt strongly with Commander Thomas and others that the important thing was to find a way of educating engineers and others in the use of computers and how they could help them in their own work. This was a twofold problem, first, of indoctrinating the young engineer during or immediately following his technical training and, second, of showing older engineers and top management how these new techniques could help them.

He could well remember 20 years ago the amount of time that was needed to do the stress calculations on steam pipes, to try to overcome the frightful problems which the Navy had had with steam leaks throughout the war. The calculations were done entirely manually, with perhaps a desk calculator for some of the more laborious arithmetic. It usually took two to three weeks to do the calculation for quite simple main steam pipes, such as were fitted in the Town Class cruisers.

The forerunners of the digital and analogue computers were the abacus and the slide rule. The tremendous breakthrough with modern electronic computers was their incredible and still increasing speed. The only limitation on the complexity of the calculations which could be made was the size of the computer available.

On the other hand the great cost of a computer, as well as the considerable labour and expertise involved in preparing programmes for it, made it essential to use it economically. Commander Thomas' plea for a library of computer programmes was therefore extremely pertinent, as there was an ever-increasing number of these and it was desirable to avoid unnecessary duplication of labour.

The recent tragedies in the fishing industry were not, as might have been surmised from the press reports, through any lack of forward thinking by the people concerned with the design and development of new fishing vessels. The White Fish Authority Industrial Development Unit at Hull was currently doing a lot of work, some involving the use of computers, particularly in an attempt to get not only better, but more economical catching units for the future.

One of the more important investigations which had been made formed the subject of one of the papers to be presented at the symposium in Grimsby on 22nd March entitled "Operational Research Applied to Stern Freezer Trawler Design." Two others were entitled:

"The Economic Effects of Cold Storage Capacity and Free-running Speed on One Series of Freezer Trawler Designs";

"The Freezer and Buffer Storage Requirements of Distant-Water Freezer Trawlers".

The first of these studies was based on the work to be reported in Grimsby, and provided a simulation model which could be used by trawler owners wishing to carry out their own assessment of particular designs using their own data for capital and operating costs, catch rates etc. The model was flexible and was being modified in such a way that it might be used for simulator studies of other types of vessel, for example wet fish stern trawlers, wet fishers with superchilling, middle water trawlers etc.

Two conclusions which came out of this study were:

1) that the economic importance of cold store (frozen fish hold) capacity is greater than that of the other variable considered, namely free-running speed. In terms of Net Present Value the results show a steady improvement with increase in cold store size for all classes of skipper considered, though the curve is very flat above 35 000 ft^3 capacity;
2) that for all classes of skipper considered the slower vessels are more economic than the faster ones, but the difference is fairly small in relation to the difference in power of nearly two to one between the fastest (16 kn) and slowest (13 kn) vessels considered of any given cold store size.

These results implied the choice of large, slow vessels for best economic results with a consequent tendency towards longer voyages.

The other investigation, much more elaborate, was undertaken by a post graduate student in the Operations Research Section of the Department of Economics, Hull University, as a project in the course for the Diploma in Operational Research for Management. The work had produced a computer programme which made possible, for freezer trawlers not equipped with filleting plant, the simulation of fishing and ship operating experiences which were realistic to the same degree as the data fed into the computer and which could produce a year's operating experience in a matter of hours.

The programme was so arranged as to indicate the effect of the choice of freezer size upon the operating profit; it also indicated the necessary size of buffer store, and the quantity of fish to be re-handled, for that size of freezer.

In this programme account was taken of many factors, some of them imponderable.

Constant:

a) the ability of the skipper;
b) the physiology of fish;
c) operational characteristics of freezer trawlers:
   i) reliability of equipment;
   ii) cruising speed;
   iii) capacity of the pounds;
   iv) number of men available for gutting fish;
   v) capacity of the refrigerated hold;
   vi) maximum range;
   vii) basic running costs;
   viii) running costs of freezing equipment;
   ix) de-heading machines;
d) the selling price of fish;
e) the distance to the Greenland grounds;
f) the number of days spent in port.

Variable:

a) journey delays;
b) weather conditions;
c) catch rates and tow times;
d) shifting times;
e) gutting rates;
f) mending times (including handling).

A part of the recommendations read as follows:

"The simulation model used in this investigation can provide, very quickly, information on certain aspects of the design of freezer trawlers. As a result of this investigation it became apparent that considerable savings could be achieved by matching the throughput of freezing equipment to the characteristics of the fishing grounds on which the freezer trawler was to operate.

The maximum advantages to be gained from the use of the simulation model will only be realized if the data used is comprehensive and reliable.

It is, therefore, suggested that particular importance should be attached to the collection and analysis of data concerning catch rates, weather conditions and other factors that directly affect the profitability of freezer trawlers. The conditions applicable to newly explored fishing grounds should be recorded intensively and as soon as possible. In this way it should be possible to build up a comprehensive library of information which will undoubtedly be of use in other fields of application.

The usefulness of the simulation model will be enhanced if the data required are instantly available in a form that can be read into a computer. It is suggested that all new information should be analysed immediately it becomes available and should be transferred onto a suitable input medium such as cards or paper tape.

Trawler owners should be made aware of the existence of the simulation model and should be encouraged to use it."

These various models were attempts to provide realistic information quickly on freezer trawler performance before spending £500 000 or £600 000 on a new vessel.

Perhaps the desirability of building a slow ship rather than a fast one needed emphasizing in relation to freezer trawlers. All skippers liked plenty of power, and on wet fishing high speeds were desirable for returning to market as quickly as possible and so retaining fish quality, and it was hoped, obtaining a higher price on landing. However, this argument did not apply with frozen fish and a day or even a week or two either way made negligible difference to the quality of fish stored at $-20°$F. Far more important was the way the fish was handled and prepared before freezing. Provided, therefore, that the ship had ample power for shooting and towing the fishing gear at the highest speeds likely to be used (a maximum of 5–6 kn, but usually 3–4 kn), there was clearly no argument for high free-running speeds. 12·5–13·5 kn was the almost universal speed of freezer vessels operating from Israel, Greece, Italy, Spain and Portugal to the South Atlantic and Grand Banks, Newfoundland, i.e., on much longer voyages than our own trawlers had ever contemplated so far, except for the recent experimental voyage of *Kirkella* to the South Atlantic under a White Fish Authority charter. Higher speeds would have little relevance on return voyages of 10 000 or 12 000 miles unless they also resulted in substantially higher annual earnings, as well as in a saving of time.

He agreed with previous contributors that one of the effects of preparing a computer programme was to crystallize one's own ideas about what was actually involved in any particular problem. It should soon settle what variable and constant factors were relevant and it would eliminate intuition, on which so many decisions had been based in the past. With the rapidity of advance in so many fields nowadays, intuition was something to be avoided as far as possible, since it depended on past practice, and the body of this available was often insufficient on which to base a really worthwhile jump forward, at least with any confidence of success.

It should be mentioned that some of the work described in this contribution was complementary to design studies etc. carried out at Y-A.R.D. and elsewhere, though not described in the present paper.

Mr. H. Cox was interested in computers as an operational part of the ship's machinery. Were the authors concerned with the use only in the design of hull and engines, i.e., as an adjunct to the design process? To Mr. Cox, ships were mysterious rather than computers. He had once driven a motorcar with no brakes, but ships were driven without brakes all the time. Digital machines were going into use on ships and suppliers of equipment wanted to know what would be useful, more particularly, if they were to design a computer for a ship, they needed to know where it would be housed and who would operate it.

The diagram, shown by the authors in their presentation, would be similar if the computer was a control computer, except that humans were now ship's engineers, not conventional computer staff. Both for operating and maintenance the ship's engineers had to do all the work. In most conventional computers, if the machine was faulty, one stopped and the manufacturers sent a service man along with the spares. One could not do that on a ship. Had the authors any comments on that somewhat different aspect which, although it fitted the title, was separate from the use of computers which had been presented?

Correspondence

Mr. E. P. Lover wrote concerning those examples dealing with the braking of ships. This field of research lay between the disciplines of naval architecture and marine engineering and it was relevant that this excellent analysis had been carried out by engineers and not naval architects because, as the authors had demonstrated, the problems were mechanical and not hydrodynamic.

Hydrodynamically, the problem of braking was easily solved. The naval architect provided the largest propeller possible. Arrangements were made for the shaft to be stopped immediately the executive order was received and, at a suitable instant thereafter, maximum astern revolutions were applied and the ship brought to rest. The authors had demonstrated the degree to which this situation was unattainable but the fact remained that the factors most decisive in limiting braking performance were those which limited the rate of change of horsepower available at the shaft. Efficient braking was indeed a case of needing to be the "fastest with the mostest".

Nevertheless, some propellers were better for braking than others and, in this connexion, the one single set of criteria determining the stopping ability were almost certainly the propeller size, and its performance characteristics at the zero speed, astern rev/min (or astern pitch) condition. The parameter:

$$E = K_T \left(\frac{D}{K_Q}\right)^{\frac{2}{3}}$$

was a measure of the astern thrust available from a given astern power at zero speed and demonstrated that a large propeller would always stop the ship more effectively than a smaller one of similar design. For a given diameter an examination of performance data for propellers, such as Nordstrom's, indicated that pitch ratios of about 0·8 were the optimum, performance falling off gradually as pitch ratio increased. The parameter E had an optimum value of about 10·0 $(\text{ft})^{2/3}$ for a diameter of 10 ft.

The standard of comparison of braking performance adopted by the authors was the conventional one, the distance taken to stop from high speed. There were however, dangers in using this as the only criterion. Some recent naval trials with sister ships showed that although one of them had a dangerously poor braking performance at low speed, her stopping distance from speeds greater than 15 kn was the same or less than that of the other, which was satisfactory in this respect at all speeds. The only difference between the two ships lay in the propellers and gearing, the machinery and control being the same. The value of E was significantly different in the two cases and gave a better indication of the overall braking performance than a real or computed figure for head reach.

The authors had remarked that the order of stopping times was the same as that of the power/weight ratio. Although this was true for the cases illustrated, it was to some extent an over simplification as it ignored the effect of propeller diameter by implying a direct connexion between power absorbed and astern thrust developed. From an analysis of stopping data for warships of 450 tons to 45 000 tons displacement, including those quoted by the authors, the writer had found that the reasonably close approximation to stopping time and distances were given by:

$$t = 100\frac{\Delta^{1/3}}{V_o}\left(\frac{H_o}{H_b}\right)^{0\cdot48} \text{ seconds}$$

$$s = 90\ \Delta^{1/3}\left(\frac{H_o}{H_b}\right)^{0\cdot40} \text{ feet}$$

where H_b = the maximum astern shp actually absorbed during braking;

H_o = the ahead shp being absorbed immediately prior to braking;

V_o = the ahead speed immediately prior to braking;

Δ = displacement in tons.

It should be noted that all the data were for warships with fixed pitch screw propellers and reversing turbines. A theoretical basis for the two expressions could be demonstrated, but correlation between ships of different size implied that they had geometrically similar hulls and propellers. This was in fact far from the truth and the degree to which the data fitted was therefore surprising. The stopping performance of *British Bombardier* was poor by these warship standards, even when very large displacement was allowed for. An analysis of the data for zero in Fig. 13 showed that this was not due to any inadequacy in the design of the merchant ship propeller as such. It was just too small. To equal the warship stopping standards at her displacement and power, *British Bombardier* would need to approach geometric similarity with say the Y-A.R.D. frigate or H.M.S. *Savage*, i.e. she would need to replace her 22 ft screw, either by one of 42 ft or by a pair, 32 ft diameter each. The large propeller diameter required by the warship to meet demands for high ahead speeds payed a handsome bonus when braking.

Finally, the data given in Figs 10–13 seemed to indicate that only a proportion of the available astern powers (given in Table II) were actually absorbed during the braking phases. It looked as though the astern gear ratios were incorrect and the authors' views on this would be appreciated.

MR. W. S. RICHARDSON in a written contribution said that it had become an article of faith that access to a computer was essential to any serious engineering undertaking. It was obvious that without a computer it was unlikely that studies, of the scope undertaken by the authors, would be essayed.

On the other hand, if the area of specific interest could be focused there was often much that could be said for manual calculations, even if tedious. For example, his firm made calculations, similar to those leading to Fig. 20, of the reversing characteristics of one type of propulsion system by manual iteration. Though time consuming, the direct familiarity with the various factors of the problem that this automatically entailed, led to a better understanding and "feel" of how they might be manipulated to achieve an optimum design.

In short, his company would confess the heresy of sometimes suspecting that less computing and more thought and insight might lead to a better result.

With respect to the authors' conclusions concerning the stopping of ships, they concurred that the quickest way, in either time or distance, was to apply as much astern torque as possible and as soon as possible. In their experience they had never encountered a situation where the propeller would "break-away" and not absorb all the engine torque applied to it. Though cavitation might exist throughout the reversing cycle in appreciable degree, they had in every case found that, with ahead way on the vessel, the astern torque was greater than the ahead torque would be for a given rev/min.

As long as propeller torque and thrust maintained a direction relationship and the laws of accelerated linear motion were not repealed, the deceleration and stopping of a ship would be in some degree proportional to the astern engine torque provided.

His company had had occasion to measure transient torque and speed during the stopping manoeuvre on single and twin screw tugs, a tanker, a high speed naval combatant vessel, a typical high powered single screw cargo ship, and, more recently, the 25 kn Ro-Ro vessel, *Admiral Callaghan*, having 20 000 hp gas turbines on each of its twin screws. In all cases, the propeller characteristics during the reversal followed a similar basic pattern, but differed widely in specifics.

All showed the torque peak prior to reaching zero rev/min, described by the authors as merely a "quirk" of propeller characteristics. However, whereas the torque at this peak reached a maximum of only 40–60 per cent of the full ahead torque in the examples presented by the authors, they had measured instances where it exceeded 100 per cent by a considerable margin. Therefore, this peak might not be as unimportant as the authors implied; it could be at least embarrassing, and in some circumstances disastrous, if the reversing provisions were not able to muster sufficient torque capacity to carry the shaft speed over the peak and into reversal.

They had not noted, perhaps because they were not concerned nor looking for it, a secondary torque peak and succeeding valley following reversal and prior to establishing essentially steady state astern rev/min. However, from the propeller characteristic curves, Figs 8 and 9, and the later illustrations, particularly Fig. 12, showing trial data, it would appear that such a secondary peak did in fact occur. Elementary velocity diagrams of the changing angle of attack of the water on a propeller blade, as it was brought through zero speed and into the reverse regime, would suggest why such a condition might occur.

In any event, the torque dip following the peak was momentary, was in magnitude a modest diminution rather than a breakdown, and was automatically self-correcting in that the resulting increase in astern rev/min quickly re-established a rising torque characteristic. Therefore, it would be incorrect to regard the phenomena as leading to potential "runaway" of the main engine and the authors' basic premise, that the greater the astern torque the quicker the stop, was undoubtedly valid.

It would only be fair to point out that high astern power created quite a commotion at the stern of a vessel and the crew might be forgiven for minimizing it by applying astern power gingerly, particularly when the difference in head reach might not be of major significance on a vessel of modest power to weight ratio.

The authors appeared to have some concern about the maximum torque to which the machinery and shafting might be subjected during reversing by one method or another which would be a matter of academic interest only. While it was true that either a steam or a gas turbine could develop substantially 200 per cent torque in the stalled condition and might do so during a reversal, depending upon how the valves, controls, or governors were handled or programmed, this should have presented no hazard to the machinery, considering the normally used factors of safety for the mechanical strength of the drive components.

In considering friction clutches, the authors had indicated a heat loading of 1·5 hp/in^2 of friction surface as satisfactory, which was probably quite true. To prevent any rash conclusions being drawn, it should be noted that the relationship of heat rate to total heat capacity of a dry friction clutch was an inverse function, and that its ability to absorb heat at this or any other rate would depend also on the duration of application.

The only criticism of this excellent study was that, inevitably, in considering so many alternate types of installation, a number of qualifying assumptions must be made, which might not present a fair comparison of the potential of each type. The authors had been careful to point out the difference between normal operating practice and what might be accomplished feasibly in reversing a large, slow speed Diesel engine installation. However, for the frigate comparison they assigned the reversing gear a 33 per cent astern power capability versus 70 per cent for a controllable pitch propeller or an electric drive, on the basis of the cost of added reverse gear capacity, which was unfair and probably incorrect. Actually, a reverse gear of any of several types could provide 70 per cent astern power or even 100 per cent astern power, if wanted, as in the case of the *Admiral Callaghan* gears. In such a case, the computations for stopping time and distance would yield a different comparison.

The objective of the paper was, of course, to present a method of analysis rather than to promote the virtues of any particular type of machinery, but even with qualification he feared that data and conclusions presented might be used for purposes not intended.

MR. M. N. PARKER, wrote that the paper provided an excellent review of the range of applications of both digital and analogue computers to marine engineering. Pipe stressing was an early and very important field of application. Other aspects of pipework design readily lent themselves to computer methods. Fully integrated pipework design and detailing packages which put out complete schedules of components had already been developed in other industries and B.S.R.A. was looking to see how they might be adapted to marine engineering.

An application of analogue computers had been made by B.S.R.A. in the field of the steering characteristics of ships. In one case features of ship trial records which were originally attributed to experimental error were found to be reproduced in the records obtained from the analogue computer and thus shown to be inherent in the equations of motion. In this work, as in cases to which the authors referred, it was found that the analogue approach was most useful in the initial exploration of the problem and that subsequent more detailed investigations could be carried out more expeditiously using a digital computer.

The paper referred to the results of stopping trials carried out with the steam turbine tanker *British Bombardier*. These trials, incidentally, were carried out in 1962 and '63 and not in 1966 as stated in the paper. At the time, a number of different theories as to the best method of taking way off a large relatively low-powered ship were being canvassed and one of the objects of the trials was to investigate these. The object of the attempt to keep the propeller turning ahead, but at reduced revolutions, in the initial stages rather than reversing the shaft immediately was to operate on the part of the characteristic indicated by the authors. However, with the *British Bombardier* propeller pitch ratio of 0·78, a rather larger negative thrust coefficient would be achieved than for the pitch ratio of 1·4 for which characteristics were given in Fig. 7 and this would occur at a somewhat higher value of 1/J. Further, since the initial retardation was relatively small for a ship of this type the term PD^2Va^2 would remain quite large and there was, on the face of it, a good case for operating in this range rather than in the range of negative J where cavitation might be expected to have a significant effect on the characteristics of the full scale propeller. The detailed results of the *British Bombardier* trials, which were shortly to be published, showed that substantial reverse thrusts and torques were, in fact, developed while the shaft was still turning ahead and the inability to hold this condition was attributable entirely to the insensitivity of the control of a stern steam pressure. Nevertheless, there was no evidence of any significant advantage in operating in this way and the crash astern manoeuvre had the advantage of maintaining a near constant demand for steam from the boilers.

DR. J. F. SHANNON wrote that illustrating the application of computers in marine engineering, the authors had taken ship manoeuvring computations as an example, and from there proceeded to an analysis of the manoeuvring performance of different ships and reversing systems. The outcome of this led the authors to the conclusion that the most effective way of stopping a ship was to use the maximum astern power as soon as possible. He agreed, providing there was no great loss of gripping power of the propeller during the stalling periods.

This conclusion was supported by a report* on the "stopping manoeuvres in case of emergency", *Schiff und Hafen* 1966, Vol. 18. The stopping distances and times were plotted as functions of (a) the astern power, (b) the time of reversal of the engine and (c) the time at which the propeller began to go astern. The yaw of the ship was also determined and shown to increase with increase in astern power. It would appear to be necessary to take yaw into account.

The shortest time of applying astern power of which he was aware was that occurring on the COSAG *Tribal* Class frigates and County Class destroyers when on gas turbine manoeuvring, reported in Dr. Shannon's contribution to the discussion on the paper by J. M. Dunlop, and E. B. Good.† Without reducing the power of the engines the astern power was applied by simply emptying the ahead fluid coupling and filling the astern fluid coupling simultaneously in 4–5 seconds. The maximum power of this manoeuvre was much less than the full ahead power so there would be little loss of astern peak thust. One result of this was to induce a propeller torque of 120 per cent of the initial torque at about zero rev/min instead of about 50–60 per cent with a delay of 15 seconds before applying astern power. This delay effect was described further in Dr. Shannon's paper on "the development of high power marine reversing gear"—6th Round Table discussion on Marine Gearing, April 1967—which was known to the authors.

It was surprising, therefore, that when dealing with reversing gears, the authors did not use the developed design of fluid coupling which gave more power and greater oil flow for the same size as used in the original COSAG frigate and County Class machinery.

The quickest way of stopping the ship with the fluid coupling system when the engine was clutched in was to reduce the engine power and fill the astern fluid coupling, the ahead coupling being empty. In this case the coupling filled in less time than it took the engine to fall in power and speed sufficiently to declutch. Thus valuable time was saved in the manoeuvre and the stopping time and distance would be much improved on that quoted by the authors.

If it happened that the clutch was out and the set was in

*Mockel, W. and Hattendorf, H. G., 1966, *Schiff und Hafen*. Vol. 18.

†1963, *Trans. I. Mar. E.*, Vol. 75, p.25.

manoeuvring condition, the engine power need not be reduced, thus giving a further improvement in stopping time and distance.

The type of ship was important. For instance, hydrofoils and very fast ships pulled up quickly when power was taken off due to the ship settling in the water and increasing the resistance of the ship. This raised the much discussed question of applying hydro-brakes, like the flaps on aircraft, to assist in the stopping of very large ships. Could the authors comment on this?

As an example of the use of computers in the mechanical design of steam turbines the development of long last row blades was outstanding. It was now possible to calculate to the required engineering precision, frequencies and nodal line patterns of all the vibrations up to the first 17 or 18 modes, taking into account all the couplings in the twisted blade due to torsion, bending, and, rotary inertia, shear and lacing wire effects; such calculations were impossible before the introduction of computers.

The blades were manufactured on computer controlled machines giving a reduction in manufacturing time.

While the paper had dealt a great deal with manoeuvring, could the authors expand similiarly on engine flexible mountings?

MR. H. SINCLAIR in a written contribution commented that the paper was of particular interest to marine engineers concerned with propulsion machinery from the aspect of the manoeuvring qualities; especially the "crash astern" stop order, to be executed reliably to achieve the minimum head reach of the vessel.

In response to the invitation expressed by the authors in the last paragraph of the Discussion of Results, attention was drawn to an article*, wherein actual test results relating to a direct reversing Diesel engined vessel were reported in a graphic form, by Dr. Otto Hebecker. The tests were carried out in the m.v. *Prauenheim*, a single screw ore carrier of 23 300 tons loaded displacement, propelled by an M.A.N. engine of 6000 hp at 106 rev/min, direct coupled to a five-bladed propeller of 5·6 metres diameter.

The graph in Fig. 24 was derived from an enlarged photograph of this diagram and had been prepared for comparison with Fig. 18. The latter figure had been redrawn as Fig. 25, and in both figures a curve had been added to show the rate of retardation expressed in kn/min and the head reach was shown in feet.

The "crash astern" manoeuvre was carried out off Dakar, with slight wind in a calm sea. From the moment when the fuel supply was cut, with the vessel at 13 kn, the engine rev/min dropped from 106–40 rev/min in 40 s, at which point the speed had dropped to 12 kn.

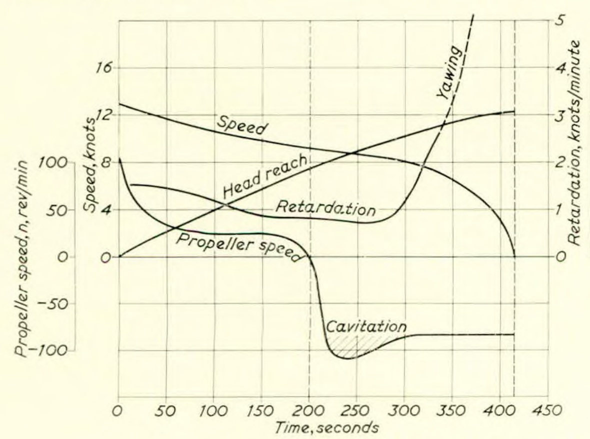


FIG. 24—(*derived from* Schiff und Hafen)

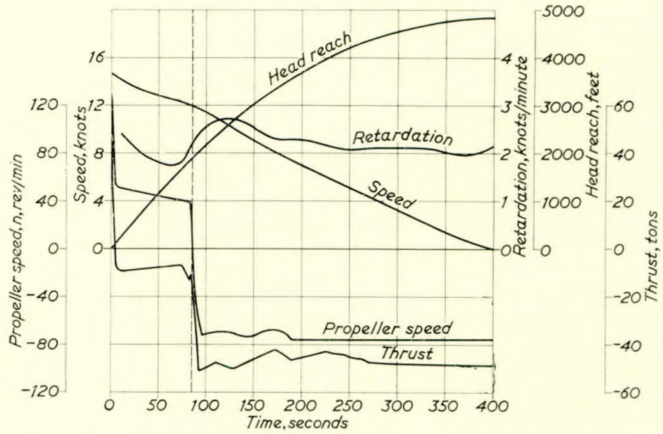


FIG. 25—(*derived from Fig. 18*)

After a further 120 s, when the speed had dropped to 9·8 kn and the propeller speed was 25 rev/min, reverse starting air was admitted causing the engine to stop and reverse 40 s later, at 200 s, the speed then being 0·3 kn. The propeller speed reached 100 rev/min in a stern rotation, at 220 s, with the vessel moving at 9 kn, under which condition propeller cavitation had evidently set in, and was sustained for about 40 s with the propeller "beating the air". At the end of this period the speed had dropped fractionally to 8·6 kn.

The graph illustrated that, as the propeller cavitation died away, at about 275 s, the retarding effect increased; noting however that the ship was then yawing strongly to starboard, with the propeller continuing in astern rotation at a speed of 83 rev/min. Eventually, after 430 s from the commencement of the trial, the vessel was dead in the water, the head reach measuring about 5900 feet.

The retardation curve in Fig. 24 was interesting as it showed an initial rate of 1·6 kn/min, which decreased to a minimum figure of 0·6 kn/min after 260 s; which corresponded with the propeller cavitation region shown on the graph, and the related condition of minimum negative thrust. Incidentally, the graph indicated that the engine speed in astern rotation increased suddenly over the five second period from 205–210 s, constituting a "run-away" condition until the governor became effective to limit the engine revolutions to the designed maximum speed.

It was pointed out by Dr. Hebecker that the head reach could be shortened by earlier admission of reverse starting air, whereby to reduce the estimated stopping time by about 60 s. In this case, the engine speed prior to stopping and reversal would be about 85 or 90 rev/min, and surely the operation would be attended by a long sustained propeller cavitation period, with a low value of negative thrust. The estimated reduction in head reach was not, however, stated.

The authors emphasized the importance of applying astern power rapidly to reduce the head reach in a "crash astern" manoeuvre, and this would appear to be axiomatic with any kind of propulsion machinery.

Some published test readings* showed however, that a deliberate time lag, e.g. of the order of 10 s additional to a normal delay of about 10 s, could be made before commencing the application of astern power, without having any adverse effect upon the measured head reach of the ships in question.

Regarding the negative thrust developed by the propeller, while it was being decelerated, but was still in ahead rotation with the vessel at relatively high speed, an intriguing graph had been published† relating to a paper by G. M. Boatwright and J. M. Turner.

*Hebecker, O. 1961. "Stoppen aus voller Fahrt mit voller Rückwärtsleistung". *Schiff und Hafen*, Vol. 13, No. 4, pp. 351–352.

*Horne, L. R. 1944–45. "The Stopping of Ships". *Trans. N.E.C.I.E.S.*, Vol. 61, pp. 311–332.

†*Bu Ships Journal*, September 1953, p. 25.

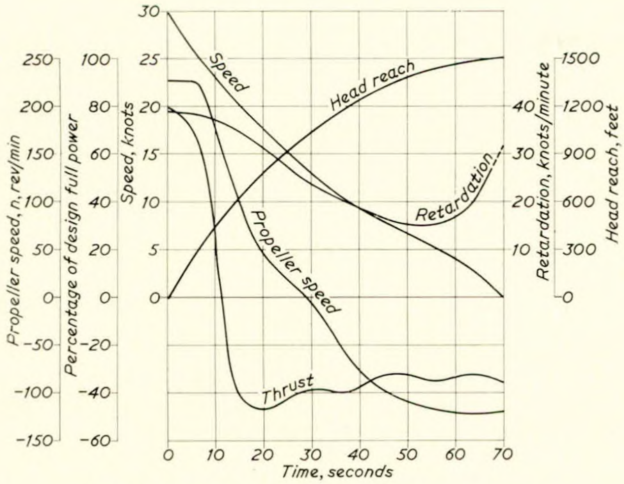


FIG. 26—(*derived from* BuShips Journal)

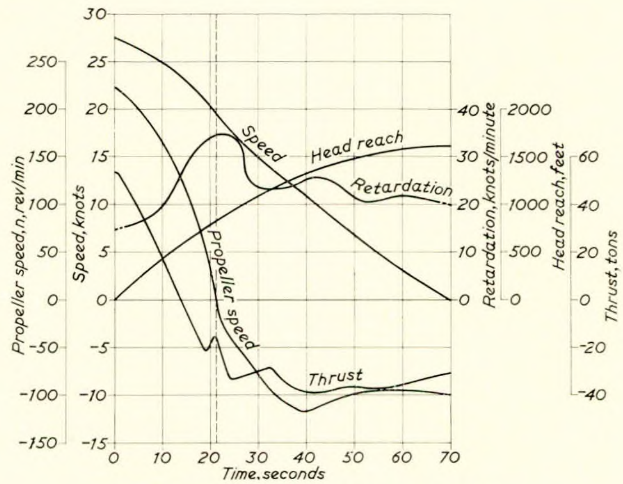


FIG. 27—(*derived from Fig. 14*)

It would seem that a 30 kn steam turbine driven vessel was the subject of "crash astern" stopping tests broadly similar to Fig. 14, and two sets of curves were shown, based respectively upon "instrumented data" and "computed results".

Fig. 26, based upon the "instrumented data", brought to light an important characteristic, the powerful, negative thrust developed by the propeller in this class of vessel, while being decelerated, but still rotating in the ahead sense during the first 25 s of the manoeuvre.

For convenience of comparison Fig. 27 was presented, based upon Fig. 14, with the addition of curves to show the rate of retardation expressed in kn/min, also the head reach in feet.

Reverting to Fig. 26, after 11 s the propeller thrust dropped to zero, while at about two-thirds of full speed, then a high negative thrust was rapidly developed and attained its maximum value 20 s from the commencement of the test, while still in ahead rotation at about 20 per cent of full speed.

The astern turbine was effective in stopping the propeller and reversing its rotation after 29 s, and when in astern rotation with the vessel still going ahead, the negative thrust was sustained but at a somewhat lower value until the vessel was dead in the water, after the lapse of 70 s.

The authors made several references to the extent of which the "crash astern" manoeuvre might be attended by overload torque in the gear system; as shown by the computed studies. It would be helpful if a view could be expressed concerning the permissible overload torque, expressed as a percentage of normal full power torque, that would be entertained for each of several classes of turbine reduction gearing, such as:

1) conventional merchant marine "soft" gears;
2) surface hardened and ground pinions, meshing with through hardened gearwheels;
3) surface hardened and ground pinions, meshing with surface hardened gearwheels.

In a turbine manoeuvring gear, such as that referred to by the authors, wherein ahead and astern hydraulic couplings were provided, the means were available to control the turbine power and speed, and to limit the rate of filling of the working circuits as might be desirable, to ensure that the recommended maximum value of overload torque on the gearing was not exceeded, even during the most severe "crash astern" operation. On the other hand, it was surely desirable to make use of the permissible short term overload capacity of the gearing, to stop the ship with the minimum of head reach.

It was advantageous in such a manoeuvring gear to arrange that in the "Stop" position of the control lever, both the hydraulic couplings were maintained in the full condition; whereby to constitute a powerful hydrodynamic brake.

Earlier types of turbine manoeuvring gears incorporating ahead and astern hydraulic couplings were based upon the practice with internal combustion engine driven reverse-reduction gears, wherein the neutral condition of the control was necessarily arranged with the ahead and astern clutches disengaged.

There were several advantages in the new control practice, notably the dissipation within both couplings of the major proportion of the kinetic energy of the turbine and gearing during the retardation period, prior to reversal of the propeller. Furthermore, full use was made of the propeller braking effect due to the high torque capacity of both hydraulic couplings, and was attended by a substantial reduction in the amount of energy dissipated as heat in the oil cooler.

The propeller braking torque was divided about equally between the ahead and astern pinions instead of being concentrated upon the astern pinion, when stopping the propeller prior to changing over to astern drive.

Reverting to the differences in performance shown by Fig. 24 in relation to Fig. 25 and likewise by Fig. 26 in relation to Fig. 27, there was no doubt that hull and wake effects were present, as well as differences in propeller characteristics. Would the authors offer their observations upon the major points of difference—with particular reference to the ship retardation features?

Authors' Reply

In reply, the authors said that Commander Thomas, in opening the discussion, had raised several interesting points, particularly the development of the programme in which they were jointly concerned, which compared life cycle costs for new naval machinery installations. Perhaps when this programme was fully developed, it would tell him the optimized proportions in the crew of marine engineers turned computer experts and *vice versa*. At Y-A.R.D. it was found that a mixture of these two types gave good results.

The authors did not agree with the doubts he and others had raised about the inevitability of losing intimate contact with problems solved on the computer. Provided one arranged for adequate print-outs of all the relevant factors involved in a problem, one could obtain an even better feel for the effect of variables than one could possibly obtain from hand methods. These print-outs could also be used for hand checking the important results and study of them would show that the computer had been supplied with the correct information and whether it was operating properly. However, any experienced computer operator could tell whether the machine was malfunctioning without reference to the results due to the built-in checking devices. Mr. Neumann had stressed the need to challenge results from the computer and with this the authors agreed, in that what one was checking was not so much the machine itself as the humans associated with it.

Compared with hand calculations, the computer programme for pipe stressing was much more accurate as illustrated by a practical study recently carried out. This study involved a comprehensive strain gauging of an operational piping system and a comparison of the results so obtained with those calculated by the computer programme at the design stage of the piping system showed agreement to within ± 10 per cent which probably bordered on the accuracy limitation of the measuring method employed.

The authors were unaware of any organization to prevent identical programmes being developed simultaneously by different bodies. Individual computer bureaux such as the authors' own did have lists which could be supplied on request, of all the programmes developed, prepared and used by themselves. Such programmes might need some tailoring to meet individual user's requirements.

The authors endorsed Mr. Stephen's comments in connexion with the danger of adopting theoretical expressions which were only a convenience for solution over limited parts of the range. It was necessary to use the basic physical laws which governed the particular processes of interest.

The simulation of the extremely shortlived electrical machine transients could be a study in its own right as part of the overall machinery simulation. So far as the computation of the stopping distances of ships was concerned, that particular aspect could be ignored, because the frequency of the transients was very high in relation to that of the ship and did not affect the actual ship's stopping time.

On the matter of the induction torque, why certain percentage slips were assumed and why a particular speed should be chosen, a particular example had been taken to illustrate the

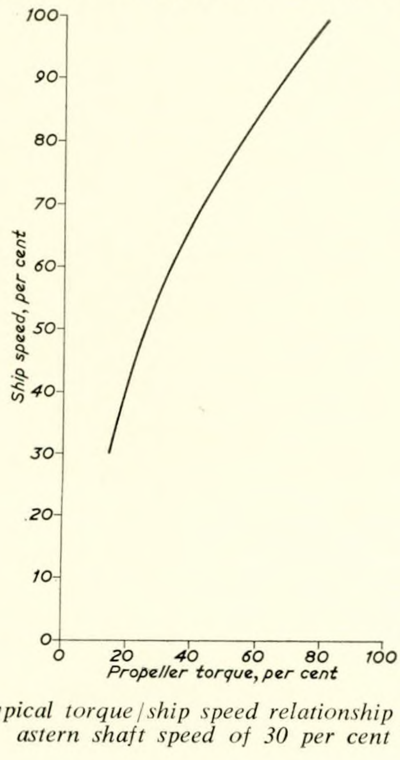


FIG. 28—*Typical torque/ship speed relationship for constant astern shaft speed of 30 per cent*

point which occurred if it was required to stop or reverse a piece of equipment. If ship's speed and the actual torque from the propeller were plotted at a constant astern running shaft speed the curve in Fig. 28 was typical.

It had to be ensured that the electrical machine could develop sufficient torque to pull the machine into synchronism at the particular chosen ship's speed and that was what was used to determine the safe ship's speed for synchronizing.

Mr. Abraham had remarked on the testing of control hardware in conjunction with computers before it was installed in a ship. The authors endorsed his remarks and agreed that it was a particularly useful way of using an analogue or hybrid computer, and that it was the intention to adopt these procedures in forthcoming naval installations.

Mr. Slatter remarked on the difficulty of setting up problems on an analogue computer and asked how one could overcome it. One of the most time-consuming things in setting up an analogue computer programme was the patching up of the problem, but with patch panels this could be done off-line. The setting up of potentiometers could be speeded up with the use of punched tape. One hundred and thirty potentiometers could be set in three minutes.

Another matter raised was with regard to the pains taken over assumptions when one had a lot of hand calculations to do and did use of computers destroy this. On a computer one had the facility to include things which could not previously be included using longhand calculation. In general, fewer assumptions had to be made and one had to think more clearly to put the whole analysis on the computer.

Mr. Neumann rightly stressed the need for an economic assessment of the cost of preparing programmes and the likelihood of their being generally useful.

The data held in the computer for steam thermodynamic cycles were not in the form of the vast amount of numbers shown in the Steam Tables, but were held as general solutions, i.e., the relationships were given as functions of one another, e.g. the enthalpy of steam $H=f(P,T)$, where P was the pressure and T the temperature, was how the machine held the information and naturally, the function $f(P,T)$ was fairly complex. The speed with which different machines could evaluate the enthalpy varied depending on their cycling and instruction obey times, but something of the order of $^{1}/_{20}$ enthalpy could be expected.

An explanation of the High Level Languages, Fortran, Algol and Cobol had been requested. "High Level" implied that programmes written in these languages transcended different types of digital computer without reprogramming and that one language statement would transcribe (compile) into one or more machine instructions. Fortran (Formula Translation) and Algol (Algorithmic Language) had their origin in the United States and Europe respectively. They were both intended to permit scientific and technical problems to be programmed without the programmer spending any time in learning how to programme or in becoming conversant with the computer on which it was intended that the problem should be run. Since the statements in both languages had a general resemblance to algebraic formulae and English sentences and the rules of syntax were simple and clearly defined, it permitted engineers to adapt themselves quickly to stating and solving their problems by this method. Cobol (Common Business Orientated Language) was intended to serve the same purpose for commercially orientated people. On the whole, Cobol statements had a general resemblance to very simple English sentences with strict rules of syntax.

An example was shown here in the three languages.

Add quantity A to quantity B, multiply this by quantity C and divide by quantity D and arrive at an answer E.

Fortran	*Algol*	*Cobol*
E=(A+B)*C/D	E:=(A+B)×C/D	Add A to B giving X_1 multiply X_1 by C giving X_2 divide D into X_2 giving E

Mr. Cox remarked on the use of computers not for design purposes but for control. Digital computer control was something which was widely used in the process industries and it was gradually coming to the marine industry. One had to consider whether there was really a need for it, whether the amount of equipment necessary to cope with the central process could be produced and whether people who were not trained in computers could use it. There was also the problem of dealing with the control of the machinery during computer failure. That was particularly important in the marine context where the necessary people and equipment to put the situation right were not available. Despite all these problems the authors believed that there would be a steady increase in the amount of computer control fitted at sea.

Mr. Ranken had given some interesting results of computations carried out by the White Fish Authority, but these were really outside the scope of the paper. It was, however, noteworthy that the computer suggested that more economical results could be obtained by designing for lower speeds than the current ones. In several economic studies for different types of ship, carried out by the authors, a similar result had been obtained, but there seemed to be a hidden bonus for speed that the computer did not recognize; perhaps sailors just wanted to get there quicker, or the shipper was always late starting.

On the question of manoeuvring, there had been a number of very useful contributions to the discussion and it was gratifying to receive almost universal agreement that "the fastest with the mostest" was the most effective way of decelerating a ship; in fact, had it not been for Mr. Sinclair's contribution, the authors would have felt that they themselves had been "beating the air".

Mr. Lover had drawn attention to the beneficial stopping effects of large propellers and of small P/D ratios. To test the magnitude of the effect of propeller size, results had been computed for two alternative propellers to that used for Fig. 14. The B.A.R. and pitch ratio were kept constant to confine the differences to propeller diameter and revolutions.

The results with an 8 ft and a 16 ft propeller were shown in Fig. 29 (cf. Fig. 14), from which the following comparative table could be drawn up:

Propeller diameter, ft	Rev/min	Time to stop shaft, s	Time to stop ship, s	Stopping distance, ft
8	430	21	84	1864
12	220	21	70	1610
16	147	23	63	1512

The difference in stopping times and distances of the ships were quite dramatic, and it could be seen that the large diameter propeller produced a much larger stopping thrust throughout and, particularly in the earlier stages, both before and immediately after the shaft reached zero revolutions. Dr. Shannon,

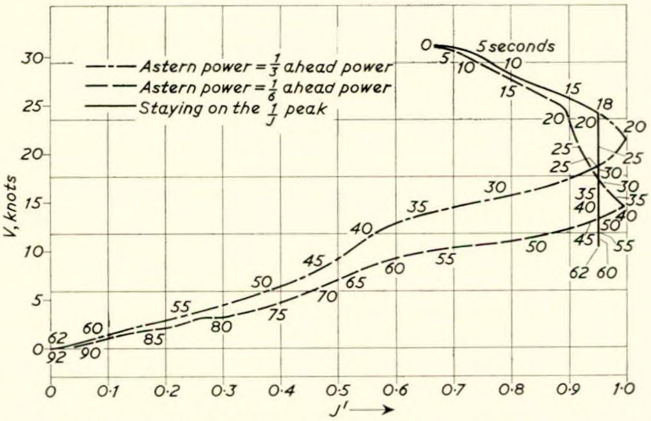


FIG. 30—*Comparison of decelerating times using astern power of one-third and one-sixth ahead power with that staying on the 1/J peak rotating slowly ahead*

Mr. Richardson, Mr. Parker and Mr. Sinclair referred to greater astern torques at and around zero revolutions than the paper indicated, and it could well be these were associated with comparatively large propellers for the power developed. In the case of the COSAG machinery where the reversing gears used for gas turbine manoeuvring were designed for only a small proportion of the combined steam and gas turbine power for which the propellers were designed, this was certainly so. Mr. Richardson's point was well taken that the characteristic applicable to the actual propeller to be used was needed before it could be assumed that the first peak of propeller torque could be easily surmounted.

Mr. Parker gave some useful additional information concerning the *British Bombadier* trials, and it was gratifying that he confirmed the advantages of an analogue approach followed by subsequent detailed investigation using a digital computer.

In presenting the paper, the authors produced Fig. 30 showing the relative effect of using astern power of one-third and one-sixth of the full speed power, and of remaining on the peak of torque and thrust that occurred while the propellers were still revolving ahead. This demonstrated that in the 62·5 s taken to stop the ship with one-third power, the use of one-sixth power reduced the ship's speed to 7½ kn while the peak method only reduced it to 10 kn.

In the light of Fig. 29, it could be seen in Fig. 30 that with

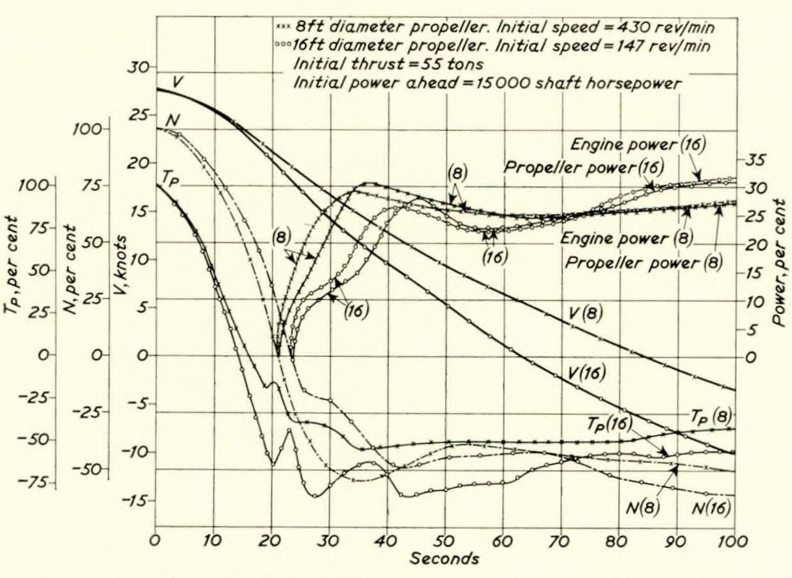


FIG. 29—*Comparison between large slow-running and small fast-running propellers*

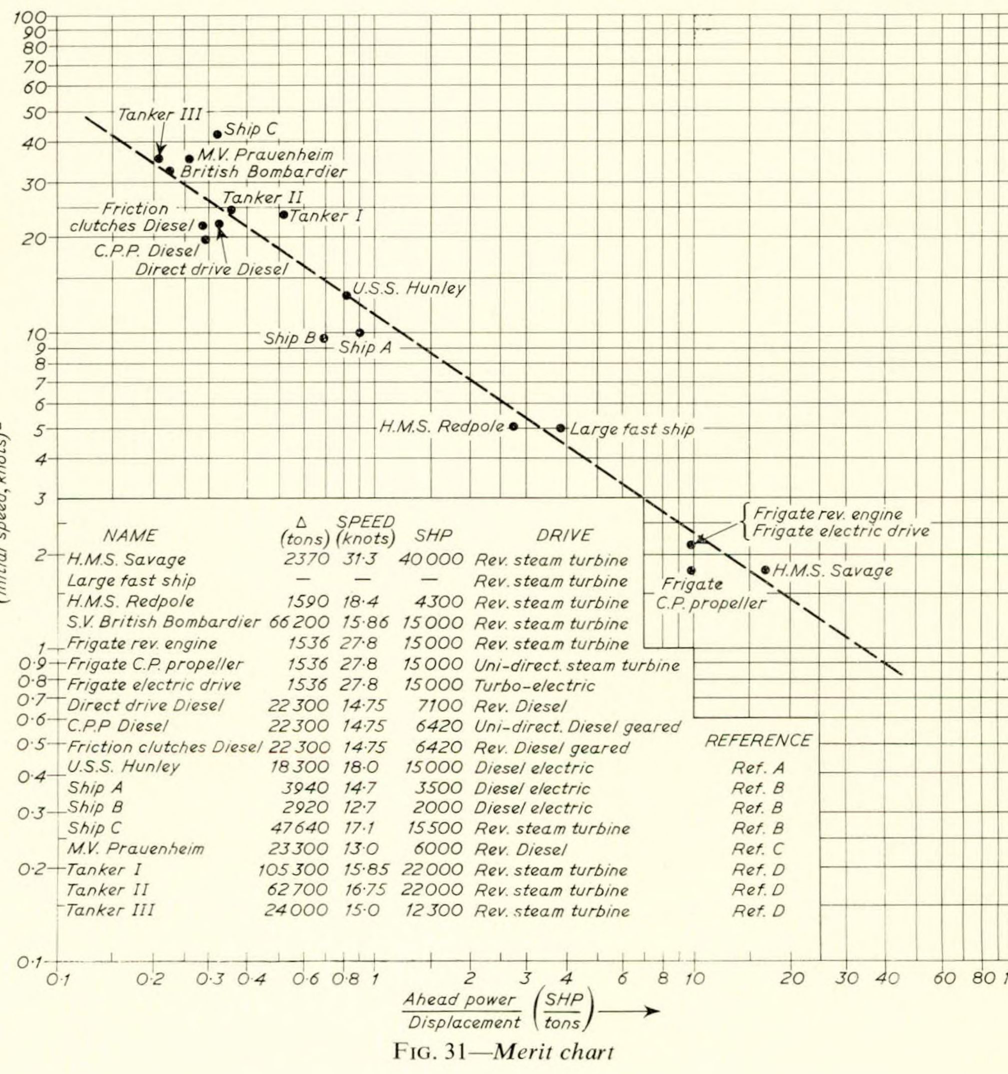
the relatively large propeller for one-sixth power, climbing over the first peak took appreciable time when only this power was used.

Mr. Lover referred to an optimum pitch ratio of about 0·8 for reversing and this could have particular significance in connexion with c.p. propellers. Mr. Slatter referred specifically to this question of optimizing the pitch throughout the reversing procedure. In a Diesel-driven c.p. propeller installation recently studied by the authors, it was found that limiting the astern pitch to such an extent that the Diesels could run at full rev/min and maximum b.m.e.p. (hence at their maximum power) was less effective in stopping the ship than using a larger pitch, although the limiting b.m.e.p. necessitated using less than the full rev/min and hence less than the maximum power of the engine. It was usual with c.p. propellers to provide a maximum astern pitch less than the maximum ahead pitch, and computer methods could be used to determine the optimum value for this. Mr. Abraham had suggested that the engine speed should be kept up, while the pitch was being altered, but care had to be taken in doing this, since in fine pitch the propeller could accelerate the engine and the governor would have no control of the resulting overspeed. The procedure adopted for Fig. 16 seemed to be a prudent limit and, for Fig. 19, it was desirable to keep the Diesel engine at least up to a speed at which the maximum torque could be exerted.

In the paper where the term astern power was used, it generally referred to the power that would be developed by that quantity of steam if the ship were proceeding at a steady speed astern. At any lesser speed, the torque would be greater and the rev/min less, and since the turbine torque rose to only 1·5 times the full astern design torque at zero rev/min, i.e. zero power, it was clear that the astern power calculated from torque and rev/min must always be less than the design astern power. The horsepower referred to by Mr. Lover as "the maximum astern shp actually absorbed during braking" must be a continuously varying quantity, hence the authors' use of the steady state astern power. In Fig. 29 curves showing the power developed by the astern turbine and the power demanded by the propellers were plotted. These were not necessarily the same and it could be seen how the shaft speed increased or decreased according to which was the larger. The greater power used ineffectively by the smaller propeller was similar to the greater power used ineffectively by a c.p. propeller in too small an astern pitch.

Mr. Lover had suggested that the discrepancy between the power at the propeller and the power that the same quantity of steam could produce at steady speed astern was due to the use of incorrect astern gear ratios. It was of course only in the case of reverse gears that there was independent control over the gear ratio, though it certainly seemed probable that some improvement in the manoeuvring performance could be obtained by selecting an astern gear ratio different from the ahead ratio; it might limit the maximum astern speed available but that was unlikely to matter.

The question of the relative merits of different means of reversing had inevitably arisen out of various contributions to the discussion and the authors now considered that this could best be examined by a plot of stopping distance divided by the square of the initial full speed, against the full ahead power divided by the displacement. Engines had to be provided for the full ahead power and the merit of reversing depended on how well these



NAME	Δ (tons)	SPEED (knots)	SHP	DRIVE	REFERENCE
H.M.S. Savage	2370	31·3	40 000	Rev. steam turbine	
Large fast ship	—	—	—	Rev. steam turbine	
H.M.S. Redpole	1590	18·4	4300	Rev. steam turbine	
S.V. British Bombardier	66 200	15·86	15 000	Rev. steam turbine	
Frigate rev. engine	1536	27·8	15 000	Rev. steam turbine	
Frigate C.P. propeller	1536	27·8	15 000	Uni-direct. steam turbine	
Frigate electric drive	1536	27·8	15 000	Turbo-electric	
Direct drive Diesel	22 300	14·75	7100	Rev. Diesel	
C.P.P Diesel	22 300	14·75	6420	Uni-direct. Diesel geared	
Friction clutches Diesel	22 300	14·75	6420	Rev. Diesel geared	
U.S.S. Hunley	18 300	18·0	15 000	Diesel electric	Ref. A
Ship A	3940	14·7	3500	Diesel electric	Ref. B
Ship B	2920	12·7	2000	Diesel electric	Ref. B
Ship C	47 640	17·1	15 500	Rev. steam turbine	Ref. B
M.V. Prauenheim	23 300	13·0	6000	Rev. Diesel	Ref. C
Tanker I	105 300	15·85	22 000	Rev. steam turbine	Ref. D
Tanker II	62 700	16·75	22 000	Rev. steam turbine	Ref. D
Tanker III	24 000	15·0	12 300	Rev. steam turbine	Ref. D

FIG. 31—*Merit chart*

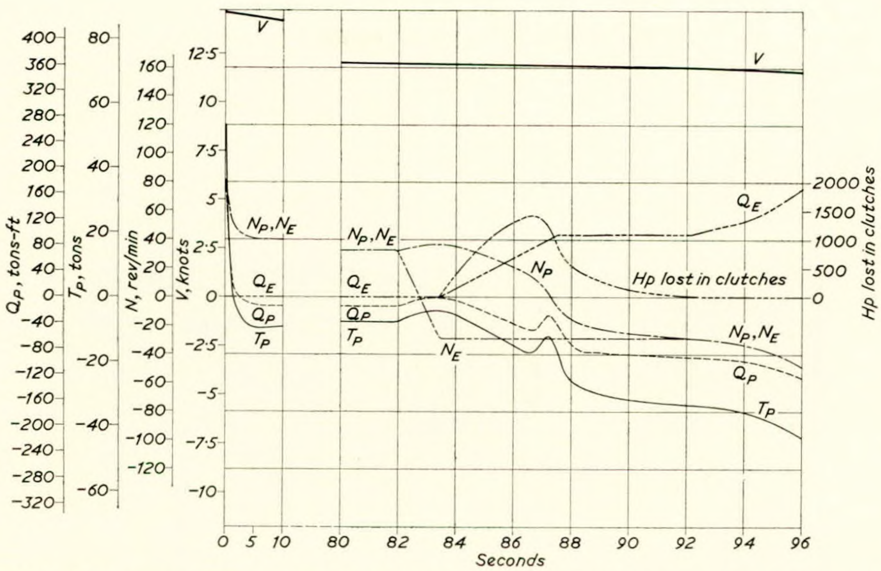


FIG. 32—*22 300 ton merchant ship—Reversing with friction clutches—Full ahead to full astern*

were utilized by the system as a whole, i.e. a combination of the astern power available, the time to apply it and the propeller design. Fig. 31 showed the results in the paper, plotted on this basis together with a number of results from other sources. These were tabulated in Fig. 31 where the References A, B, C, D were:

A) Burford, H. M., Koch, R. L., and Westbrook, J. D., 1962. "Performance of a Diesel Electric A.C. Propulsion Plant based on the Design and Sea Trials of U.S.S. *Hunley*". SNAME, Hampton Roads Section, October.

B) von Möckel, W., and Hattendorf, H. G., 1966. "An Investigation of Emergency Stopping Manoeuvres". *Schiff und Hafen*, Vol. 18, May.

C) Hebecker, O., 1961. "Stopping from Full Ahead using Full Astern Power". *Schiff und Hafen*, April.

D) Shell International Marine Ltd. "Handling Characteristics of Tankers".

A line was shown in Fig. 31 representative of the steam ships *Savage*, frigate, large fast ship, *Redpole* and *British Bombadier*, and, on this showing the latter's performance seemed very adequate, in fact, a lot better than Ship C reported by Möckel and Hattendorf, and Shell tanker I. Shell tankers II and III lay almost on the line.

In deference to Messrs. Bilton, Richardson and Sinclair, and Dr. Shannon, no point had been shown on the merit chart for frigate reversing gears since, as they all said, it would have been possible to design them to give an even better performance than the c.p. propeller. In any particular case, value engineering would determine the best design point.

M.V. *Prauenheim*, referred to in Mr. Sinclair's comments, certainly seemed to have a poor stopping performance for a Diesel engine with 75 per cent ahead power available for reversing, despite the remarkable final decelerations produced by yawing. In the early stages and even with the propeller stopped the deceleration seemed to be less than one would have expected from the ship's resistance alone. Initially, the retardation should have been over 2 kn/min as in curve B, and even at 9 kn, retardation from resistance alone should have been over 1 kn/min without any contribution from the stopped propeller. Accurate measurement of ship's speed always presented difficulty.

Mr. Lea's contribution on Diesel manoeuvring was particularly helpful, showing how, with an established propeller characteristic, there was a unique ship's speed at which the engine at a selected speed could provide sufficient torque to sustain and increase its speed, after the clutches were re-engaged. Discussions had in fact taken place between the authors and Mr. Lea which had shown that Fig. 20 would lead to stalling and, in presenting the paper, a revised diagram for this was shown and was reproduced as Fig. 32. This showed a delay of 80 s before carrying out the manoeuvre at idling speed. This delay could of course have been substantially reduced if the engine speed before re-engagement was increased to make a larger b.m.e.p. available. The authors were pleased to note that Mr. Lea now agreed that shaft brakes were not worth including.

The argument about a unique speed applied equally with direct drive reversing engines, but the medium speed geared Diesel engine showed up slightly better on the merit chart because of its larger propeller and lower power required for full speed ahead.

There were several questions concerning permissible overtorque. Naval installations were based on the fact that overtorques in certain items had been survived in the past before it had been possible to measure them and one, therefore, had guidance from previous practice. As Dr. Shannon would agree, the frontiers of knowledge were sometimes pushed forward by inadvertently going beyond previous practice. Since merchant ship design criteria were usually well below naval ones, it was probably possible to survive momentarily even larger overtorques. Due to the short time involved, pitting of gears was unlikely to occur, but scuffing and tooth breakage were possible and on that account hardened and ground gears which were designed on the latter two were more vulnerable than "soft" gears designed mainly on a pitting criterion.

With regard to Mr. Sinclair's suggestion that the ahead and astern hydraulic couplings should both be filled and thus dissipate the kinetic energy of the turbine, Fig. 15 showed the excellent deceleration that the propeller could receive when divorced from the turbine and, in fact, one of the advantages of reversing gears which they shared with the c.p. propeller and electric propulsion was that turbine inertia, much the greatest part of the total inertia, did not have to be destroyed and built up again as it did with reversing steam turbines. When the authors prepared Fig. 15 they were unaware of Mr. Sinclair's new "invertible" coupling which could be used to minimize the delay time shown as about 15 seconds before effective propeller shaft braking commenced.

In conclusion the authors wished to express their gratitude to the contributors to the discussion, both verbal and by correspondence, who had thrown light on a number of doubtful points, particularly concerning manoeuvring. As far as the operator was concerned "the fastest with the mostest" was still correct, but it was clear that in the design stages, collaboration between the propeller designer and the engine designer was needed to ensure the best results.