BRIDGE CONTROL OF MARINE STEAM TURBINE PLANT, AN ANALYTICAL APPROACH

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The introduction to the paper outlines the reasons why bridge control is considered desirable for large tankers. The subsequent section describes how a new design concept was applied in the development of a pneumatic bridge control for s.s. *Myrina* and other ships, built from standard components and adaptable to different designs of turbine.

Operating requirements are stated and the important question of machinery constraints discussed before consideration of the basic design. An analogue computer simulation model of a ship, including the boiler, turbine, gearing and propeller, was used to examine the dynamic behaviour of the main machinery, and to establish the required characteristics of the control system, the main elements of which were included in the model. This model is discussed, one of the main points emerging from this study being the need to design a limiter to prevent constraints being exceeded when large steps in rev/min are ordered from the bridge. A further important point was the desirability of having manoeuvring valves characterized to allow the use of a standard controller in a non-linear system.

Subsequent sections describe the actual system constructed and installed in the s.s. *Myrina,* and an account is also given of the shipboard trials which were very satisfactory.

Finally, comment is made on future requirements and possible design trends for turbine bridge control systems.

INTRODUCTION

Development of bridge control systems for turbine driven ships has been rapid over the past few years and a number of turbine ships are now equipped with some form of remote control from the bridge.

Early experience in the use of bridge control was obtained by Société Maritime Shell in three ships, s.s. *Sitala, Sivella,* and *Dolabella.* The first remote control systems, fitted in *Sitala* and *Sivella,* were simple valve positioning controls. Later, an advanced design of system, based on the closed loop principle and built from electronic components, was fitted in *Sivella* and in *Dolabella*[1, 2]. Experience with this system has provided valuable background in the development of the pneumatic system described in this paper.

All Shell ships are now being fitted with bridge control because:

a) a quicker and more consistent response to bridge orders is obtainable and this facilitates manoeuvring and berthing operations;
b) a more precise speed control is possible and this is valuable on occasions when keeping station;
c) it is essential that the bridge officer should have control of the main engine when operating with the main machinery spaces unattended;
d) the engineer is freed from his positions at the controls.

As experience with the French ships showed, designing for remote control of a turbine driven ship necessitates taking into account a number of important basic requirements. These are dealt with in this paper as also is the design philosophy which has been followed in the development of the pneumatic system for s.s. *Myrina* and other new ships in the Shell Fleet. Use is made in the text of certain terms peculiar to control engineering. For those perhaps unfamiliar with such terms a brief explanation is included as an appendix to the paper.

DESIGN CONCEPT

General

Steady progress is being made towards unattended operation of steam turbine ships. An essential part in achieving this is the careful and rational study of the design and engineering requirements of control systems. Such a study was carried out by the authors' company some two years ago when the usual practice was to purchase equipment complete with instrumentation as a package deal.

This purchasing arrangement was considered unsatisfactory for a variety of reasons. One of the most serious disadvantages was that selection of individual items of engine room equipment was made from a number of different manufacturers. This resulted in a variety of makes of instruments being fitted. Maintenance was therefore more complicated and an unnecessarily large range of spare components had to be carried on board. The study showed

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that by rationalization and standardization of instrumentation, this problem could be largely alleviated and the total cost of equipment reduced. In addition, consistency in control loops, panel layouts and types of instruments would facilitate the training of engineers in instrument calibration and general maintenance.

As far as design was concerned, it was considered that the designer of the propulsion plant should work in close co-operation with the process control engineer and *vice versa*. A modern main steam turbine plant is an integrated unit and should be considered as such when designing control systems for it. As a first step, therefore, the control and instrumentation requirements for the entire propulsion plant of the ships in the present newbuilding programme were reviewed with what was regarded as a new system design philosophy.

The propulsion plant, including boiler, main engine and auxiliaries was treated as a complete process into which various control loops were to be built. The design approach was similar to that used in the petrochemical industry where the main complex is broken down into sub processes or systems, each of which has its own control requirements. Interaction of the various systems was studied and taken into account as far as possible in the design of applied control loops.

Each loop was developed from consistent rules; standard symbols and identifications were used in diagrams to avoid ambiguity when ordering or discussing equipment.

The main turbine/gear/propeller system was included in the study and it is the development of the control for this system which is the subject of this paper.

Operating Requirements

From the outset there was close contact between plant designer and control engineer and the first objective was to establish the basic requirements of control. These are reviewed below:

1) During manoeuvring the conventional method of controlling ship's speed from the bridge is by means of telegraph orders relayed to an engineer who is standing by the engine controls continuously. On ocean passage, orders for speed change require the watchkeeping engineer to man the controls regardless of what other activity he is engaged in at the time. The response to telegraph orders at any time is entirely dependent on the engineer and will vary according to his experience and manual skill.
   In replacing the human element, an automatic remote control system must respond to bridge orders at least as efficiently as an experienced engineer and be able to take account, as he does, of machinery constraints.
2) The bridge officer's primary task is in the handling and navigation of the ship. He is only concerned with the main machinery in so far as he expects a prompt and reliable response to engine orders. He should not be concerned with machinery constraints or any other machinery operating factor.
   The control system must therefore be such that bridge orders are issued with one action only, and recorded automatically.
3) To safeguard the main machinery in emergencies or in case of failure of the system, it must be possible at any time for the engine room to take over control from the bridge. This must be a "bumpless" transfer. If a single lever on the bridge is used for control of the main engine as well as for use as a telegraph in emergency, the telegraph part of the system must remain operative regardless of any faults in the control system.
4) Fine adjustment of rev/min must be possible when required, e.g. when keeping station such as during a canal transit.
5) Emergency astern power should be obtainable by one simple action, but with precaution against inadvertent operation.
6) A remote control system should replace the engineer completely. It must, therefore, be capable of providing a turbine blasting sequence during stand-by and of executing automatically the routine normally carried out by the engineer when transferring from "full ahead" to "full away" condition or when reducing power from "full away".

Machinery Constraints

The subject of boiler, turbine and gearing constraints was discussed with both manufacturers and operators. Opinions appeared to differ appreciably as to what were considered reasonable manoeuvring rates and there was some difficulty in establishing precisely what constraint limits should be applied. For optimum response of a turbine remote control, it is important to have reasonably accurate information and there appears to be scope for further study of this question. Possible constraints in the main propulsion plant can be considered in two sections, the boiler and the turbine/gearing/propeller system.

Boiler Constraints

Manual operation of turbine manoeuvring valves is carried out with due consideration for boiler steam pressure and drum level. These are kept within arbitrary limits based on the operator's experience of what is acceptable. The two events which should not be allowed to occur are lifting of safety valves due to excessively rapid shut-down and carry-over due to opening up too quickly. Even with automatic boiler controls, the thermal inertia of the boiler and the swell or shrink of water cannot be fully taken into account and these are the limiting factors in so far as the boiler is concerned. They define the maximum opening or closing rate for the manoeuvring valves and the bridge control system must be so designed as to avoid unrealistic demands on the various boiler designs. The rate of response of the control loop to a bridge order must, therefore, be adjustable.

Firing rate was considered as a possible further constraint. However, with modern marine boilers having little exposed refractory, it is generally believed that this need not be a limiting factor.

Turbine, Gearing and Propeller Constraints

The following points appeared valid:

i) During rapid manoeuvring the accelerating torque applied to the gearing can exceed that at normal full power as also can the root bending stresses in the turbine blades. However, these factors are taken into consideration in the design of the turbine and gearing and need not be considered serious limiting factors. Nevertheless, too violent swings in applied torque are undesirable and some restraint was considered necessary. More important are the marginal lubrication conditions between the gear teeth when changing direction of rotation. Under such conditions there is a risk of low speed scuffing occurring and again, therefore, some restraint in acceleration and deceleration rates was considered a necessary feature of the bridge control.
ii) Application of astern power while the propeller is trailing ahead, or prolonged use of astern power at low speed astern could result in excessive differential expansion of the turbine casing. Generally, turbine manufacturers did not appear to consider this a serious limitation. One manufacturer quoted a safe limit as half-an-hour full power astern. It was decided to avoid any possibility of trouble by incorporating a logic circuit to prevent opening of the astern steam valve until the propeller rev/min, and hence ship's speed, had dropped to a predetermined value. This circuit, also, was made adjustable.

iii) When working up from "full ahead" to "full away" conditions, it is desirable to attain thermal equilibrium of the turbine at a controlled rate and here again different opinions were given as to what this rate should be. The bridge control was, therefore, designed to incorporate a time delay for this operation, adjustable over wide limits.

iv) Propeller ventilation (drawing of air into the propeller) or cavitation can occur if high astern revolutions are applied when the ship has been proceeding ahead at appreciable speed, say in excess of 6 knots. To take account of this it would be necessary to make use of ship's speed as a control parameter.
As this still cannot be measured accurately over a wide range, the idea of controlling propeller effects was not followed up with the particular system under discussion.

Basic Design Considerations

It is important that a bridge control system be designed as an integral part of the engine and not some sort of attachment. As such, it should receive the same priorities for attention by engineers as other control equipment in the ship. However, on board ship the real or apparent complexity of the bridge control can lead to reluctance on the part of engineers to rectify faults and make adjustments, particularly if they have not had the benefit of some training in control and instrument engineering or there is insufficient literature on board to help.

To satisfy owners and engineers, the bridge control must be as reliable and simple as the specified requirements and constraints permit. At the same time the cost must be reasonable and not represent too high a proportion of the total for the main machinery. At the moment the cost of electronic instruments is still some 30 per cent higher than their pneumatic equivalents. Since the particular advantages of electronic equipment were not considered to be of paramount importance for turbine control, it was decided to use pneumatic components for the first design to be fitted in s.s. *Myrina* and subsequent ships in the newbuilding programme. In these ships, all the instrumentation is pneumatic, components being purchased wherever possible from one manufacturer. Again, therefore, it was logical and in keeping with the new design philosophy also to use pneumatic components for the bridge control.

Control Loop

Marine turbine plant can be controlled either by valve positioning (open loop) or by rev/min feedback control (closed loop). The valve positioning method has certain fundamental disadvantages, the main two of which are:

a) The relationship between valve position and rev/min is not constant. For a particular bridge order setting, appreciable changes in rev/min can result from changes in steam supply conditions or from external effects such as rudder angle or depth of water under the keel. Such variations can be allowed for only by the bridge officer and this is clearly undesirable.

b) Small movements ahead or astern cannot always be executed with one movement of the bridge control lever. It is often necessary to overshoot the desired rev/min setting in order to overcome system inertia and then reduce to the setting required.

Sea experience with the original valve positioning systems in the s.s. *Sitala* and *Sivella* confirmed the above disadvantages, frequent changes in control settings being found necessary to maintain reasonably steady rev/min.

A rev/min feedback control system does not suffer these limitations and controls the speed at the desired value irrespective of changing conditions.

It was therefore decided to adopt this method and the basic considerations discussed hereafter refer to a closed loop rev/min, feedback system.

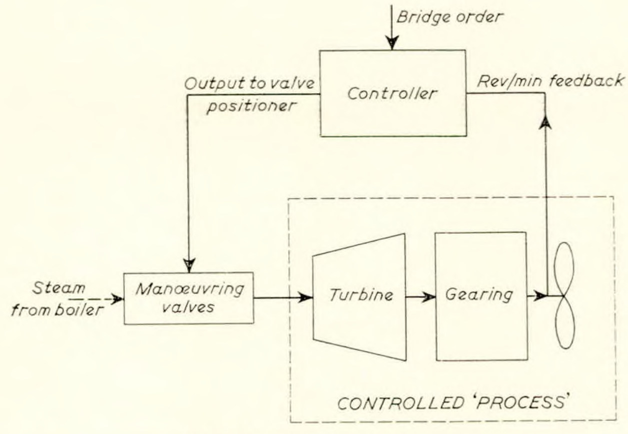

FIG. 1—*The basic loop*

Fig. 1 illustrates the basic feedback loop in which the turbine/gear/propeller unit is considered as the "process" to be controlled. If a feedback loop is to be stable under all conditions it is necessary for the loop gain to be sensibly constant throughout the controlled range, i.e. the corrective action must be linearly related to the error which causes it. The relationship between turbine steam flow and rev/min is an immutable cubic function and if a manoeuvring valve having a linear characteristic is used the resulting process characteristic would be non-linear. To achieve stable operation when using such a valve it would be necessary to incorporate somewhere else in the control loop a function generator giving approximately the inverse of the cube law.

A neater solution is to provide a control valve specially characterized to cope with a non-linear process and in this case discussion with a leading control valve manufacturer resulted in selection of valves having "equal percentage" characteristics. Two such valves were chosen, operating in

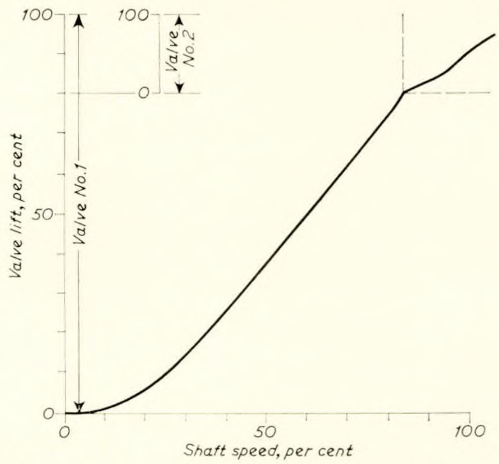

FIG. 2—*Valve lift versus propeller shaft speed characteristic of combination of two "equal percentage" manoeuvring valves*

split range, to cover ahead rev/min. Fig. 2 illustrates the combined characteristic of the valves and process cubic law. The process is sufficiently linearized to ensure good control over the whole range of ahead rev/min. A similar "equal percentage" valve was selected to cover astern rev/min. For the sake of contrast, Fig. 3 shows the combined characteristic of a conventional manoeuvring valve and process cubic law as actually measured. It is clear that any attempt to control this with a standard controller would almost certainly result in instability at some part of the range and poor control in another.

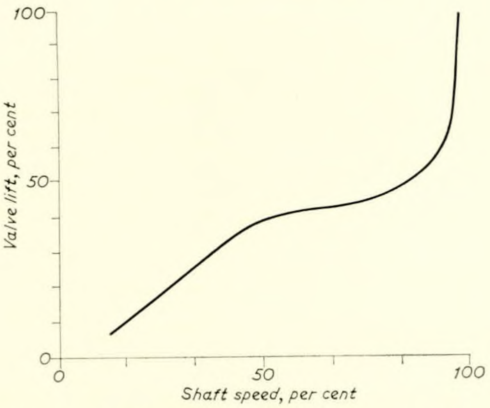

FIG. 3—*Valve lift versus propeller shaft speed characteristic of a conventional manoeuvring valve*

Delay and Derivative Units

Use of "equal percentage" valves meant that a simple standard controller could be used without fear of instability in the control system. However, it was realized that if the system were to have a fairly fast response, i.e. high gain, operating constraints would be exceeded if the bridge ordered large changes in rev/min. A unit was therefore designed to automatically limit the difference between set value (SV) and measured value (MV) to an acceptable value, and also to control the rate of increase or decrease in power between "full ahead" and "full away".

Another point was possible overshoot in the rev/min control resulting from the time lag in system response due to rotational inertia. A derivative unit was therefore incorporated in the rev/min measured value line to the controller, the effect being to cause the controller to respond not only to the measured value but also to its time derivative or acceleration. Thus overshoot would be prevented by anticipatory action of the controller. The derivative unit was deliberately positioned in the measured value line as opposed to being incorporated in the controller itself where it would act upon the error signal and hence have a derivative influence on the ordered rev/min signal.

Rev/min Transmitter

One of the most important control parameters in a rev/min feedback system is the measured value itself. For optimum reliability it was decided to use two rev/min transmitters, both operating continuously, with a differential alarm to register warning if the two signals differed by more than 5 rev/min. The units selected were pneumatic transmitters, capable of bi-directional operation, and driven off the intermediate shaft of the main reduction gearing.

Further study showed that the same rev/min transmitter signal could also be used for automatic opening and closing of drain valves, bleed steam valves and astern guardian valve when changing from manoeuvring to "full away" conditions and *vice versa*.

Telegraph System

To complete the control system the generation of the bridge order signal had to be considered.

The solution was to modify the conventional telegraph system by linking the bridge order lever mechanically with, again for maximum reliability, two pneumatic position transmitters, the output of one of these being passed to the controller as a pneumatic signal directly proportional to the desired rev/min. If the two outputs differed by more than 5 rev/min an alarm would sound. For security reasons, the pneumatic transmitters would have to be designed such that operation was parallel with but independent of the synchro system. The latter would then never be de-energized or interfered with due to a fault in the bridge control.

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Having considered the basic design features, there remained only the ancillary circuits such as the means for automatic turbine blasting. It was at this stage, therefore, that the principles of the basic design were tested. To do this it was necessary to obtain a better understanding of the

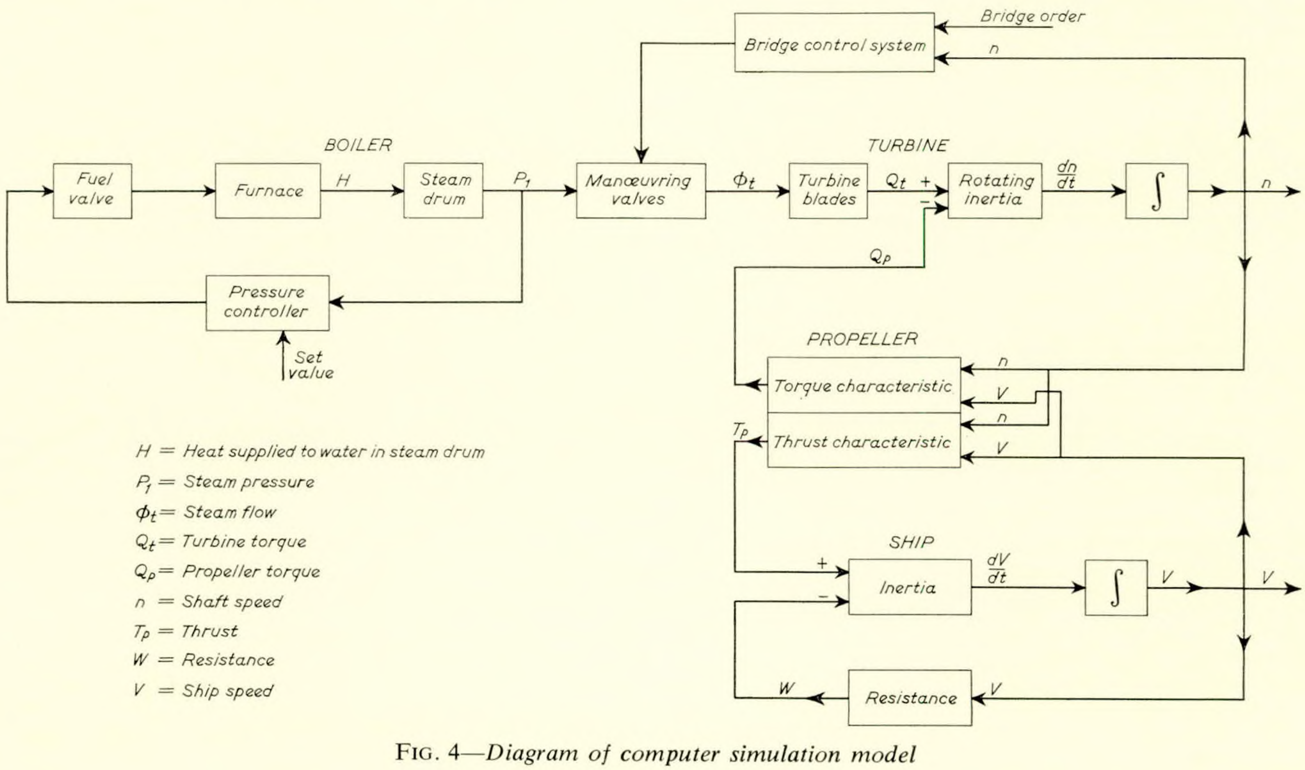

FIG. 4—*Diagram of computer simulation model*

dynamic behaviour of the main propulsion plant. The factors to be taken into account were carefully studied on an analogue computer as described in the next section. From the beginning, the object was to avoid unnecessary complication.

ANALOGUE COMPUTER SIMULATION

General

As discussed in the preceding section, with the adoption of equal percentage manoeuvring valves no great difficulty was anticipated in the design of the basic rev/min control loop. However, the requirement that the system should have fast response and yet operate within the constraint limits laid down for certain plant variables, posed something of a problem.

It was clear that to take account of operating limits by measuring all the relevant plant variables was impracticable. Quite apart from the fact that a number of these variables were almost impossible to measure, it would have rendered the control system too complicated. As discussed under "Design Concept", the simpler solution was to build into the control loop suitable means to limit the rate of change of rev/min and power. The effect of such limitations was very much dependent on the dynamic behaviour of the plant and the computer simulation model was built primarily to investigate this. In addition, the model was used to establish initial settings of the controller and the limiter, as well as to check results obtained during sea trials. For all of this work, use was made of the computer facilities available at the Amsterdam laboratories of the authors' company.

Model Description

A paper presented to this Institute in February this year[3] contained excellent information on the use of analogue computers for marine work. As the computer simulation model at Amsterdam was set up basically similar to that described in [3, 4] the description given here is brief.

Referring to the diagram in Fig. 4, each block represents a mathematical expression giving the relationship between plant variables. In order to keep the model within the capacity of the computer, a relatively simple simulation for the boiler was made. It was based on a pressure control loop because it was anticipated that difficulties in boiler operation could be expected to occur primarily due to wide pressure variations. For the simulation of the furnace, data given by Profos[5] were used. The remainder of the boiler was considered to be centred around the steam drum. This was simulated by using the heat balance of the water part of the drum and the mass balance of the steam part. A pressure controller completed the boiler model.

The heat balance described above implies that the amount of heat flowing to the "drum" is used for:

a) heating the feedwater entering the boiler;
b) heating the whole water content to a particular saturation temperature;
c) evaporating the water;
d) superheating the steam.

In mathematical form this becomes:

$$H=\phi_f\,(T_s-T_f)+M_w\,\frac{dT_s}{dt}+\left\{\phi_s r+\phi_{tot}\,C_s\,(T_{sh}-T_s)\right\} \quad (1)$$

The mass balance of the steam part of the drum is represented by:

$$(\phi_s-\phi_{tot})=C_1\cdot\frac{dP_s}{dt} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(2)$$

The relation between equations (1) and (2) above can be established by reference to the saturation line of the Mollier diagram. In the model drum level and superheater outlet temperature are considered to be controlled ideally.

The boiler delivers steam with pressure P_1 to the manoeuvring valves for which the following equation holds:

$$\phi_t = C_v\,\Psi\,P_1 \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(3)$$

The C_v factor depends on the size and type of valve and is further determined by the degree of opening of the valve. It is in effect an index of capacity and its basic value is the same for all media whether liquid or gaseous.

Ψ is a factor depending on the ratio nozzle box pressure (P_{nb})/pressure at superheater outlet (P_1) (Hengstenberk et al.[6]) and generally increases with decrease in $\frac{P_{nb}}{P_1}$. For values of $\frac{P_{nb}}{P_1}$ in a region smaller than 0·5 the value of Ψ is almost constant (about 0·6) due to reaching sonic velocity in the valve. The nozzle box pressure is assumed to be proportional with steam flow[7].

For the calculation of the turbine torque a method as described by Goodwin et al[3] was used which states that:

$$Q_t = \phi_t\,(\alpha - \beta n)$$

in which α and β are constant factors which depend on the design of the turbine. They differ also for the ahead and astern turbines.

Propeller characteristics were taken from Bindel and Garguet[8] which gives propeller torque and thrust coefficients as functions of shaft rev/min and actual ship's speed based on sea trials with a 96 000 dwt tanker. For the resistance of the ship a simple square relation with ship's speed was chosen:

$$W = C_2\,V^2 \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(4)$$

This was considered acceptable because the main objective of the project was to study the transient behaviour of the propulsion plant and not the manoeuvrability of the ship as such.

The bridge control system was also simulated; it receives the bridge order and the shaft rev/min as inputs and gives an output to the manoeuvring valve. The actual system is discussed in detail in the next section of the paper.

Results from Computer Simulation Model Study

A large number of experiments were carried out on the analogue computer to arrive at a considered view about the scheme of the bridge control system. It would be outside the scope of this paper to describe the results in detail but the most important conclusions are summarized below:

a) It was confirmed that the characteristics of the manoeuvring valves should be of the "equal percentage" type. As already discussed this choice results in a more or less constant value of loop gain over the entire rev/min range.
b) It was further confirmed that the stability of the rev/min control loop is improved appreciably if a derivative unit is incorporated in the measured value line of the controller. This was also predicted earlier and is, in fact, common to most fast acting servo positioning systems.
c) Both ahead and astern shaft speeds could be controlled by one standard controller only. This was especially checked because the ahead and astern turbines might have shown a significant difference in dynamic behaviour.
d) The rev/min controller should have proportional plus integral action. The integral action time should be small, in the order of five seconds. This had its effect on the engineering of the system because not every controller available from the instrument market offered this possibility.
e) Exceeding operating constraints could be avoided satisfactorily by installing a "limiter" which restricted the difference between set value and measured value, and hence the error signal. A smooth response was obtained to large bridge order changes without having too slow a response to small changes.

TECHNICAL DESCRIPTION OF BRIDGE CONTROL SYSTEM

General

Rather than give an exhaustive description of the system in its final form, it is intended to direct attention to those parts which are unique to this particular design. Referring to Fig. 5, the basic principles of operation will be explained

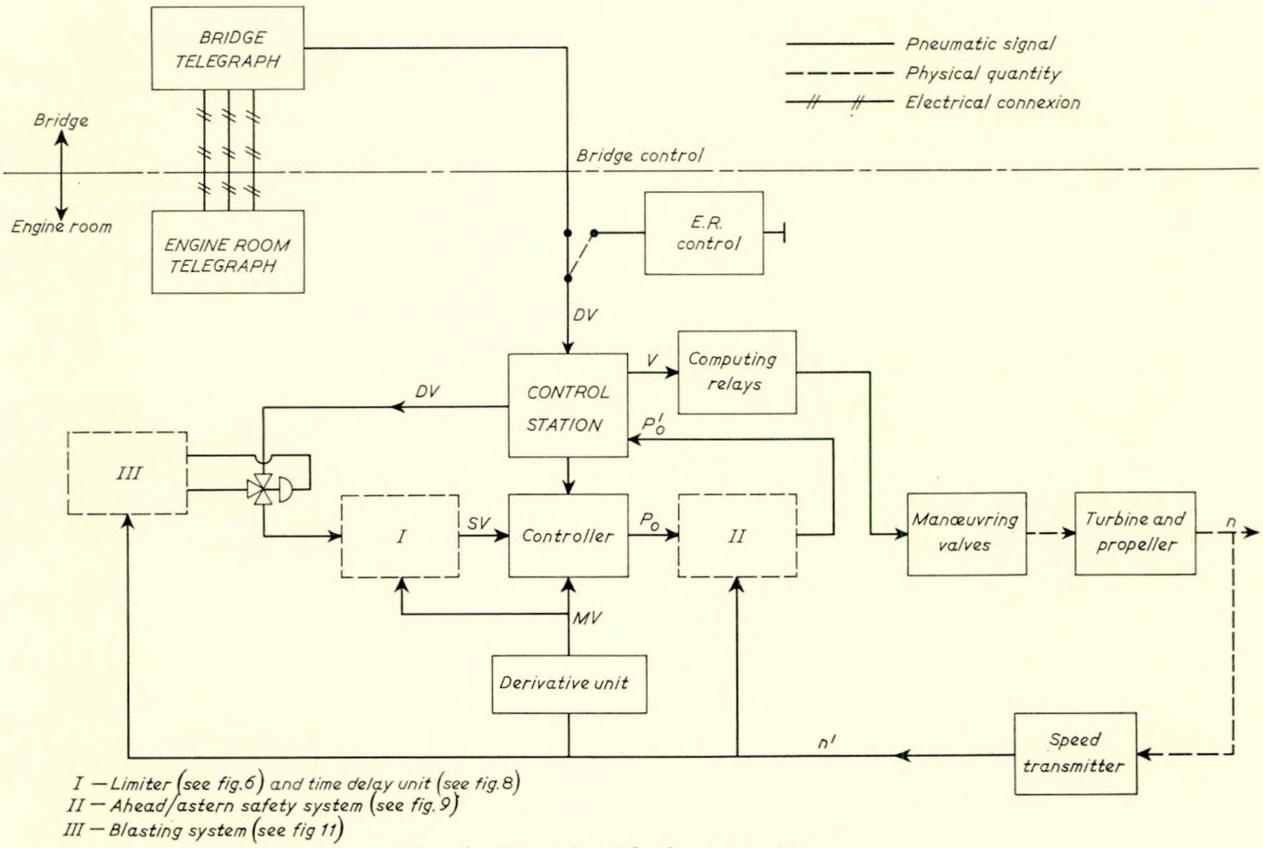

FIG. 5—*Principle of bridge control system*

followed by discussion of the particular units represented in the diagram by dotted blocks. It should be emphasized that none of the components was specially designed for this system. They are all standard pneumatic components and can be purchased from a variety of instrument manufacturers.

The system provides three modes of operation, viz: fully automatic bridge control, fully automatic engine room control and manual control.

In both the automatic modes, the heart of the system comprises a control station and controller having proportional plus integral action. The controller receives two input signals, the measured value (*MV*) and the set value (*SV*) and gives one output signal (P_o). The *MV* signal is derived from a speed transmitter which is coupled through a small gear drive to the intermediate shaft of the main gearing. An air signal of 3 lb/in²g corresponds to 80 rev/min astern, 8 lb/in²g to 0 rev/min and 15 lb/in²g to 110 rev/min ahead. Closed loop rev/min control is therefore operative throughout the manoeuvring range and until the ahead valves are fully open at a shaft speed of approximately 101·5 rev/min. For bridge settings higher than this speed, rev/min feedback is non-operative and indeed would be undesirable. In the output line of the speed transmitter a derivative unit is mounted to improve the dynamic stability of the rev/min control loop. The *SV* signal is derived from the desired value (*DV*) signal generated either on the bridge by means of a pneumatic position transmitter connected to the telegraph or in the control room by means of a pressure reducer. Transfer of control to or from the bridge can be made by means of a switch. The output signal of the controller goes through a set of computing relays to the manoeuvring valves. In the computing relays the output signal of the controller is split up into ahead and astern components, and scaled appropriately to serve both the astern and two ahead manoeuvring valves.

The control station permits change over from the automatic modes to manual, in which case the manoeuvring valves are positioned directly by an air signal generated in a pressure reducer. In order to achieve bumpless transfer during auto/manual switching the connexion from controller output to controller feedback (*FB*) goes via the control station. Thus it can be seen that the system does not differ in principle from any other control loop, for example a pressure or temperature loop. The following paragraphs describe the units which had to be included to meet the particular requirements laid down under "Design Concepts".

Limiter

The delay unit, already mentioned and indicated in Fig. 5 by block I comprises two main parts, the limiter and the time delay unit.

The purpose of the limiter is to restrict the value of *SV*-*MV* (error signal) going to the controller. This is in order to keep boiler pressure and level variations, and turbine torque, within safe limits by preventing too high a rate of change of power during manoeuvring. In a standard pneumatic controller the error signal cannot be readily modified and therefore the solution shown in Fig. 6 was chosen in which the value of *DV* is itself modified. The limiter consists of a high pressure selector, a low pressure selector, two computing

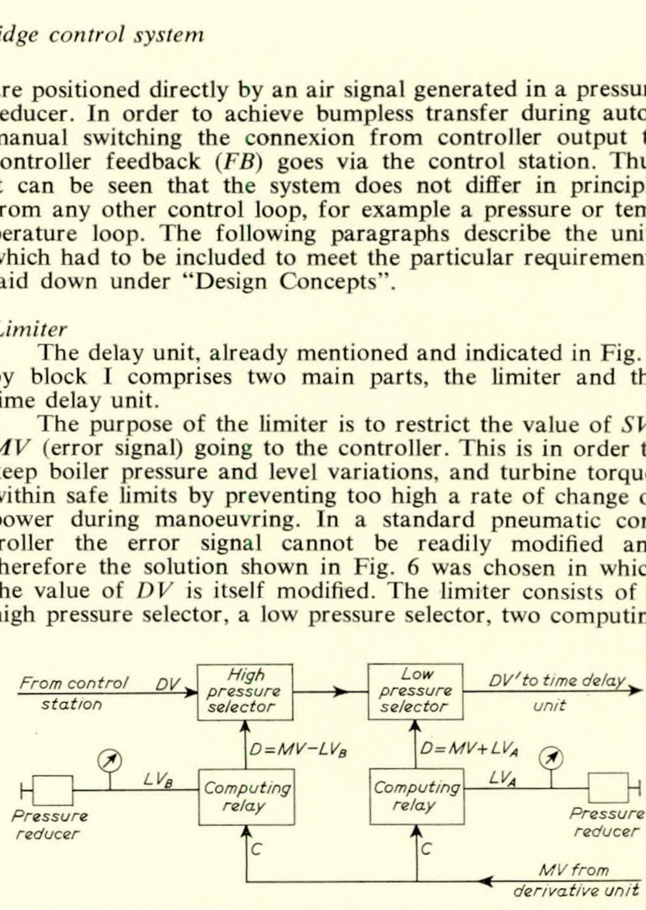

FIG. 6—*Layout of limiter (block I in Fig. 5)*

relays and two pressure reducers. One of the inputs of the computing relays is formed by the *MV* signal after the derivative unit, the other inputs being given by the pressure reducers.

The computing relays can perform the following mathematical equation

$$D = K(A - B) + C$$

In the above, *A*, *B* and *C* are input signals, *K* an adjustable multiplication factor, in this case = 1, and *D* the output signal. In the first computing relay, associated with the high pressure selector, $A = O$, $B = LV_B$ and $C = MV$ so that it delivers an output $(MV - LV_B)$. The second, associated with the low pressure selector, has inputs $A = LV_A$, $B = O$ and $C = MV$, and delivers $(MV + LV_A)$. LV_B controls the rate of decrease in rev/min and LV_A the rate of increase.

When decreasing rev/min, the *DV* signal goes first to the

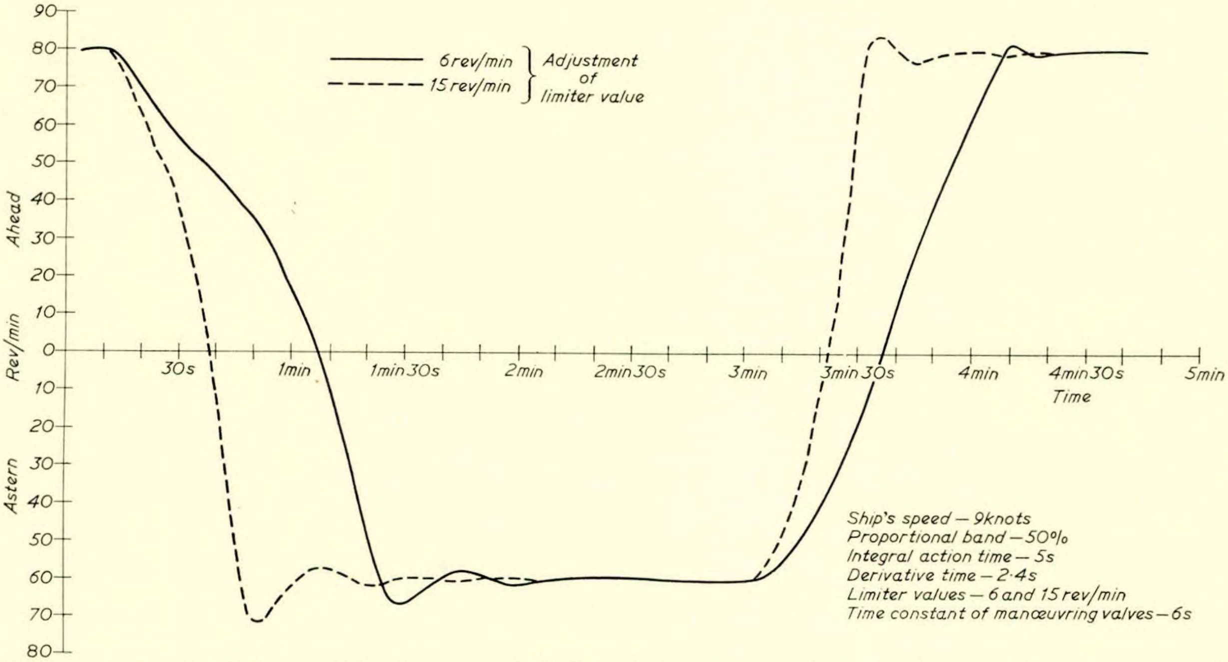

FIG. 7(a)—*Computer simulation model—Response of shaft rev/min to step in bridge order from 80 rev/min ahead to 60 rev/min astern*

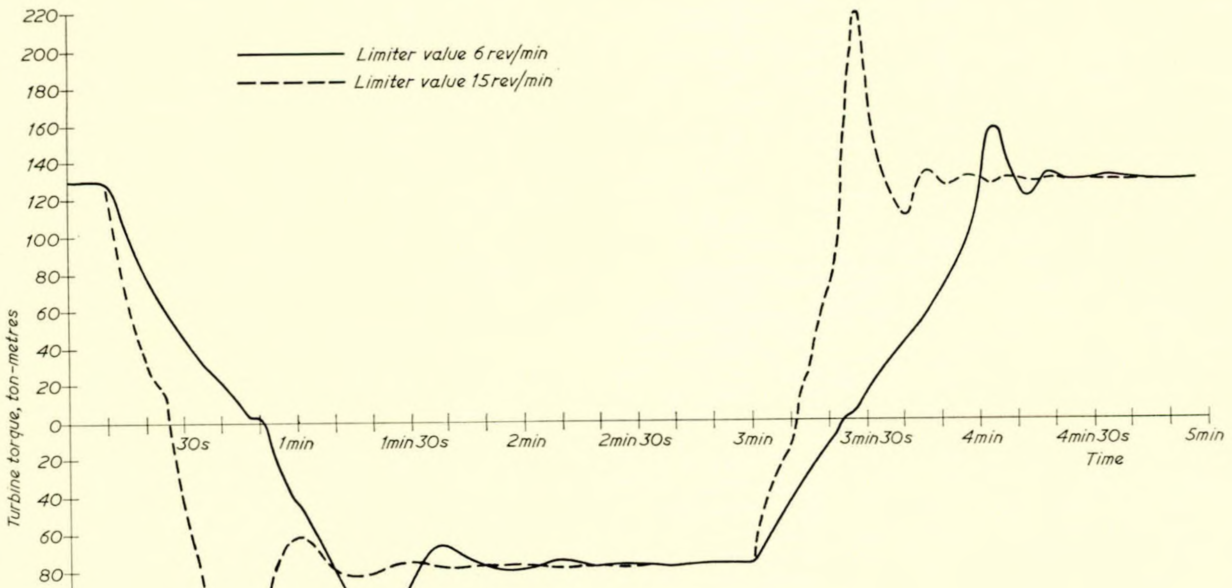

FIG. 7(b)—*Computer simulation model—Response of turbine torque to step in bridge order from 80 rev/min ahead to 60 rev/min astern*

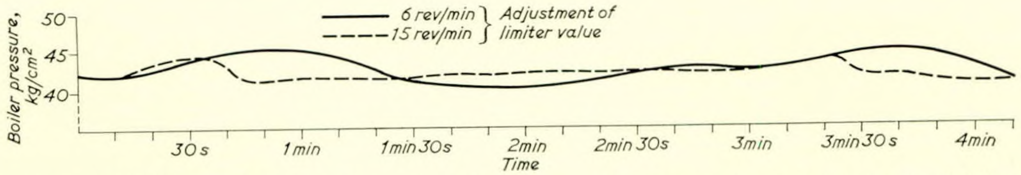

FIG. 7(c)—*Computer simulation model—Response of boiler pressure (after superheater) to step in bridge order from 80 rev/min ahead to 60 rev/min astern*

high pressure selector. If greater than $(MV - LV_B)$ it passes unchanged to the low pressure selector. If smaller, it is cut off and replaced by $(MV - LV_B)$ which again passes to the low pressure selector. In either case the resultant value is smaller than the input to the low pressure selector and therefore passes straight through the latter to the time delay unit. Similarly when increasing rev/min the low pressure selector either passes DV if smaller than $(MV + LV_A)$ or, if larger, cuts it off and replaces it by $(MV + LV_A)$.

By means of this limiting device the value of the signal DV' never falls below $(MV - LV_B)$ when decreasing rev/min nor exceeds $(MV + LV_A)$ when increasing rev/min. Summarizing, the value DV' going to the time delay is equal to:

DV when $-LV_B < (DV - MV) < LV_A$

$(MV - LV_B)$ when $(DV - MV) < -LV_B$

$(MV + LV_A)$ when $(DV - MV) > LV_A$

The pressure reducers providing the values LV_A and LV_B can in principle be replaced by function generators with which it is possible to set the values of LV_A and LV_B as functions of, for example, shaft rev/min. For the sake of simplicity, constant values of LV_A and LV_B were chosen for the time being.

To illustrate the effect of the limiter some results of the analogue computer study are shown in Figs 7a, b and c. The computer simulation model was based on the 200 000 dwt tanker s.s. *Myrina*. For two different settings of the limiter: $LV_A = LV_B =$ 6 rev/min and 15 rev/min respectively, the responses of shaft speed, turbine torque and controlled boiler pressure on a change in bridge order from full ahead to full astern are given.

As might be expected the rev/min response is faster with the wider limiter setting. However, under these circumstances the overshoot in astern turbine torque is considerable. It is interesting to note that the analogue computer model provides additional valuable information about turbine torque which cannot readily be measured in practice.

The boiler pressure changes less with the wider limiter settings because then the period during which both manoeuvring valves are closed is shorter. Nevertheless, even with the narrow limiter settings the pressure variations appear acceptable.

Time Delay Unit

The output of the limiter (DV') goes to the time delay unit. This unit is operative only for shaft speeds higher than "full ahead" and was included to regulate the rate of increase or decrease in power between the manoeuvring range and "full away" condition. The layout of the time delay unit is shown in Fig. 8.

It comprises some adjustable pneumatic resistors, a volume, some valves and trip relays. When the shaft speed is smaller than 80 rev/min valve V_1 is open, the resistors (R_1) and (R_2) are by-passed and $SV = DV'$. When the shaft speed exceeds 80 rev/min trip relay (1) closes valve (V_1). As at the same time valve (V_2) is closed the DV' signal has to pass resistor (R_1). The combination of R_1 and the volume causes a time lag resulting in a response of SV on a step-change in DV (e.g. from the bridge) as shown in the graph. When the "full away" order is given, DV increases to 110 rev/min but DV' is only an amount LV_A larger than the MV due to the limiter action. The SV can change only very slowly and therefore the rev/min control loop keeps MV almost equal to SV.

A fixed pressure difference equal to LV_A will therefore be developed across R_1 and the corresponding air flow through it will increase the pressure in volume (V) at a constant rate. In this way an increase in SV of say 0·5 to 1 rev/min can be realized.

Working down the installation can be carried out faster than working up. When the ship is under full away and the DV is put at a value smaller than 80 rev/min, valve (V_2) opens and the time delay for reducing values of DV' and SV is now determined by the combination of R_1 and R_2 which will result in faster response.

Ahead-Astern Safety System

It has already been mentioned that the admission of steam to the astern turbine should be avoided when the ship has an appreciable ahead speed.

The answer, in terms of hardware, to this requirement was called "ahead-astern safety system" and this is shown schematically in Fig. 9. The system is taken up in the output line of the controller and its operation is quite simple: when the shaft speed is higher than say 30 rev/min the trip relay will force the three-way valve into the position shown and a signal corresponding to closed manoeuvring valves, e.g. 9 lb/in²g, will go to the high pressure selector. This means

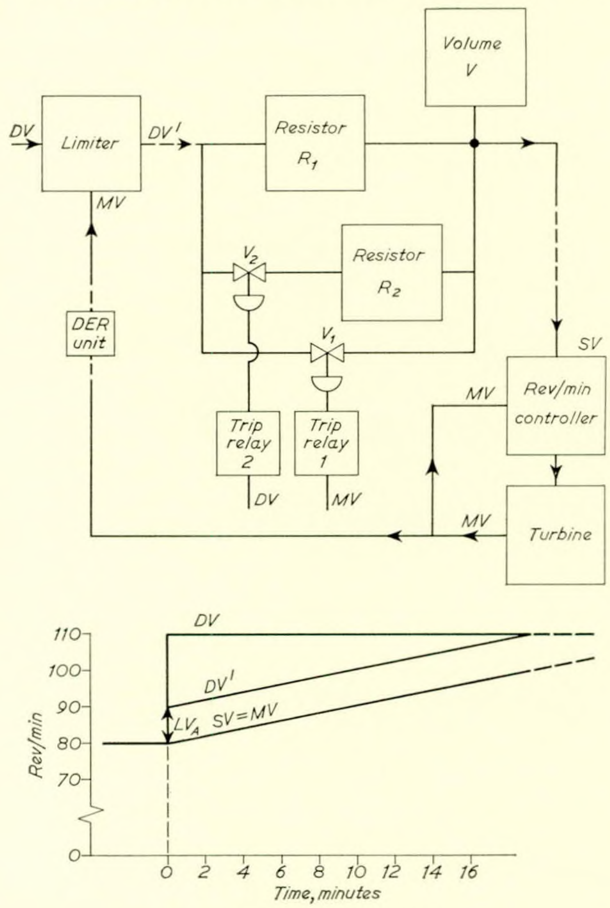

FIG. 8—*Time delay unit*

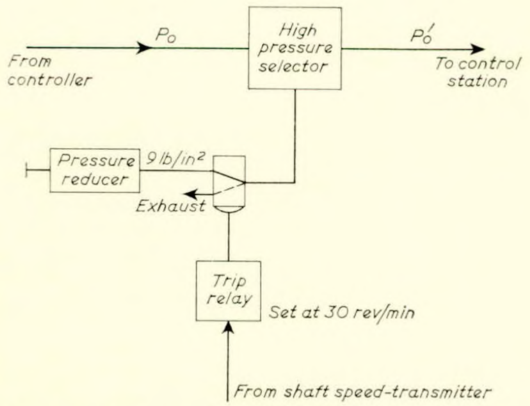

FIG. 9—*Ahead-astern safety system (block II in Fig. 5)*

that although the signal P_o can be smaller than 9 lb/in²g, P_o' cannot become smaller than this value and the astern manoeuvring valve remains closed. The propeller is now trailing and its speed is determined by the speed of the ship. When the speed of the ship has come down so far that the propeller speed becomes lower than 30 rev/min, the trip relay causes the three-way valve to take the alternate position so that the 9 lb/in²g is taken away from the high pressure selector. Then $P_o' = P_o$ and the astern manoeuvring valve is allowed to open.

The action of the ahead-astern safety system may be illustrated by giving some results from the analogue computer study. Fig. 10 gives the propeller rev/min response on a change in bridge order from 80 rev/min ahead to 60 rev/min astern for three different settings of the system: 35, 32 and 30 rev/min.

After giving the astern order the ahead manoeuvring valves close and the propeller rev/min reduces relatively quickly to a value where the propeller turns at trailing speed. If the manoeuvring valves are kept closed, propeller rev/min will then continue to be determined by the ship's speed. Especially for large tankers the further decrease in rev/min will occur slowly and the response curve will show a "knee" at about 30 rev/min. That is the reason why the ahead-astern safety system should be adjusted carefully. Adjusting at too

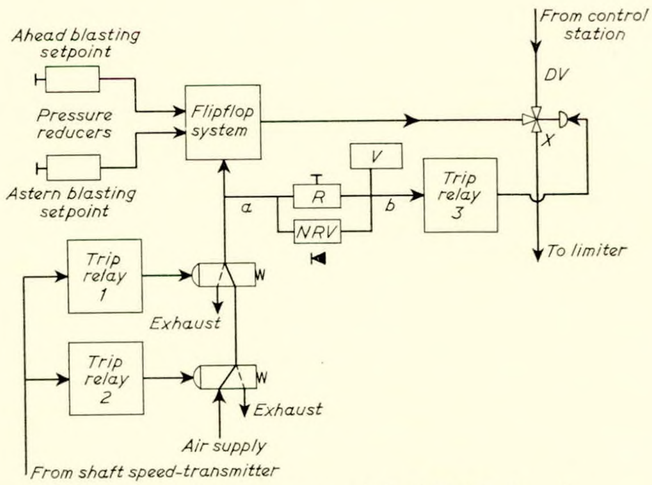

FIG. 11—*Blasting system (block III in Fig. 5)*

high rev/min would not give protection whereas adjusting at too low rev/min might result in too long a stopping time of the ship.

Information on astern turbine over-heating and propeller cavitation is scanty and therefore these phenomena have not yet been included in the computer simulation model. Thus no tests could be carried out to find the optimum setting. The setting which, when starting at 80 rev/min, keeps the astern valve closed during one or two minutes appeared reasonable. With this setting, the influence of the ahead-astern safety system on the stopping time of the ship was negligible because the resistance of the ship is the most important braking force and a full astern running propeller would most probably cavitate.

The Blasting System

After careful study of the possibilities for automatic steam blasting during stand-by, using the controller set point line was considered the simplest and safest way to realize it.

Fig. 11 gives a layout of the blasting system as it was arranged in s.s. *Myrina*. The output of the speed transmitter is fed to a set of trip relays. When the shaft speed enters the

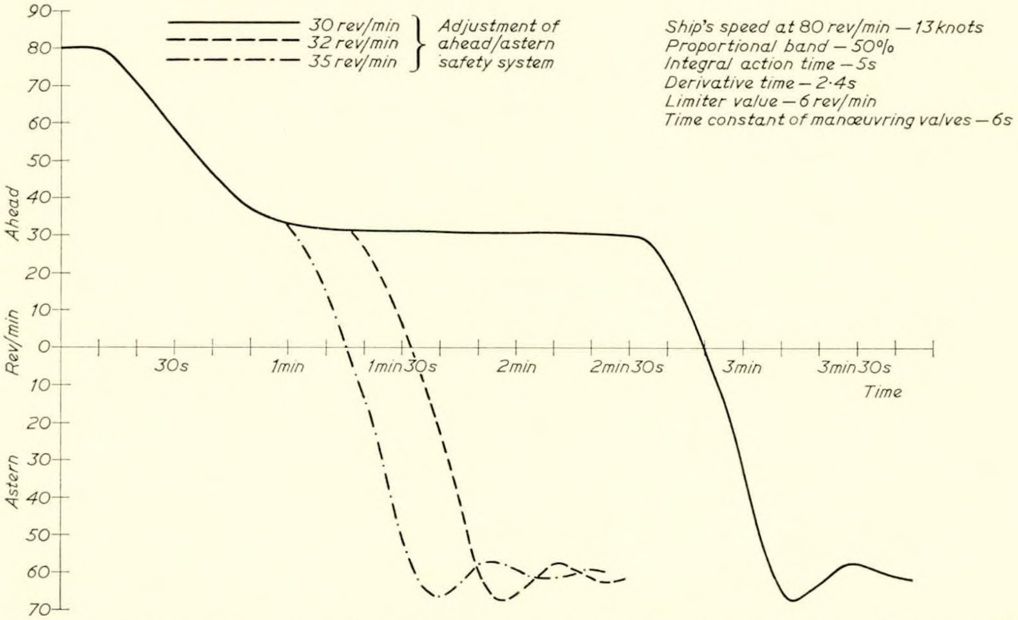

FIG. 10—*Computer simulation model—Effect of ahead-astern safety system—Response of rev/min to bridge orders step change from 80 rev/min ahead to 60 rev/min astern*

range between, say, 4 rev/min ahead and 4 rev/min astern, both trip relays are energized and the three-way valves are in the position shown. Supply air is allowed to the flip-flop system which applies sequentially either a small "ahead" or an "astern" set point signal to one connexion of three-way valve (X). Supply air is also fed to the time delay unit formed by the adjustable resistor (R) and volume (V). The time lag can be adjusted so that after, for example, three minutes, signal (b) is large enough to energize trip relay (3) which then operates valve (X). Then the normal DV is replaced by a blasting set point. When the shaft rotates, trip relays 1 or 2 are de-energized so that volume (V) can exhaust via the non-return valve (NRV) and one of the three-way valves. Then valve (X) returns to the original position and the normal DV is reconnected to the limiter. Then the cycle starts again. The next time the flip-flop system will give a "blast" in the reverse direction. When the bridge control system is put on "manual control" (via the control station) the automatic blasting system has no effect. With the latter arrangement, blasting has to be done manually.

Testing Facilities

When a number of instruments are tied together in a complex system, it is always advisable to build in suitable checking facilities because taking out the system as a whole for testing on a bench is practically impossible. Therefore a test unit was designed consisting of a number of pressure gauges, pressure reducers and a dummy turbine.

In the bridge control system itself, almost all interconnecting lines are provided with test connexions which make possible the checking and adjustment of each separate component.

The dummy turbine is nothing more than a simple time delay unit adjusted to have about the same time lag as the real turbine. By operating a multiple switch the following simultaneous actions can be taken:

a) The real turbine is put on valve positioning control.
b) The valve output (V) of the control station (see Fig. 5) is connected to the input of the dummy turbine.
c) The output of the dummy turbine goes as a rev/min signal to the derivative unit instead of the speed transmitter signal.

In this way, the rev/min control loop is closed via the dummy turbine and a quick check can be carried out on the system as a whole. This is an important feature, particularly when the ship has been under way for two weeks or so when it is possible to test the proper functioning of the system before starting actual manoeuvring.

SEA TRIALS

The bridge control system built for s.s. *Myrina* was checked and adjusted during preliminary trials in December 1967 and commissioning trials in February 1968.

The first objective was to find suitable settings for the controller to give stable performance over the entire rev/min range. This turned out to be easy, although the results were

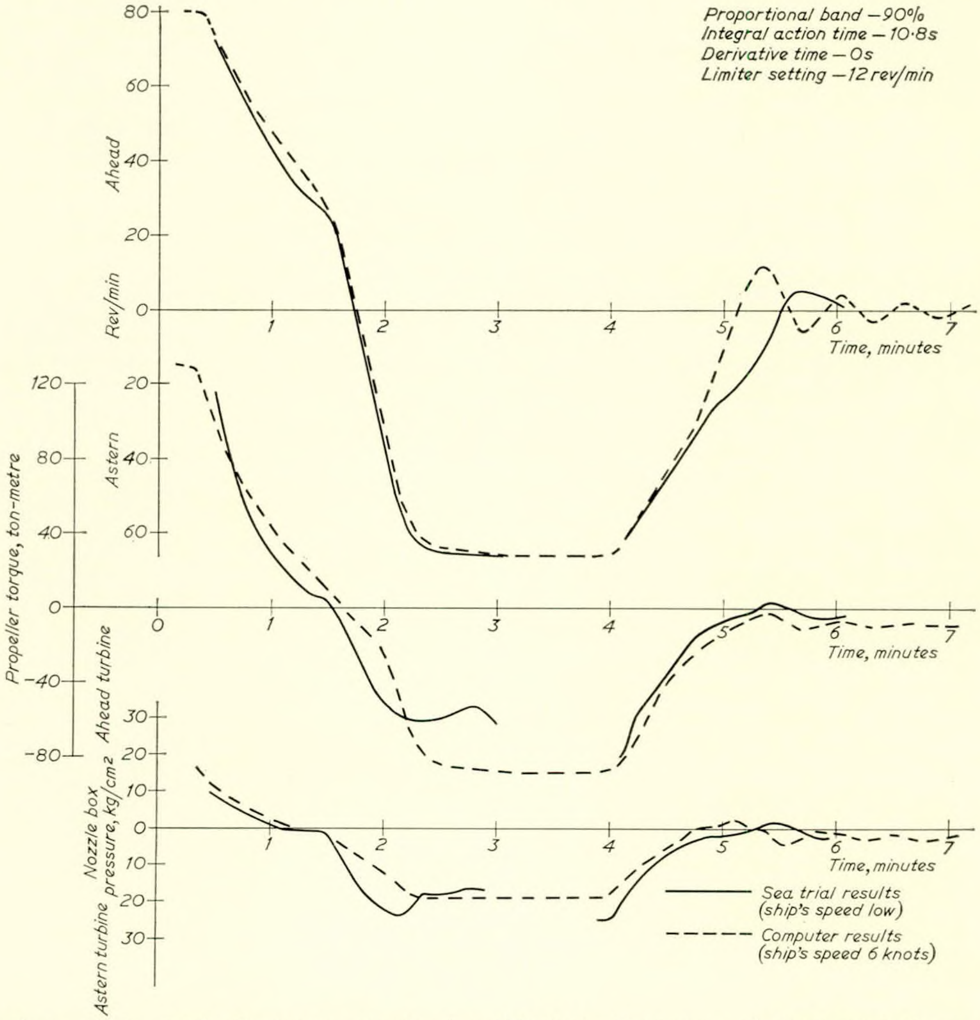

FIG. 12—*Comparison ship's trial—Computer results (s.s.* Myrina)—*Responses of shaft rev/min propeller torque and nozzle box pressure to step change in bridge order from 80 rev/min ahead to 65 rev/min astern*

slightly different from the controller settings predicted from the computer work. This was caused by two facts: first, the manoeuvring valves appeared to be slower than was anticipated; second, during the trials the derivative unit appeared to be defective and consequently was not used, this resulted in a somewhat slower rev/min control loop. It was possible to compensate for this by widening the limiter setting and this was done. A further improvement was obtained by speeding up the action of the manoeuvring valves. Optimum settings will be possible when a new derivative unit has been installed. Suffice it to say that the ship's personnel were well satisfied with the performance of the system.

During the trials, the response of the plant to changes in bridge order were measured by taking readings of selected plant variables every 10 seconds. Fig. 12 gives some of the results. The controller was adjusted at a proportional band of 90 per cent and an integral action time of 10·8 seconds, the limiter being adjusted at $LV_A = LV_B = 12$ rev/min.

From the measuring programme two typical responses have been reproduced: From 80 rev/min ahead to 65 rev/min astern and from 63 rev/min astern to stop. The full lines in Fig. 12 correspond to the shipboard measurements. The response time from 80 rev/min ahead to 65 rev/min astern appeared to be about two minutes. This is not very fast as compared with the rate at which Diesel engines can be reversed but the general feeling during the trial was that it was fast enough bearing in mind the enormous mass of a ship the size of s.s. *Myrina*. In any case, the speed of response can be improved by widening the limiter still further. Because of the low ship's speed the ahead-astern safety system did not come into operation during this manoeuvre.

Naturally, it was of interest to check whether the computer model performance could be made to correspond with the performance of the ship during sea trials. Therefore, in the computer model the manoeuvring valves were made slower (time constant 6 seconds) and the manoeuvres mentioned above were run with settings of the bridge control system equal to those prevailing during the sea trials. The responses measured on the analogue computer are also given by the dotted lines in Fig. 12. The results show that the agreement is reasonable enough to use the model as an aid for the design of bridge control systems. In particular, the rev/min responses practically coincide.

From an instrument performance point of view, the bridge control system operated quite satisfactorily, apart from some teething troubles. The equal percentage characteristic of the manoeuvring valves is indeed a particularly happy choice, not only because of rev/min control loop stability but also because of the vibration-free operation of the control valves themselves. The automatic blasting system also operated well and was greatly appreciated by the engineers.

Concerning operating constraints it can be said that the results of the sea trials related to Fig. 12 do not show an important overshoot in nozzle box pressure (and hence in steam flow and turbine torque).

Boiler pressure variations during the above manoeuvres were smaller than 1 kg/cm^2.

However, there is some evidence that when the system has been on stand-by (blasting) for a prolonged period of time, the manoeuvre to full ahead rev/min within for example two minutes is too fast for the boilers.

Also, propeller cavitation is a point which deserves further attention. Contrary to one of the conclusions in the paper by Messrs. Goodwin, Irvine and Forrest[8] and in accordance with Jaeger and Jourdain[9] the authors are of the opinion that giving full astern rev/min when the ship still has a high speed ahead is certainly not the optimum way of stopping the ship. There is no doubt that cavitation or some other propeller efficiency reducing effect can take place. By avoiding cavitation no great reduction in stopping time and distance is expected. However, when cavitation occurs, having a very large ship quivering like a wet dog for no purpose is hardly considered good practice.

CONCLUSIONS

The bridge control system described in this paper offers closed loop shaft speed control. The various adjustment possibilities of the system are sufficient to give any degree of controllability required. It is a universal system which can be built by any instrument manufacturer or shipbuilder and is not tied to specific makes of instruments.

The predictions of the analogue computer were reasonably close to the practical truth as was proven during the sea trials, and the analytical approach, making use of this technique, is considered thoroughly justified.

So far, the controller settings of the system are rather conservative and consequently, response times may not seem too impressive. However, it should be borne in mind that it is not the rate of change in shaft speed but the rate of change in ship's speed which determines the manoeuvrability of the vessel. From the sea trials discussed, the authors also have good reasons to believe that stopping a very large ship does not require a fast reversal of shaft rev/min as is often propagated, but a considered manipulation of the manoeuvring valves based on feedback of ship's speed as a control parameter. This has been touched upon in this paper and the possibility will continue to be investigated.

The bridge control system fitted in *Myrina*, being a prototype, has always been regarded as capable of improvement. In fact, improvements were foreseen before ever it was commissioned, but due to lack of time it was necessary to freeze the design at a certain stage.

A second system has now been built to control a turbine in a Japanese-built ship. In this system the improvements foreseen have been incorporated but the main principles of design have been retained.

Changes have been made in the arrangement of some components but the main improvement has been to build the system as far as possible in modular form. Another practical improvement is the incorporation of a test system as an integral part of the bridge control unit. The operator can now, at any time, check the operational conditions of the system by means of only one switch. All the ships in the current newbuilding programme have pneumatic controls and instrumentation. The bridge control systems for these ships will, therefore, also be pneumatic.

The advantages of cheap, universal components has already been stressed. Other advantages, at least in the present stage of development, include reliability and ready acceptance by marine engineers. However, with increased reliability for marine conditions and lower cost, it is clearly possible that in future complete electronic instrumentation and controls will be seen in ships and it would then be logical to include an electronic bridge control system. Because of the rational philosophy behind the design of the bridge control system there would be no difficulty in producing an electronic version. It is simply for the customer to state which he wants.

During the preliminary discussions leading to the design of the system there was a great deal of discussion about the required characteristics for the turbine manoeuvring valves. For *Myrina* the "equal percentage" control valves were chosen because they presented a simple and straightforward solution. The valves used in other ships of the present programme are not all of this type and it has been necessary to linearize the control loop by means of a function generator in the valve positioner. This provides a solution to the problem but it is not basically the correct approach. The bridge control system and manoeuvring valves should form an integral part of the turbine control to achieve optimum results. The design must, therefore, be considered in the early stages in the design of the whole plant including the boiler, boiler controls and auxiliary controls. If necessary, the concept of anticipatory or feed forward control to the boiler may have to be considered.

Finally, there is the important question of maintenance. A good design must take account of how maintenance and adjustment will be carried out on board ship, and what quality of personnel will be available to do the work. At present, there is a great deal of discussion about how ships will be manned in future. For the time being it must be assumed that endeavours to reduce manning in ships will continue but the quality of personnel may have to increase. This is a debatable point and one which cannot be discussed at length in this paper. But higher quality personnel of advanced training will more readily accept the general introduction of electronics or fluidics. Even so, this must not result in unnecessary complications resulting from insufficient study of basic requirements.

At the other end of the scale, it could be argued that ships of the future may be manned largely by untrained personnel and this could result in the need for complete automation systems, reliable and foolproof. Whatever the outcome, there will always be a need at the appropriate level for a minimum amount of maintenance, and it is important therefore that full, clear information about control equipment is provided on board. Too often, equipment goes out of service due to lack of manufacturers' maintenance instructions and particularly a clearly written fault finding procedure.

ACKNOWLEDGEMENTS

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Appendix I

SYMBOLS AND NOMENCLATURE

LV_A	=	Limit value for increasing shaft speed
LV_B	=	Limit value for decreasing shaft speed
C_1, C_2	=	Constant
C_v	=	Valve size factor, defined as the number of U.S. gallons per minute of water which will pass through the valve with a pressure drop of 1 lb/in^2.
C_s	=	Specific heat of superheated steam
DV	=	Desired value
DV'	=	Modified desired value
H	=	Heat entering the boiler
K	=	Proportional gain
MV	=	Measured value
M_w	=	Mass of water in boiler
n	=	Shaft speed
n'	=	Output of speed transmitter
P_{nb}	=	Nozzle box pressure
P_o	=	Output of controller
P_o'	=	Output of high pressure selector (AASS)
P_l	=	Pressure after superheater
P_s	=	Pressure in steam drum
Q_t	=	Turbine torque
r	=	Latent heat of evaporation
SV	=	Set value
T_f	=	Feedwater temperature
T_s	=	Temperature in steam drum
T_{sh}	=	Temperature of superheated steam
t	=	Time
V	=	Speed of ship through the water
W	=	Ship's resistance
α, β	=	Constant factors
ε	=	Deviation
τ_i	=	Integral action time
ϕ_f	=	Feedwater flow
ϕ_s	=	Evaporation rate
ϕ_t	=	Steam flow to turbine
ϕ_{tot}	=	Total steamflow leaving the drum
Ψ	=	Factor

Appendix II

CONTROL ENGINEERING TERMS

For automatic control of a plant variable four elements are essential: the process, the measuring element, the controller and the correcting element.

These four elements form together the "control loop" a block scheme of which is given in Fig. 13. Each block represents a mathematical operation which has to be applied to the input(s) to obtain the output.

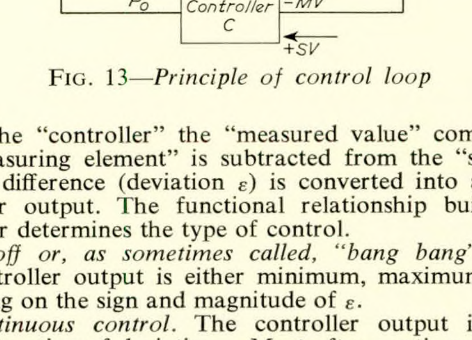


Fig. 13—*Principle of control loop*

In the "controller" the "measured value" coming from the "measuring element" is subtracted from the "set value" and the difference (deviation ε) is converted into a suitable controller output. The functional relationship built in the controller determines the type of control.

On-off or, as sometimes called, "bang bang" control. The controller output is either minimum, maximum or zero depending on the sign and magnitude of ε.

Continuous control. The controller output is a continuous function of deviation ε. Most often continuous control is applied in one of the following forms:

1) Proportional control; the controller output is directly proportional to ε i.e. $\Delta P_o = K \varepsilon$.

2) Proportional plus integral control; the controller output has a proportional component plus the time integral of ε i.e. $\Delta P_o = K\,(\varepsilon + (1/\tau_i) \int \varepsilon\, dt)$.
The "proportional gain" K (generally the inverse of "proportional" band) and the "integral action time" can be adjusted in the controller.

Proportional Control

Suppose that in Fig. 13 the functions of all blocks are constant. The change in MV as a result of change in SV can be calculated:

$$\Delta\, MV = \Delta\, SV \; \frac{CVPM}{1 + CVPM}$$

The product CVPM is called the "loopgain" of the system. The above formula shows that with proportional control $\Delta\, MV$ never equals $\Delta\, SV$. The higher the loop gain the closer $\Delta\, MV$ approximates to $\Delta\, SV$.

However, problems of "stability" impose a limit to the practical value of the loop gain caused by the imperfect responses of the elements in the loop. This is illustrated in Fig. 14 giving the response of MV on a step change in SV for different loop gains. At high loop gain the response is fast with little "permanent offset" but the "overshoot" can be substantial.

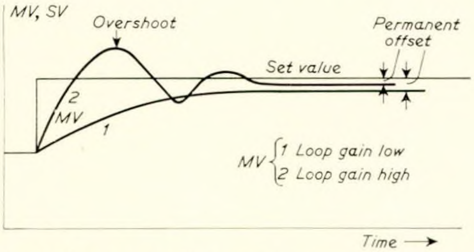


FIG. 14—*Influence of loop gain on system response—Proportional control*

Proportional Plus Integral Control

By including integral action in the controller the permanent offset can be reduced to zero; when MV differs from SV the controller continues to change P_o until the deviation is, in fact, zero (see also reference 10). The result is a consistent control.

Loop Linearity

As already mentioned the loop gain influences the stability of the control loop. It is therefore of practical importance that the value of the loop gain is constant over the entire SV range. In that case only, one and the same set of controller adjustments is suitable for the whole range. If the process characteristic is non-linear (e.g. cube law between steam flow and rev/min) this has to be compensated for by complimentary non-linearity somewhere else in the control loop. This is usually done by choosing an appropriate characteristic for the "control valve" (correcting element).

Discussion

MR. J. McNAUGHT (Member) was impressed with Mr. Bretveld's statistics, especially the growth rate of instrumentation in the last eight years; there was no doubt, on Lloyd's and other figures, that this increase was very large. It remained to be seen how far the increase would go, but this paper would give it a further impetus.

Much had been written about Diesel ships and bridge control, but few turbine ships had been talked about or technical articles presented in journals, and this was the first paper he had studied where the subject had been dealt with so thoroughly. The approach was good, systematic and analytical—which was the only approach which mattered.

All those people trying to make progress today would appreciate the difficulties which the authors and their colleagues must have experienced in collecting the material, information, data, opinions and so on, in order to build a successful model. He had a sneaking feeling that if something was not right in a model, one simply changed something to right it, but perhaps he was wrong.

Although the paper dealt with bridge control, it appeared to him that it was just the natural outcome of the study of how the turbine plant could be designed, controlled, and made automatic as far as possible, in order to avoid human intervention, which was sometimes but not always good. It would tend to take the art out of operating a steamship, and involve more science.

When on board a large steamship he had thought of the blasting technique (which would normally be called warming through), as he saw the engineer officer opening and shutting the four valves for the two engines. If nothing else was achieved, there would be less exercise in that area.

He had also had the problem of how to reduce the number of component manufacturers for instrumentation, and sometimes it had been very difficult to find a middle-of-the-road solution. One found a certain component made by the chosen manufacturer was not quite as good as that of another. He took the view that he would put in what would do the job, and then let the maintenance find its own place. Perhaps in due course all the manufacturers would make the right components.

Mr. McNaught asked, in the build-up of information, what was the record of stops on tankers at sea due to navigational hazards, such as a fishing boat getting in the way or a sudden fog? This was important in view of the remark: "It is essential that the bridge officer should have control of the main engine when operating with the main machinery spaces unattended."

He would like to have seen more mention of the effect on the control system of emergency conditions. On the ship he was on last week, if there was a black-out or if something happened to the governor and the bulkhead stop valve shut, there was an astern steam connexion which by-passed that, so that the engineer on watch, on instructions, immediately put the astern steam on, and braked the turbine to avoid any damage to the bearings. In the earlier ships this was not possible, and the engineer on watch had very quickly to gag open the bulkhead stop valve, and then apply braking astern steam. With the control system described in the paper, what happened in such a condition? Did it put on the braking steam, if so, how? Was there a risk of turbine bearings being damaged?

The other part of the study which appeared to be more arduous from the point of view of boiler control was to have a stop after a reasonably long full astern. In "old-fashioned" boilers—say seven or eight years old—the brickwork held a certain amount of heat, and this could cause a certain amount of confusion in the boiler room if a heavy steam demand suddenly ceased. He believed that the practice was for the engineer on watch to open ahead and astern steam as quickly as he could, and so reduce the confusion in the boiler room.

How was this dealt with on the control system described?

The full ahead to full astern application, where the machinery was slowed down to 30 rev/min, seemed rather like Diesel practice, but he thought the considerations were a little different. Fig. 15 referred to *Pendennis Castle* and, although it was made ten years ago, it still applied. This showed very different characteristics from those shown on the *Myrina*.

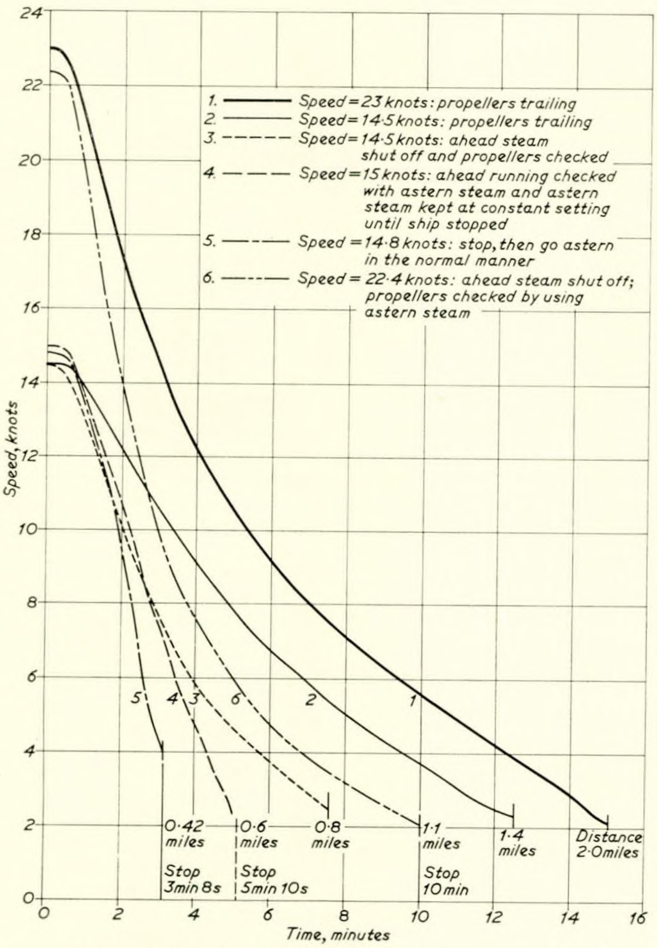


FIG. 15—*Pendennis Castle, 28 582 tons gross—Stopping trials, 30th October 1958*

It was not clear in the paper what over-rides there were on the bridge for machinery control, if any.

Mr. McNaught hoped that in due course there would be a follow-up paper with service experience, and more details of the systems within the ship, which were just as important as the bridge control.

MR. J. FORREST, B.Sc., thanked the authors for providing operational data as verification of the mathematical models used to evaluate and simulate the plant and control scheme. He believed the paper should be re-titled: "Bridge control of marine steam turbine plant—the only practicable approach".

The first point of major importance which the paper made was that there was a gulf existing between those who manufactured and sold control equipment and those who had plant to be controlled. The bridging or otherwise of that gulf determined largely the success or failure of a control scheme. That point could not be over-stressed.

In the *Operating Requirements* section, there was one point which had been under-stressed. The guidance of the plant designer as much as the plant operator must be sought in defining the plant operation. In the design of control systems it was also vital to obtain a clear definition of the plant operating parameters. The overall system designers must force manufacturers to do some homework and, for example, provide data on propeller cavitation and the effect of astern steam application on turbine rotor distortion.

Mr. Forrest would have liked to see a more forceful statement on this issue, and he was slightly disappointed to note that the authors had tended to steer clear of a problem which might have been easily tackled—or which might not even exist. He referred to the first and second points made in the section on *Operating Requirements*. Machinery must not only be designed to last the lifetime of a ship for normal operation, but also must be made to take heavy overload when required in an emergency. If that was not the case, why were design margins so high, and why should they not be pared down? Indeed, the restraint imposed in the second point applied the worst possible transient load to the boiler plant, as the boiler dynamics particularly of the feed system were not happy at a run down to minimum steam outputs.

It was evident that the marine industry was still smarting from some of the early excursions into the use of electronics for control, judging by the comments regarding basic design considerations. These fears could now be put aside, provided one was diligent in the choice of equipment manufacturer and in the preparation of the equipment specification. There were in particular two companies in the United Kingdom which produced equipment of this type which was ideal for schemes such as bridge control, which required logic and modulating signal handling with only one signal-mechanical interface. Pneumatics were far from ideal for such applications. Furthermore, modular construction and easy repair by replacement of printed circuit cards made electronics attractive in such contexts. It had to be realized that few engineers wanted to understand in detail what went on inside any moderately complex control scheme, and the "fear barrier" introduced by the adoption of a new medium became insignificant.

With regard to the section headed *Control Loop*, paragraph b, this statement applied to both open and closed control; even more so to closed loop. An open loop programmed control was dead beat, and only operator action created over-shoot.

Also the referred valve position/shaft speed relationship was actually less than cubic because of the non-constant efficiency of the propulsion plant and the ship power speed law. Indeed, it was almost a square law relationship at the lower power region. Many pneumatic speed detectors had square law output, and this might be used to advantage to provide a constant loop gain with a linear control valve characteristic. The linear characteristic was more acceptable under manual operation.

In the section *Delay and Derivative Units*, there was a statement which should be treated with caution, and that was the use of derivative action for stabilization. Mr. Forrest asked if the authors had any results from the system for motion in a seaway where fluctuating propeller torque was proportional to speed derivative. Past experience indicated that the boiler plant was lively under such conditions. This also emphasized the importance of the limiter circuits.

The use of time delays was described. Mr. Forrest was not happy with the use of time sequencing in such applications because there were too many undefined "ifs" and "buts" in the overall plant response. A constant application rate for desired speed could provide a satisfactory solution; this could be easily engineered from standard pneumatic units. He would like to have seen the system operating more vigorously in respect of astern steam application.

It would be interesting to see the ship speed traces superimposed on Fig. 10.

The analogue computer could be used for system check out and would provide a much more representative evalution of the system than the use of physically scaled down units. Had this approach been tried?

One of the highlights of the paper was the statement: "However, when cavitation occurs, having a very large ship quivering like a wet dog for no purpose is hardly considered good practice." Mr. Forrest hoped he did not detect a note of complacency concerning crash manoeuvring. His own paper [4] still gave the quickest way of stoping a ship, using the propulsion plant. This had been confirmed by both trials results, and by the use of optimum control techniques, including the use of Pontriagin's maximum principle of optimization. If it was required to crash stop a ship from high speed, perhaps not only should the ship quiver but the control system should be designed to find the person responsible, pick him up by the scruff of the neck and shake him like a wet dog; such were the consequences of failing to stop a large oil carrier.

As regards the design philosophy, was the equipment designed to fail set or to shut down? Had a turbine overspeed trip been considered in this application?

Finally, was there any consideration for the avoidance of critical speed ranges?

COMMANDER A. J. H. GOODWIN, O.B.E., R.N. (Member), in a written contribution read by Mr. Forrest, commented that it was natural for those who were responsible for the well being of the engines to wish to nurse them, and from this point of view the authors' comment that the "fastest with the mostest" was not the optimum way of stopping a ship was understandable. In his early days at sea in the Royal Navy, the engineers had established a position where ships' captains accepted that it took a full hour to work up to full power and at least half an hour to work down to cruising speed again. This attitude was supported by the Admiralty who, even up to the beginning of the 1939–1945 war, laid it down that increases or decreases of sprayers in use must not be carried out at a greater speed than one sprayer per two minutes. Naturally this procedure did not last long during the war when rapid manoeuvring mattered, but there still lingered the feeling that it would be nice if it could be shown that violent use of the engines did not reduce the stopping time and distance. One series of trials in H.M.S. *Savage* after the war was therefore devoted to testing this theory, and these trials and others carried out in H.M. ships had shown that for naval vessels at least, the quicker and greater the quantity of astern steam, the more quickly the ship stopped. It was difficult to see why this should not apply also to large tankers despite their great mass and relatively small power. If cavitation played a significant part during the stopping of the ship, one would expect it to show up either in trial records on board or on model propeller tests as a distinct falling off in thrust, with or without a corresponding falling off in torque. If there was such a falling off in torque it would be accompanied by racing of the engine, which did not seem to have been recorded, except verbally perhaps.

In a trial recently carried out in a fast cargo liner fitted with medium speed geared Diesel engines and c.p. propeller, the guarantee engineer for the latter recommended that better stopping times would be obtained by dwelling a pause in the zero pitch condition before applying the full astern pitch. Two stopping trials were carried out, one with the delay and the other with no delay in applying astern pitch. The stopping times and distances measured were:

With no delay: 3 minutes 15 seconds; distance 0·45 n miles.

With 20 seconds delay: 3 minutes 45 seconds; distance 0·50 n miles.

Cdr. Goodwin sympathized with the authors in recommending procedures and fitting control systems which limited the stresses imposed on the engines. If the relatively small sacrifice in stopping distance could be accepted, it would open the way to increasing the permitted design stresses in turbines, gearing, shafting and propellers and hence to appreciable savings in capital cost. The use of an automated control system with torque limitation built in as recommended by the authors would permit this to be done safely.

MR. D. GRAY, B.Sc. (Member) gained the impression that the shipowner's staff had not only carried out the design philosophy, but had also carried out the system engineering for the control scheme; for example there were references to the selection of both instruments and components.

If this was so, he congratulated the authors for carrying out such work. However, few shipowners had sufficient technical staff to carry out both design philosophy and system engineering.

The paper did emphasize the necessity for one design team to assume responsibility for system engineering. For the shipowners who could not carry out complete system engineering, the general pattern in the marine industry was that there were two specifications—one written by the shipowner, and the other by the builder. In such cases it was better if the owner's specification stated the broad operational requirements and omitted technical detail. The specification by the builder could then be the technical specification, and it would represent the shipbuilder's method of solving the problems posed by the owner. It would be the specification on which enquiries for control equipment would be based.

Unfortunately, this arrangement was not always realized. Many owner's specifications were a bit of a mixture of a performance specification with odd technical details often inserted. Such random technical insertions frequently made system design difficult.

The approach towards specification writing suggested by Mr. Gray did not place the owner completely at the mercy of the builder, as might be thought. If the owner wished to employ a particular control medium, e.g. for reasons of standardization within a fleet, this would be a perfectly valid operational requirement and would be completely appropriate to the owner's specification. If the owner had had previous unsatisfactory experience with a particular detail item of equipment and did not wish to use it in a future ship, this objection could be made known when the technical specification, prepared by the builder, was being vetted by the owner.

The authors had given very sound and valid reasons for the choice of pneumatic equipment, and they had referred to relative costs. Nevertheless, it was a curious fact that in the field of the large slow-running Diesel, where standard engines had had standard control schemes designed for them, electrical and/or electronic control was the most favoured.

This feature of the Diesel field possessed all the advantages which the authors were claiming for their scheme, such as familiarization by ships' personnel of equipment within a fleet, and an easier spares problem. But the greatest and most important advantage was that the machinery and the controls had been developed together, and had been tested and proved on the test bed and not in the ship.

The same standardization did not appear to exist in the steam turbine field. With the possible exception of one North American sub-contractor who offered a package deal of steam machinery plus controls, all other control systems for steam plants appeared to be one-off jobs. One of the main reasons for this was probably that there were not enough steam turbine ships being built to enable a sub-contractor to set up a production line and so effect a measure of standardization. Consequently control systems for steam plant could rarely obtain the benefits available in the Diesel field.

Mr. Gray was interested that they had thought it necessary to carry out an analogue exercise before the system was designed. This approach was being used more and more in the modern steam plants of electricity generating stations ashore. And the authors agree that such an exercise was almost essential for the large steam plants being proposed for ships on order or in prospect. He believed that more than one of the large steam turbine ships completed recently would have been improved, from the point of view of control, if an analogue simulation had been carried out before the control system was engineered.

He agreed with the use of rev/min feedback as an aid to precise control. However, it was not as novel as the authors had suggested; it had been included in steam plant control for several years by some designers.

Like the two previous speakers, Mr. Gray would like to hear some details of the safety features for the plant. In his view these were an essential part of the system engineering.

MR. H. O. WALKER agreed with the author's proposals with one or two exceptions. He suggested that an alternative method should be employed to convert the command into an acceptable "DV" or "Desired Value" which might be directly applied to the controller as shown in Fig. 5.

From Fig. 16 it could be seen that the command and desired value were compared by a precision relay (9). Should the command exceed the desired value (i.e. a rising command) then the relay would trigger and by means of the slave relay (10) two pilot valves (1 and 8) would be operated. Pilot valve (8) isolated the command from the desired value for the time being. The second pilot valve (1) applied a 20 lb/in^2 signal to the time delay unit consisting of two needle valves (2 and 3) and a capacity chamber (6). If the 20 lb/in^2 supply was applied to the needle valve-capacity chamber unit then the pressure in the chamber would increase exponentially. To prevent this occurring a flow control unit (5) was arranged to measure the pressure drop across the needle valves (2 and 3) and maintain it constant. The rate at which the pressure would increase is thus linear. The rate would depend on the valve setting. Two valves were normally employed so that if both valves were in circuit together, a fast rate of increase might be obtained, whilst if valve (3) was isolated a slower rate was available.

Normally a signal was taken from the measured value or speed signal and used to trip a relay which in turn operated the pilot valve (4).

Ultimately the desired value equalled the command and the relay (9) would trigger once more. The two pilot valves (1 and 4) were operated once more and the 20 lb/in^2 system was isolated from the system. The command and desired value were now connected through the flow restrictor (7). A ramp function generator was now included in the command circuit and the "bang-bang" command could be converted to an acceptable signal which might be used for control purposes. (For both run-up and run-down.) Widely different rates of increase were possible, an example being:

Manoeuvring: 0–80 rev/min in 30 seconds
Cruising: 80–120 rev/min in 30 minutes.

He noted that the authors used only one controller for both ahead and astern turbines. Did they consider this satisfactory or did they feel that individual controllers having different proportional band and integral times would be more satisfactory?

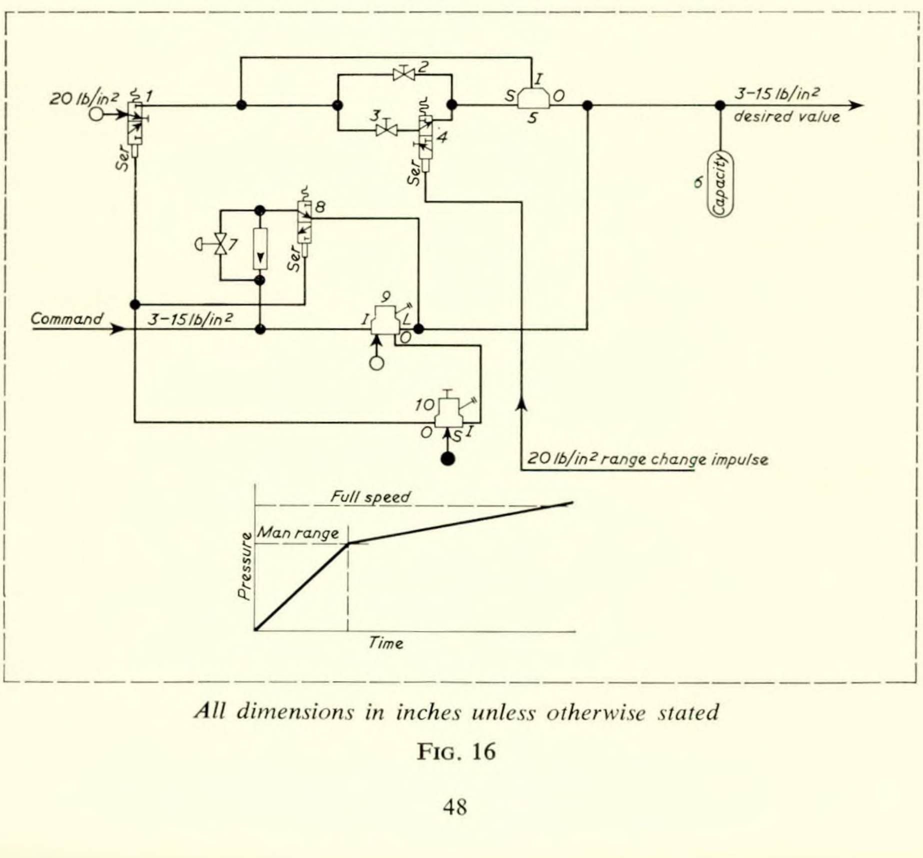
MR. C. CANE said that the paper related a theoretical analysis of the problem of bridge control to actual practical operating experience, together with a statement of the results achieved.

The application of industrial process instrumentation and control to the specialized needs of the marine industry had taken a distinct step forward with the achievement of the results noted in the paper.

The use of industrial process instrumentation for the marine industry was not new. His company had had experience of process instrumentation for marine use as far back as the Second World War, when they had to meet the arduous "g" operating requirements of the Admiralty. After the war, temperature and humidity measurement and control instrumentation were widely applied in cargo vessels.

The real stimulus, however, for the use of process type automatic control on board ship seemed to have come rather later as a result of worldwide political moves and the ever increasing difficulty of obtaining satisfactory manpower at the right economic rates of pay.

Mr. Cane concurred with the point made by the authors concerning the importance of the study of the control system as a whole. The design of the control system must proceed concurrently with the design of the plant if a properly integrated whole was to be achieved. Would the authors comment on the boiler control, since this must form an important part of the overall control problem, and also on the problem of



All dimensions in inches unless otherwise stated

FIG. 16

the responsibility for the overall control design? He believed that this could only be satisfactorily resolved by discussion and agreement between the shipbuilders, the owners and the instrumentation control manufacturers.

The largest constraint on the system was likely to be the boiler level control. Control systems had been developed using programmed level control. Did the authors have any experience in this area?

Most ships' turbines employed sequential nozzle control for reasons of efficiency. As this point was not mentioned, could it be assumed that sequential nozzle control was not satisfactory for reasons of characterization and the need for constant loop gain?

The characteristics of the ahead and astern turbines were normally different. Would the authors comment on the use of a single controller to achieve both functions?

The ahead/astern safety system was designed in such a way that astern steam was not admitted until the controlling rev/min of the propeller dropped to a predetermined level. Would the authors comment on the reaction of the operators to not applying astern steam in an emergency situation? Although it was undesirable in practice to have too large a difference between ship's speed and propeller's speed (otherwise cavitation might occur), surely it was a little extreme not to apply any astern steam until the propeller speed had dropped to 30 rev/min. Was it not possible to design a system which compared ship's speed with propeller speed and admitted steam to the astern turbine to an amount just short of cavitation?

Mr. Cane finally commented on Mr. Bretveld's philosophical criticisms of the instrument industry. He could only reply on behalf of his company. Business marriages were only consummated if both partners were willing. This meant that the marine industry must be willing to use automatic control instrumentation, as well as the instrument manufacturers feeling confident that they had a contribution to make. As far as his company was concerned, he was certain that they had reached that state of affairs. They had the instrumentation and could provide the necessary installation service; they would commission at sea and could provide a worldwide service at very short notice for spare parts.

MR. B. WICKSTROM said that the authors had made a realistic approach to the bridge control concept. They had taken the mystery out of it, and reduced the studied part of it to a simple speed control loop with remote speed setting as used on all turbines for electric power generation.

He agreed that the correct selection of valves and valve positioners was essential for performance and price. Different solutions were possible, and consideration must be given to valve characteristics as well as to valve and controller prices, and also to the turbine manufacturer responsibility.

In regard to valve characteristics, the authors had proposed a non-linear valve which was certainly correct when the servomotor time constant was in the order of seconds. Fig. 17 showed two closed loops, including one necessarily non-linear link, the torque feedback. A non-linear valve could compensate for the non-linear steady state gain of the lower loop. The dynamic gain, however, would then be non-linear, giving varying stability criteria if the upper loop had a more rapid response than the lower. In that case the stability was obtained by the upper loop alone, without the aid of the torque feedback, and thus it was not influenced by the torque non-linearity. A rapid linear valve could therefore be a good alternative to a slow non-linear valve, and give better stability at low speeds.

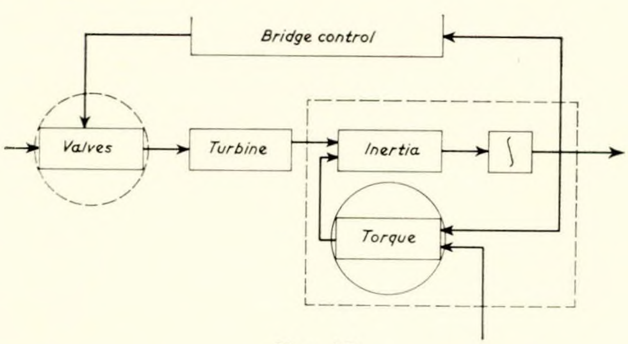


FIG. 17

Mr. Wickstrom agreed that today pneumatic controllers were generally cheaper than electronic ones, although some of the functions of the described system, such as limiting and derivative action, were much easier to arrange electronically. However, the main cost lay in the valve with servomotors and positioners. The least expensive valve with good performance would be the best choice. His company was developing electrohydraulic valves with a time constant of 0·2 seconds, intended to be essentially linear; they would be considerably lower priced than the non-linear pneumatic operated valves available.

Concerning the turbine manufacturer's responsibility, Mr. Wickstrom believed that he must be responsible for the manoeuvring valves, including the turbine safety system. The valves should be designed to respond correctly and rapidly to any external control signal as far as the turbine safety was fulfilled. His company's electrohydraulic system incorporated the principle shown in Fig. 18, including a top valve for basic manoeuvring and starboard and port valves which could be blocked, making it possible for the most economic operation to be obtained at any speed. Any commercially available electronic controller could be used, but the electronic input could also easily be derived from a pneumatic controller by means of a normal pressure transducer.

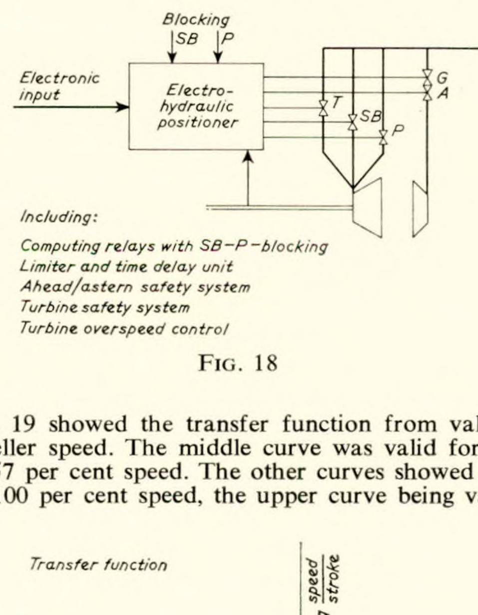


FIG. 18

Fig. 19 showed the transfer function from valve stroke to propeller speed. The middle curve was valid for approximately 57 per cent speed. The other curves showed the function at 100 per cent speed, the upper curve being valid for a

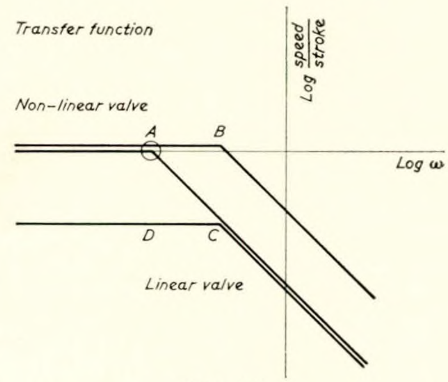


FIG. 19

non-linear valve and the lower curve for a linear valve. It was clear that at low frequencies, as at point A, the gain was constant when using non-linear valves, but varied to D with linear valves. At higher frequencies, as at C, the gain was constant with linear valves but varied to B with non-linear valves. When one used a rapid servo, cross-over frequencies higher than C could easily be obtained, giving constant loop gain at all speeds with a linear valve. Controller settings giving proportional band 10 per cent and integral action five seconds should be possible, eliminating the need for derivative action.

Mr. Wickstrom hoped to be able to follow up and take part in the future development in this field.

Mr. A. R. Hinson (Associate Member) first discussed the standard symbols which the authors used in the paper. Bridge controls and other automatic equipment were here to stay, and there would be many papers on this subject presented in the years to come. There had been several in the last three years, but this was the first which had stated: "Standard symbols and identifications were used in diagrams to avoid ambiguity when ordering or discussing equipment."

Mr. Hinson believed that it would have added to the value of the paper if the control engineering terms used had conformed with British Standard 1523, Part I, 1967. Further, if all papers on control engineering conformed to that standard it would make them easier to refer to in the future. The subject was sufficiently complicated without having to learn a fresh set of symbols for every paper. What were the authors' views?

It was particularly interesting to see that the authors treated the propulsion plant and auxiliaries as a complete process, which they then broke down into various sub-processes. This was a logical approach which had been used by designers of steam turbine installations when preparing heat balances. It enabled the plant to be assessed as a single unit, as it should be, while still making it possible to check each sub-process individually. However, it was not always possible to treat the sub-processes separately; they interacted, as in the case of the turbine and boiler when manoeuvring.

The authors had taken care to include a limiter and an ahead-astern safety system to limit steam pressure and water level swings in the boiler. These devices also ensured that the astern turbine did not overheat and that the propeller speed did not vary so much that the propeller cavitated.

Concerning the swings on the boiler, in at least one other bridge control system the astern valve began to open before the ahead valve was closed. This was often done when turbines were controlled manually. It appeared from the description of the ahead-astern safety system that this could not happen, but would the authors confirm that that was the case.

With regard to overheating of the astern turbine and propeller cavitation when trying to stop the ship in the shortest time, it was better to dissipate the kinetic energy of the moving ship at the propeller or in ship resistance than at the astern turbine. If, when stopping quickly, the propeller speed was so regulated that cavitation did not take place, the ship would be driving the propeller. The astern turbine might overheat since the energy which was being fed up the propeller shaft must be dissipated in the astern turbine. It was evident that while the propeller might be used as a water brake, the astern turbine should not be used as a steam brake if this caused overheating. From this point of view, cavitation could be a blessing in disguise, and the conclusions reached by Messrs. Goodwin, Irvine and Forrest could hold good.

As the authors had said, information on astern turbine overheating and propeller cavitation was scanty. A great deal could be learned by simultaneously recording the astern turbine temperature and propeller shaft torque.

Incidentally, it was partly as a safeguard during manoeuvring that Lloyds Register recommended a high temperature alarm in the astern belt when a vessel was to be operated with an unattended machinery space.

Mr. F. H. Proctor commented on the claims put forward for pneumatic bridge control systems. One of the authors had made several visits to Japan in connexion with the new building of Shell, and he probably had discussions with Japanese shipbuilders on their electrohydraulic bridge control systems. What were the authors' views on those systems compared with the pneumatics which the paper approved of one hundred per cent. Did it mean that Shell would standardize only on pneumatics, or would the electro-hydraulic systems be considered?

Mr. Proctor's second point was about the primary elements, because any control system was dependent upon the accurate measurements that had to be signalled into the controller. There was shaft rev/min and shaft torque that must be measured accurately, and the repeatability must also be good. Had the authors found the primary elements accurate and repeatable in their bridge control systems?

There was one sentence which needed further explanation: "The control system must therefore be such that bridge orders are issued with one action only, and recorded automatically". He did not understand the use of the word "recorded". Was this recorder the ordinary telegraph recorder, or was it part of the operation of the control system? The two functions seemed completely different.

Mr. P. R. Owen (Associate Member) quoted the authors' remark about the approach which "should consider a main steam turbine plant as an integrated unit." His interpretation was that the boiler/turbine plant should be considered as one, with bridge control operating via a boiler/turbine/shaft loop. This type of control gave "a turbine following system", in contrast with that described which was known as a "boiler following system". With a "turbine following system", there was no need to build constraints into the control circuits to take account of boiler dynamics, because an increase of demand for higher rev/min acted upon the energy input, that is, by fuel and airflow control, instead of acting upon boiler output, that is, steamflow. An increase in energy input would tend, after the thermal and mechanical lags of the boiler had been overcome, to increase the boiler steam pressure. This tendency would be sensed by a proportional plus integral controller which would open the manoeuvring valves, and rev/min feedback acting upon the energy input would control the energy so that the desired rev/min was obtained. Had the authors considered such a system?

Mr. Owen's second point concerned the pneumatic transmitters connected to the bridge control lever. These transmitters were arranged so that a signal of 3 lb/in^2 corresponded to a demand for full astern. With this system, an air failure in the control circuit would result in the machinery running full astern. Would the authors comment on their choice in relation to the alternative of having individual ahead and astern pneumatic transmitters, where a 3 lb/in^2 signal corresponded to stop?

The Authors' Reply will be published in a later issue of Transactions.