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Naval Experience in the Design and Operation of Machinery Control Systems*

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THE BACKGROUND

Philosophy

Control in the generalized sense is a function of management and has, therefore, to be regarded in hierarchical terms as comprehending several levels of intellectual ability and of responsibility. For present purposes, these may be labelled—operation, supervision and direction (see Fig. 1). Operation

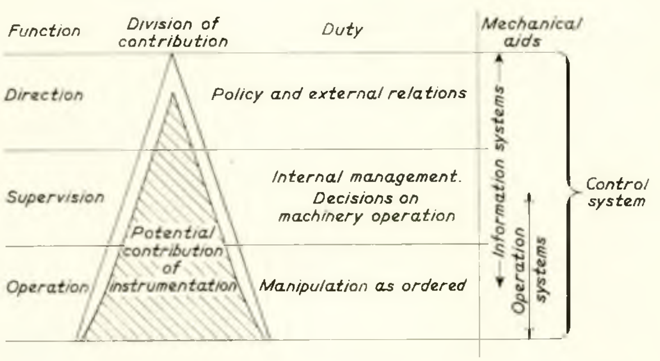

FIG. 1—*Levels of decision and control*

is held to deal with the variation of plant output to meet demand and the adjustment of running conditions to meet specified operating conditions. Supervision and direction are middle and higher level functions concerned principally with maintenance and operating economy. At all levels the fundamental element is the measurement of quantities, on the basis of which plant conditions are interpreted and corrective action is decided and executed. It is the level at which decisions are made which is the key to the consideration of automation.

In the present context it is assumed that measurement devices exist and the authors' interest is directed to decision making and consequent action. Here we are faced with the choice between men and mechanisms to undertake the functions of interpretation and action. This will depend on the requirements to be met for performance (in the widest sense) and on the relative capabilities of up-to-date men and mechanisms. The choice made determines the level of automation (a word which, unqualified, is meaningless).

While general consideration of this aspect of plant management demands that all levels of decision shall be considered together, it is convenient, for discussion and for design, to apply labels as follows:

a) where the decision is taken and applied directly to the operation of the plant—this is an "operating system";
b) where the decision is necessarily made by a man and may be applied at any level of management—this is an "information system".

Speaking pedantically, a control system comprises both. Under current convention however, the "operating system" is of course called a control system and this is the term the authors will use. The use of systems explicitly designed as integrated information systems is excluded from this paper.

The Field of Discussion

It is necessary next to define the particular field of control which will be discussed. Remote and automatic controls in various forms have been used in marine engineering for very many years and it is necessary to be explicit about the field of the present discussion. Firstly, the machinery concerned is steam and gas turbine propulsion plant, including the auxiliaries directly connected with them. Many automatic controls are fitted to non-propulsion auxiliaries as well, but are excluded. Secondly, the control arrangements, which will be discussed refer, for remote controls, to power-assisted manual controls; for automatics, to closed loop analogue servo-controls: both powered from sources external to the plant proper. Thus, consideration of spring-loaded relief and reducing valves, thermostats and similar self-contained types of control are not specifically covered.

Naval Requirements—Operational

The demand which originally gave rise to the use of comprehensive remote controls was a military requirement for the control of main propulsion plant from a central position remote from the machinery compartments. This was subsequently extended to include manœuvre over the whole power range, which brought with it the need for automatic controls in addition. This raised remote controls to the level of executing load changes (and the operation of certain standby auxiliaries), with all the lower grade control performed automatically.

Experience, since then, in about twenty ships fitted with full remote control, has confirmed the fact of other important advantages, associated more with the automatic controls rather than just the remote controls. These include high accuracy of control, fast response to changes in demand. There is also a clear indication of reduction in deposit rates in some heat exchangers, and an obvious, but not yet measured, saving in wear and tear of pumps. The advantages for personnel include reduction in fatigue of those watchkeepers who do not need to be in the machinery compartments, centralized presentation of information and grouping of remote

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controls which, in certain cases, have been found to offer a higher accuracy of control than can be obtained locally. Remote boiler feeding may be quoted as a surprising example of better quality of control, though this has only been required, so far, for training operators.

Naval Requirements—Technical

The technical requirements to be met may be defined in terms of space limits and manœuvre requirements, with fuel economy and refinement of performance necessarily subsidiary targets. The impact of strictly limited deck height, constant demand for compactness of the installation and the necessary water-tight sub-division, is to impose a division of the plant into compartments like an egg box. The egg box layout enforces a degree of physical separation of the operators from some, if not all, of the plant, which has to be compensated for in the mode of keeping watch and control over the plant.

Manœuvre requirements are severe and not in the same order as those for other steam power plant ashore or afloat. Power swings of 70 per cent and more are required to take place in seconds rather than fractions of a minute, with plant delivering upwards of 20,000 s.h.p. Such swings, it should be emphasized, are routine in the normal duty of a warship, not just occasional.

Naval Requirements—Personnel

Circumstances in the field of personnel and administration have, also, a bearing on the design of naval control systems. The matter of reduction of operating personnel is always a target, but must clearly be considered in terms also of maintenance requirements, other duties required of engine room staff, notably in battle, and the extent to which better management of the machinery may offset the retention of staff. The authors feel too, that to approach this subject principally from this angle is illogical, and will therefore produce inelegant designs which are inevitably accompanied by disadvantageous side effects. Subject to meeting the operational requirements, it is preferred, perhaps puristically, to approach with the intention of achieving the optimum division of labour in the hope that this will do the job much better and save long term costs. On this basis, the saving of manpower can only be a welcome by-product.

Discontinuity of operating personnel afloat is a problem which bears on the design of control systems. Requirements for training and promotion and the impact of conditions of service limit the time, for which a man will serve in one ship, to a normal maximum of two years. It is also implicit in the nature of the Navy's function that ships' personnel shall be self-reliant in maintenance and repair to a considerable degree.

Organization for Research, Development and Design

The organization for these functions has been built up gradually, since 1958, around one marine engineer, with some knowledge of control engineering, and one control engineer. The total numbers involved are about seven engineers and scientists (four of whom are part-time only) working at the Admiralty, the Admiralty Engineering Laboratory, the Admiralty Fuel Experimental Station and the Yarrow-Admiralty Research Department.

So far as facilities are concerned the organization has ready access to an instrument laboratory at the A.E.L. and to a comprehensively-instrumented full scale test vehicle at the A.F.E.S. The latter is a naval boiler of current design with a full outfit of auxiliaries. In addition it has the use of several small analogue computers and two digital computers.

This, then, is the background against which the authors' experience in design and operation is set.

DEVELOPMENT OF DESIGN DOCTRINE

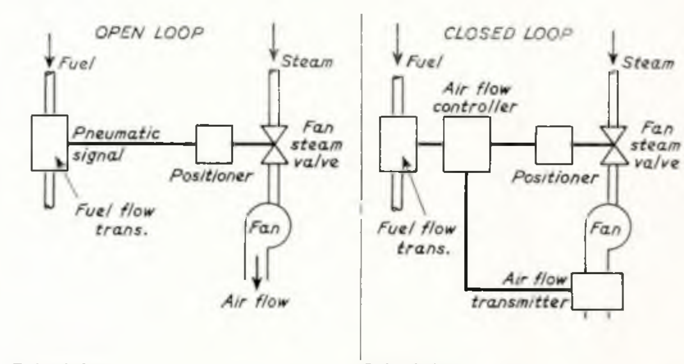
Nature of the Control Process

Before embarking on the design of control systems it was necessary to gain a clear understanding of the nature of the processes of executing control over anything. This would apply whether the control was exercised by instruments or by men.

The fundamental element is, of course, measurement of the controlled quantity and this is followed by the interpretation of the significance of a departure or "error" in the measured value from a memorized operating instruction, e.g. specified boiler drum pressure. This leads to the decision whether a correction should be applied or not. If the decision is to make a correction then a computation follows to determine sense and degree of correction, which is then applied, without further ado, to alter the controlled quantity.

If accurate control is required then the perceptive senses must be in almost continuous use. In the process of restoring a specified operating condition, the whole sequence is constantly repeated until restoration is complete, that is the applied correction will be varied to ensure precise alignment of actual and desired conditions without overshoot, or undershoot.

Where a man is in control, he can exercise all these functions except one by the use of sensory perception, memory and low level logic. To this extent he can take over a strange job quickly. The one function which he cannot exercise accurately, at first, is the computation of the correction to be applied. This is a problem of learning by trial and error and requires him to exercise, in addition, the faculty of higher logical thought. In fact, the information which he has thus to gather is related entirely to the dynamic behaviour of the parts of the plant which he is trying to control and, especially, to the differences in dynamic behaviour of two or more associated units, e.g. fuel pump and forced draught blower.



Principle:
Fuel flow is measured and pneumatic signal positions fan steam valve.

Steady Condition
a) Fuel flow
b) Steam inlet pressure and temperature
c) Steam exhaust pressure and temperature
d) Fan friction
e) Air resistance of boiler and register.

Changing Conditions
If fuel flow steadily rises, air flow lags fuel flow by fan time constant, e.g. 20 sec.

Principle:
Fuel flow and air flow are measured. Controller adjusts fan steam valve until air flow signal is equivalent to fuel flowing.

Steady Condition
Air flow depends on:
Fuel flow only
Fuel flow = air flow
The only errors are instrument errors.

Changing Conditions
If fuel flow steadily rises, air flow lags fuel flow by less than fan time constant, e.g. 4 sec.

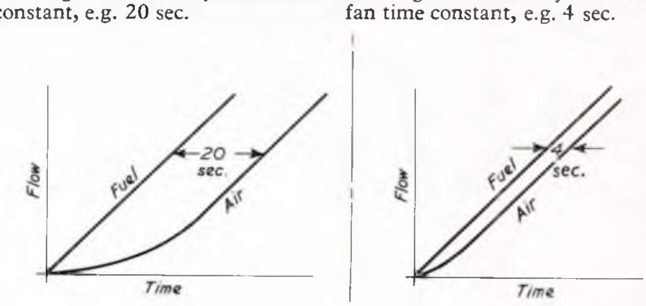


FIG. 2—*Air flow control—Open loop, closed loop*

In the case of control by instrument, the lower grade faculties can be provided by choice of measuring device and by an instrument setting. The further settings required to provide the correct computation are not easily established, however, since they will depend on measurement of the actual dynamic characteristics of the function being controlled. These must be predicted or established experimentally if an elegant and simple design is to be achieved.

This reveals a major point on design organization, namely, that erudition in control engineering theory is of no value unless it is accompanied by a sufficient understanding of the plant itself, as to both its dynamic characteristics and the methods which are to be used in operating it.

Organization of Control Systems

Two general techniques are available for executing a control function—open-loop and closed-loop control (see Fig. 2).

Open-loop constitutes the use of one parameter to control another which is closely, but not necessarily rigidly dependent upon the first. For effectiveness the complete relationship between the two must be predictable.

Closed-loop control requires the comparison between the desired and the current actual value of a parameter, the error between these being used to adjust the value of the same parameter. It is, therefore, a direct self-correcting arrangement which compensates for external variations such as may occur in e.g., steam and exhaust pressure and temperature, deterioration of machinery performance. Thus it has, inherently, a faster, more accurate response than open-loop and is used almost exclusively in automatic controls for naval machinery.

Control Action

In considering the degree of corrective action to be applied it is obvious that this should be regulated so as to be directly proportional to the instantaneous value of the error between desired and actual value of the controlled quantity. It must be emphasized that there is, here, the clear implication that the generation of a corrective action depends upon there being an error present. If the error is made small by proportional action alone, the machinery will be caused to oscillate about the desired condition. Alternatively, a residual error will remain. Some device is necessary to nullify this, which must also, obviously be related to the magnitude of the error. A convenient and natural way to do this is to generate a signal which is the time integral of the error and add this to the proportional control signal. This gives two-term (proportional plus integral) control action, which meets practically all requirements (see Fig. 3). In some cases, however, where it is necessary to accelerate the control action sharply an additional term is used generated by differentiating the error as it develops, i.e. derivative action.

Certain snags have been experienced in the application of integral action. For example, during a fast power swing the error may become large and the control signal generated to

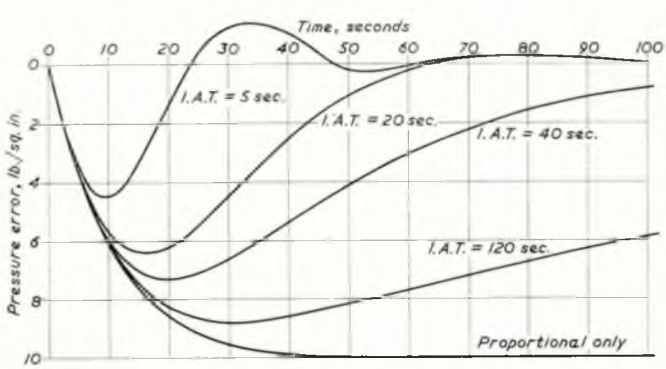


FIG. 3—*Effect of integral action time on steam pressure response*

deal with it may exceed the scope of corrective action which the plant can achieve, e.g. during a power increase the turbo-blower steam throttle may open wide and still not satisfy the control signal applied to it. The element providing the corrective action, the throttle valve, is then "saturated" and the control loop is out of effective control until the blower has accelerated and brought the error value down to within the scope of the throttle, which will then move away from full open and resume proper control. The saturation condition is one in which the control signal may continue to build up (due to integral action) while the correcting element can do no more to follow it. If, in this condition, a sudden reverse load change occurs, which is perfectly likely in fast manœuvring, the error value is reversed and proportional control action will start to develop. Before this becomes effective, however, it has to reach a level sufficient to overcome the integral term which has been built up, and for the moment the plant is out of control. To overcome this it has been necessary to introduce a simple signal limiter which inhibits the development of an excess integral term.

There is another point which deserves mention. There is a tendency to regard control instrumentation as comprehensively superhuman. Whilst, as a piece of engineering design, a control instrument may be very elegant, functionally it is sub-moronic. It can attend to only one physical quantity, and this very well, but it cannot make allowances for extraneous events or predict situations, cannot endow machinery with performance not originally designed into it, and is certainly unable to keep the machinery under complete supervision in addition to its intended job. This is completely obvious, but is often entirely overlooked.

Design of Control Loops

Up to the present date it has been necessary to accept settled designs of machinery and to devise the ways of controlling them. This has meant firstly, determining dynamic behaviour of machines from their static characteristics, as designed, then deriving mathematical models for each machine. The next step is to establish how the machine is going to be operated with reference, in particular, to the nature of its contribution to the generation of power and the extent of the acceleration of output which will be required of it. The next step is to decide what instruments and valves will be required to control it, and then to conduct a simulation of the behaviour of the machine, control valve and control instruments. If the machine is controllable then the loop so designed and modified will deal with it and, further, the simulation will give a good prediction of the instrument settings required.

The first experience in boiler control system design was in 1958 with H.M.S. *Anon* (a ship with machinery of pre-war design), which was based on power station practice (see Fig. 4). In this, the error in boiler drum pressure directly controlled the F.D. blower speed. The value of the consequent air flow was then used to control fuel flow, using constant pressure fuel pumps and lance type, wide range burners. This proved unworkable for high rates of manœuvre because of the large time lags, due not only to the boiler itself, but also to the high inertia of the blowers.

This system was re-designed by arranging drum pressure error to control fuel flow direct. The measurement of actual fuel flow being used to control the blowers, using a closed control loop around the blowers themselves. The result was reasonably satisfactory, drum pressure being held to within plus or minus 15lb./sq. in. during violent load changes.

Residual difficulties remained, however, notably:

a) the lance type burner with its drives was not the right configuration for the rapid load change requirements and in subsequent ships spill burners were used;
b) the accuracy of control of fuel pump pressure was inadequate: with the change to spill burners the automatic control of fuel supply and spill pressures became necessary anyway;
c) the magnitude of the time lags in the pneumatic

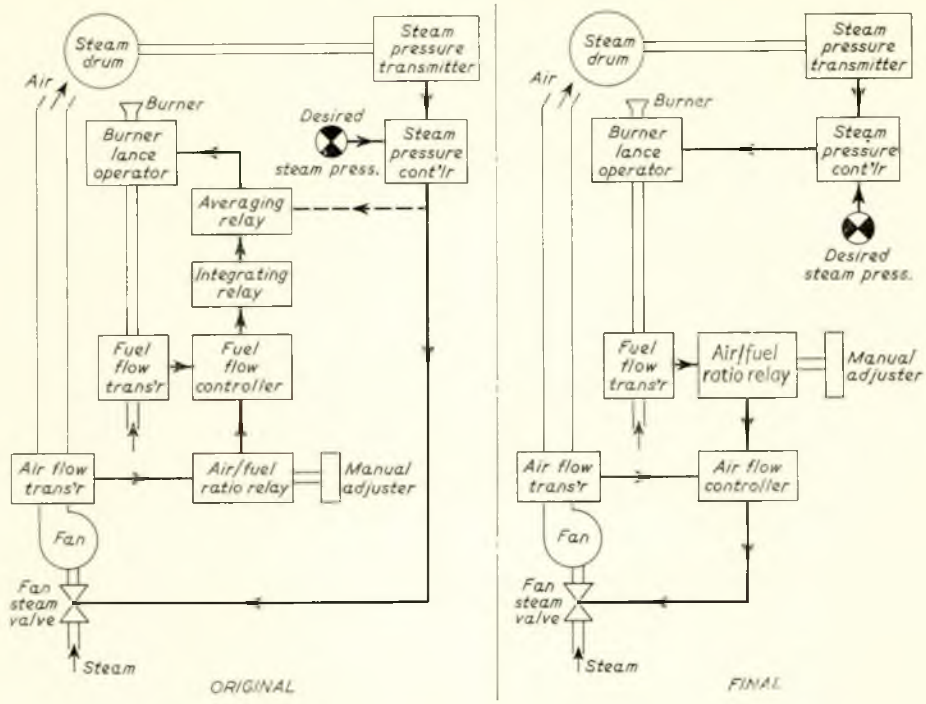


FIG. 4—*H.M.S.* Anon *boiler control system*

transmission lines was unacceptable; this was because these lines were made to run back and forth between the machinery and the control room because of an original misapprehension about the function of the control room and ignorance of the means available to simplify and much reduce the length of these circuits;

d) the somewhat crude location of sensing points, which admitted time lags within the control loops which were neither necessary nor helpful.

While there was time to remedy many deficiencies of these types in the next generation of ships before they completed, there were still primary lessons to be learned, notably in connexion with running F.D. blowers in parallel. In some ships, as originally designed, the throttles of both blowers were mechanically connected together and jointly driven by a control drive. This arrangement was unsatisfactory because of backlash in the drives and minor variations in throttle valve behaviour. The better method here was to use a single throttle controlled by a closed loop to feed both blowers.

With the experience gained in these earlier installations, some of these larger difficulties can be eliminated at the design stage.

A hard lesson has been that of the correct selection of control valves. Whilst the choice of lift/flow characteristic is more or less fixed by the characteristics of the plant, the sizing is dictated by the variation in load to be catered for. This approach is necessary if accuracy of control is required, since the whole travel of the valve needs to be exploited to permit precise positioning at any point in the power range. Solution of this problem demands first the recognition that a control valve is a variable orifice, hence serious attention is necessary to the fluid dynamic characteristics of individual designs of valve. Sizing to pipe size is hopeless for control valves, but fortunately the specialist valve makers provide all the data necessary to make the sizing process relatively simple. A startling example of over-sizing of control valves showed itself in one class of ship. Here a remote controlled boiler feed check valve had ten revolutions of travel from shut to wide open, and yet was found to pass almost 100 per cent of the feed pump output after only half a revolution of the handwheel, the remaining 9½ turns contributing practically no additional flow. Reasonable control was obviously impossible, but by modifying the valve so as to reduce the port area and to produce a chosen lift/flow characteristic, precise control became easy.

It has been the authors' general practice to go for simple and stable systems and this has been achieved very largely by applying the closed-loop concept to individual correcting elements. These have then been connected together in a cascade with remote setting of the ultimate parameter only, when in full automatic control. Thus with steam turbine plant, the only human operation used on the main power line is remote positioning of the turbine throttle. The load variation im-

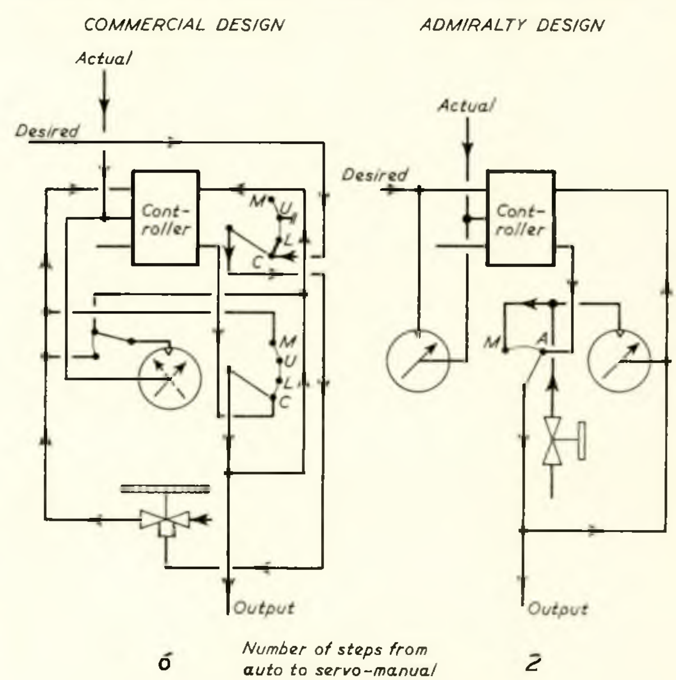


FIG. 5—*F.F.O. Pressure control station*

posed by throttle movement is automatically followed by all the steam generating plant under the control of subsidiary closed loops.

Design for Operating Simplicity

However comprehensive and automatic the system may be, the point of contact with the human operator is reached at the control panel. An important point to consider here is that the operator has been relieved of all the low level regulation work and is only controlling the higher level parameters. It is, therefore, demanded of him that he use his intellect to a greater extent than hitherto. Training and recruiting arrangements do not cope with overnight advances of this kind, on a broad front, and it must be accepted that a gap has to be bridged. In the particular context of rapid manœuvre and, especially, of fast development of new situations, it is clear that human manipulation of the control system has to be simple. This is a design function to which it has been necessary to pay much attention (see Fig. 5).

INSTRUMENTATION MEDIUM

The earliest ventures relied on either pneumatic or hydraulic instrumentation and took place prior to the development of reliable electronic process control instruments. Hydraulics, however, do not satisfy the particular requirements for lightness and for simplicity in signal processing. Electronics are not as yet well suited to the environment of steam machinery. Pneumatic instrumentation, on the other hand, meets all these requirements, provides all the performance needed and has the additional advantage of having been developed in industrial use for thirty years and more. The authors have, therefore, adopted pneumatic instrumentation for all major control systems. Although there are several excellent ranges of proprietary pneumatic instruments they have adhered substantially to two in order to simplify training and logistic problems.

EXPERIENCE OF SHIP INSTALLATIONS

Installation and Trials

The installation and trials phases of work have caused considerable problems. These have stemmed from insufficient appreciation of the nature and function of control systems and instruments and of the mechanical requirements which must be met in the course of installation. Particular problems in the trials phase have arisen because of the difficulty of precisely specifying performance.

As an example of installation problems, the location of sensing points may be cited. For good control, the correct location of sensing points is important and in some cases critical. Thus, to position a sensing point a few feet away from where it is required, for ease of manufacture, may destroy the possibility of achieving reasonable control. Since a sensing point almost always constitutes a penetration of a pressure vessel it will obviously be both costly and tedious to shift it to its correct position. There are so many such points concerning installation of control systems which have been learned the hard way, and so little published guidance available, that it has been necessary to produce a guide book on the subject.

With regard to bringing control systems into use it has been found necesary to establish a formal sequence of testing and adjustment, culminating during the sea trials period in an appraisal of performance. This sequence starts with static inspection and tests, goes on to dynamic tests and adjustments of individual loops and finally a series of overall system dynamic tests. The start of each phase in the testing sequence postulates successive degrees of completeness of the machinery installation and availability of the machinery and the necessary services and of load.

Performance specifications have not yet been obtained and the basis of acceptance has, therefore, to be subjective. This requires some care in administration and, in order to maintain a substantially consistent standard, it has been found necessary to repose the duty of performance assessment in a single, advisory body of people—a small trials team. Again there is no published information on this phase of the work and plans are in hand to fill this gap.

One of the key points which always come up in this stage is, again the moronic nature of instruments. These are quite unable to allow for even quite minor errors in machinery installation, or the unpredictable temporary influences of shipbuilding activity. There was a case in one ship of an F.D. blower which seemed unable to produce its proper output. The instrumentation came under a storm of vituperation before the real cause of the trouble was found. A welder working in the air intake space had needed a stage of planking which extended right across the intake duct and effectively obstructed air flow to the blower.

Service Experience

In the early installations, insufficient account was taken of the need for clean, dry and oil-free air, and serious trouble resulted with fouling of the small nozzles and restrictors built into pneumatic instruments. A film of oil on the walls of an orifice of bore 0·005in. obviously wrecks its performance. This lesson has been painfully and completely learned.

Apart from this, such difficulty as there has been has stemmed mainly from insufficiently detailed training of operating and maintenance personnel. This has manifested itself in the form of misguided alteration of instrument adjustments and settings, principally when the machinery operation was, itself, imperfect. The training required lies in two fields. First, on the construction, balancing and adjustment of the instruments themselves. Secondly, on the concept that machinery and the associated controls are not disparate entities, but rather that each is a part of the other and both are entirely interdependent. In this connexion, it also needs to be put across that unless the plant is capable of working in the crudest form of manual control, the instruments cannot by magic make it work.

Reports from sea have shown that the remote and automatic controls are popular and accurate, simplify efficient plant operation, and are definitely causing a reduction in the creation of deposits in heat exchangers, thus allowing the maintenance of better performance for longer periods and reducing maintenance liabilities. The improved working conditions for control room watchkeepers are also very much appreciated. Some watchkeepers find a little difficulty in adapting themselves to the remoteness from machinery noise and smells, but this is confined to the older man. The remoteness of the watchkeepers from machinery demands the placing of greater reliance upon the indicating instruments than ever before and this has called for a new emphasis to be placed on the maintenance of accuracy of such instruments. Equally, for accurate operation of machinery, the control instruments themselves must undergo periodical tests for accuracy and balance. It has, therefore, been found necessary to provide ships with a range of simple test equipment, of which the most important are those for checking the accuracy of pressure and temperature measuring instruments.

HISTORY OF RESEARCH

Initial trials on H.M.S. *Anon,* pointed the need for a quantitative knowledge of machinery characteristics, both steady state and dynamic. Without this knowledge it is impossible to make effective use of modern instruments, retain simplicity of conception, or to make any sort of guess at performance when the plant is still at the design stage.

As a ship's machinery installation is a very complex arrangement, the task of building up a theory from simple fundamentals looked too formidable to attempt. Therefore a pragmatic approach was adopted, in which relatively crude measurements were made to highlight the most significant parts of the picture. This was followed by analysis using orthodox control techniques, often making quite gross assumptions for the sake of simplicity and the need for rapid results. This approach brings out the main factors and illuminates the interdependence between blocks of machinery, but of course it never yields a completely precise picture.

BOILERS—EXPERIMENTS IN H.M.S. ANON

Consider first boiler pressure. The main influences are heat supply from the fuel, and heat off-take in the form of steam output. To find the effect of fuel the boiler was steamed steadily at constant pressure for a few minutes and then the fuel flow was suddenly changed by switching a burner on. The effect on steam pressure is shown in Fig. 6. There is no

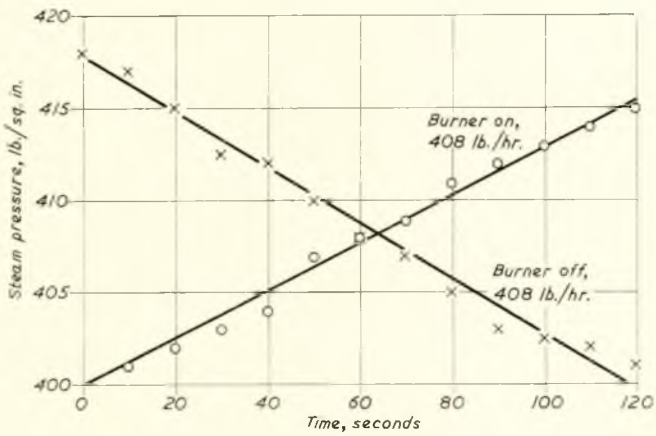


FIG. 6—*Stop watch record of boiler pressure*

significant time delay between the injection of fuel and its effect upon steam pressure. This suggests that, for practical purposes, the boiler behaves as a constant thermal inertia K which is suddenly subjected to an increase in heat supply from the fuel F. Assuming that, over the working range steam pressure is linearly related to the heat content of the boiler the relation between steam pressure and fuel flow may be expressed mathematically by equation (1):

$$\frac{dP}{dt} = KF \qquad (1)$$

To find the effect of steam off-take on drum pressure the boiler was steamed steadily for a few minutes and a sudden disturbance to steam flow imposed by a sudden opening of the manœuvring valve. The effect is shown in Fig. 7 from which it is seen that there is a negligible time delay between

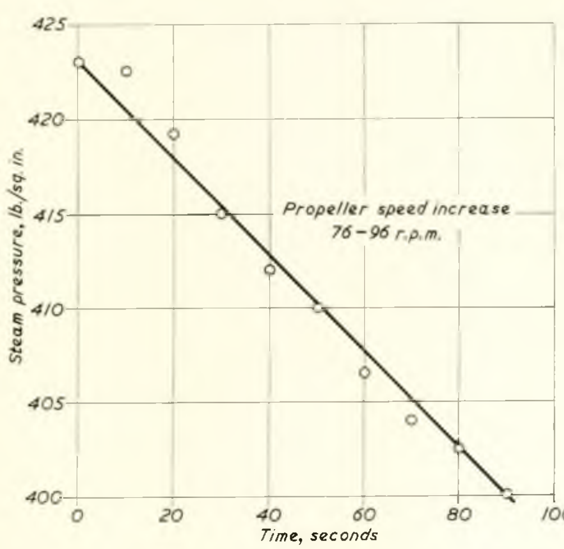


FIG. 7—*Stop watch record of boiler pressure—Sudden change in manœuvring valve*

the occurrence of steam off-take and its effect upon boiler pressure. The effect of steam flow on boiler pressure may be represented by equation (2):

$$\frac{dP}{dt} = -KS \qquad (2)$$

These two pieces of information permitted a reasonably accurate calculation of steam pressure fluctuation when the boiler was fitted with an automatic steam pressure control system.

Calculation of Steam Pressure Response

The assumptions made are rather sweeping, but are necessary if the whole problem is not to be weighed down with side effects whose quantitative significance is of secondary importance. The assumptions which were made are:

a) the thermal inertia of the boiler is constant;
b) the heat supplied to the boiler for evaporation is proportional to the fuel flow;
c) the thermal effects of feed flow are neglected;
d) steam within the boiler does not contribute to the thermal inertia of the boiler;
e) the steam pressure swings are sufficiently small to permit assumption of a constant latent heat and a linear relation between pressure and sensible heat of the water.

The following symbolic notation will be used:

M = Equivalent water weight of water content of boiler and pressure parts—lb.
P = Steam pressure—lb./sq. in.
S_o = Steam off-take from boiler at full load—lb./sec.
S = Steam off-take as a fraction of S_o.
L = Latent heat of steam at the operating condition—B.t.u./lb.
h = Sensible heat of water at the operating condition—B.t.u./lb.
F_o = Fuel flow to the boiler at full rated load—lb./hr.
F = Fuel flow to the boiler as a fraction of F_o.

When the boiler is steady steaming, heat supply will equal heat demand. This may be represented by equation (3):

$$S = F \qquad (3)$$

If a sudden step increase is made in steam demand δS, without changing fuel, the net rate of heat extraction from the boiler will be:

$$L\,\delta S\,S_o \text{ B.t.u./sec.}$$

This is obtained from the sensible heat of water within the boiler. In a short time interval δt these two quantities may be equated producing a reduction in sensible heat of the water δh, giving equation (4):

$$L\delta S\,S_o\,\delta t = -M\,\delta h \qquad (4)$$

or a reduction in pressure of δP given by:

$$L\delta S\,S_o\,\delta t = -M\frac{\partial h}{\partial P}\delta P \qquad (5)$$

where $\frac{\partial h}{\partial P}$ is a constant which may be obtained from steam tables. Equation (5) may be re-arranged to give the rate of change of pressure:

$$\frac{dP}{dt} = \frac{-LS_o}{M}\left(\frac{\partial P}{\partial h}\right)\delta S \qquad (6)$$

Similar reasoning may be applied if fuel flow is suddenly increased, without changing steam off-take, giving a rate of change of pressure:

$$\frac{dP}{dt} = \frac{LS_o}{M}\left(\frac{\partial P}{\partial h}\right)\delta F \qquad (7)$$

Equations (6) and (7) may be combined to give the more complete equation (8):

$$\frac{dP}{dt} = K(F - S) \qquad (8)$$

$$\text{where } K = \frac{LS_o}{M}\left(\frac{\partial P}{\partial h}\right) \qquad (9)$$

It is worth noting that the constant K is derived from basic design data of the boiler, so that a check may be made on the slopes shown in Figs. 6 and 7. The results of this check on K were:

K from fuel flow change (Fig. 1) 6·3 lb./sq. in./sec.
K from steam flow change (Fig. 2) 7·6 lb./sq. in./sec.
K from boiler design data 7·2 lb./sq. in./sec.

Steam Pressure Response in Automatic Control

The dependence diagram for the automatic control system used in H.M.S. *Anon* is shown in Fig. 8. Using conventional

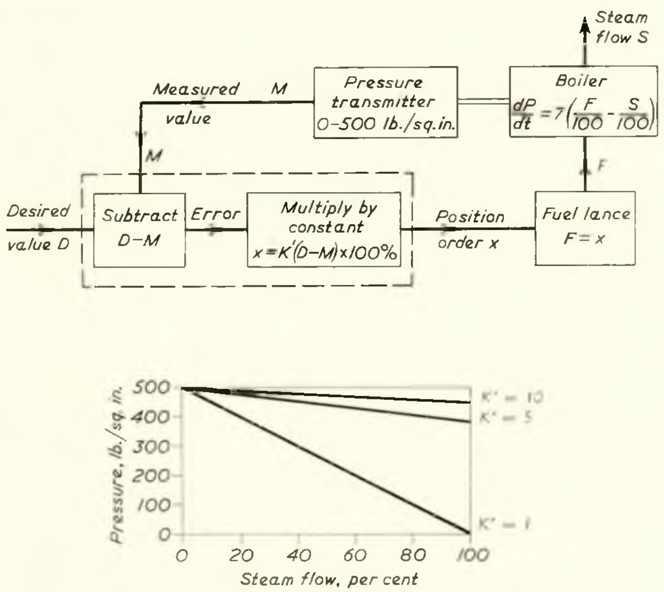


FIG. 8—*Dependence diagram for H.M.S.* Anon

analytical techniques based upon Laplace transforms and Bromwich inversions, the form of the steam pressure response following a change in steam flow can be calculated. In equation form these responses are complicated and unwieldy. For example, the steam pressure response following a sudden increase in steam demand S is given by the formula:

$$P(t) = SK \frac{2}{\sqrt{4\frac{A}{T} - A^2}}\, e^{\frac{-At}{2}} \sin\left[\frac{\sqrt{4\frac{A}{T} - A^2}}{2}\right] t \qquad (10)$$

and the fuel flow response by equation (11):

$$F(t) = S\left[1 + \frac{\sqrt{4\frac{A}{T}}}{\sqrt{\frac{4A}{T} - A^2}}\, e^{\frac{-At}{2}} \sin\left(\frac{\sqrt{4\frac{A}{T} - A^2}}{2}\, t \quad \tan^{-1}\frac{\sqrt{4\frac{A}{T} - A^2}}{A}\right)\right] \qquad (11)$$

where A is K divided by the proportional action factor (the proportional action factor is also known as the steam pressure control range or droop) and T is the integral action time in seconds.

Equation (10) was evaluated for one particular steam flow change on the ship and compared with experiment. The comparison is shown in Fig. 9.

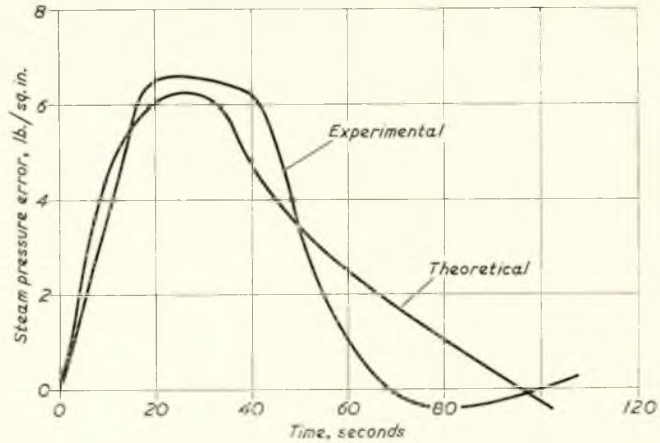


FIG. 9—*Comparison of theoretical and experimental values of pressure response—H.M.S.* Anon

Lessons Learnt from Initial Trials on H.M.S. Anon

The exercise showed that the performance of an automatic boiler control system was calculable on the basis of the thermal inertia of the boiler, the droop and the integral action time. Also it was shown that these calculations gave some insight into why's and wherefore's affecting performance. For example, the number of burners in use has a profound effect on steam pressure response because of the effect upon droop. The exercise also brought home the difficulty of making pen and pencil calculations on this subject, and indicated the need for some form of analogue computer support to reduce the effort spent on calculation.

Harmonic Response Trials at A.F.E.S.

Experience upon other ships broadly bore out the accuracy of the boiler representation established on H.M.S. *Anon*. In the spring of 1960, a more ambitious series of experiments was started at the Admiralty Fuel Experimental Station based upon harmonic response techniques. The boiler system is disturbed with a sinusoidal variation and the effect measured in terms of amplitude and phase. This technique yields a much finer insight into physical happenings within the boiler. A typical result is given in Fig. 10, where the effect of fuel flow

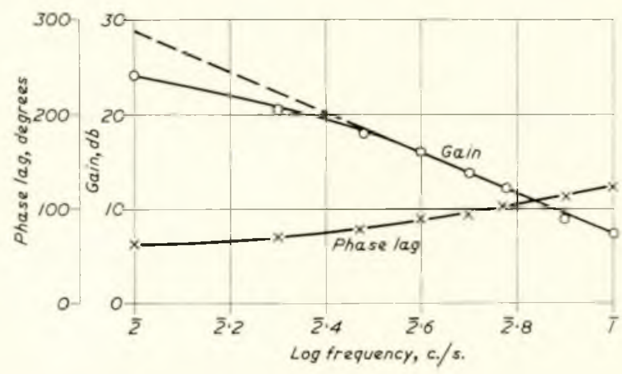


FIG. 10—*Steam pressure harmonic response to fuel flow*

variation upon steam pressure is shown. The most significant thing about Fig. 10 is that it indicates that the time lag between fuel flow change and its effect upon steam pressure is very small indeed, less than one second.

However, as these trials showed up very serious deficiencies in the dynamic response of valve operators and other control equipment which were of far greater immediate practical importance, their use for research into boiler characteristics was curtailed. In the autumn of 1962 a further comprehensive

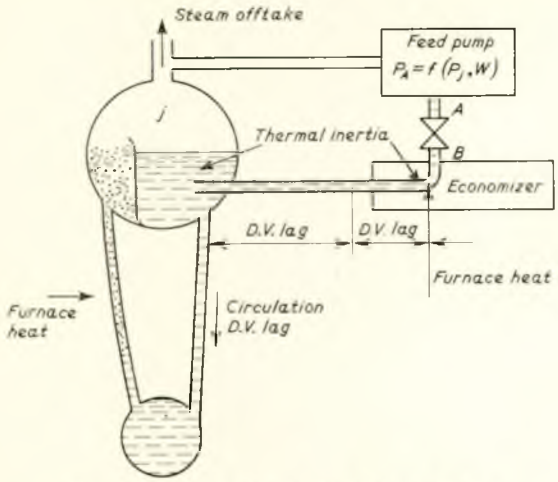
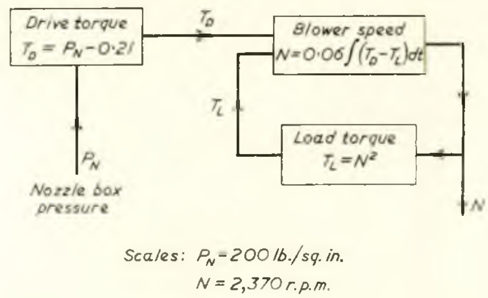


FIG. 11—*Physical model of D type boiler*



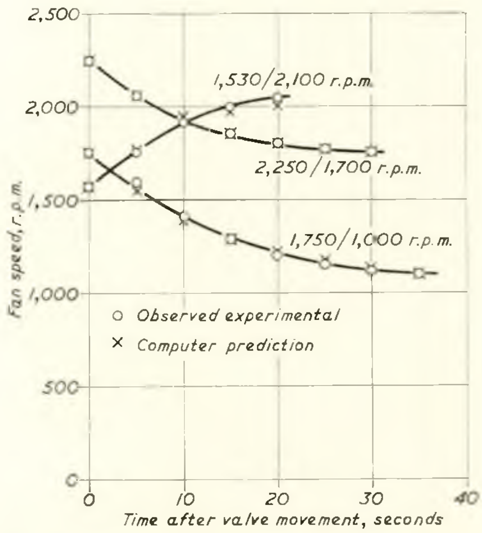


FIG. 12—*Stop watch records of blower speed and mathematical model*

series of harmonic response trials was carried out on another boiler at A.F.E.S. Although these results have not yet been fully analysed they do indicate that the physical model shown in Fig. 11 gives a fairly accurate representation of the effects of feed, fuel and steam flow upon steam pressure.

BLOWERS

Initial trials on H.M.S. *Anon* indicated that the blowers formed the most sluggish element in the combustion control system. At low powers the blowers were particularly slow off the mark, although some improvement was effected by imposing a closed-loop control system. Full advantage of closed-loop control could not be taken for two reasons. First, mechanical stops on the blower steam valve associated with integral action in the controller caused erratic performance at low power. Secondly, poor dynamic characteristics of the steam valve operator caused stability problems at higher powers.

As in the case of the boilers, a broad brush treatment was applied initially to account for sluggish blower response at low power. A few records of blower speed and air flow following sudden movements of the steam valve were taken using a stop watch. These are shown in Fig. 12.

Calculation of Blower Time Constant

An attempt was made to calculate the blower response on a linearized basis of small changes of speed and torque. If the blower is discharging into a constant aerodynamic resistance, the flow pattern through the blower will remain constant regardless of speed, provided compressibility effects are negligible. The blower torque will be proportional to the square of the speed and the time constant of the response of speed to a small increase in driving torque can be calculated as follows:

Let I be the rotational inertia in lb. ft.^{2}
Let N be the speed in r.p.m.
Let T be the torque in lb. ft.

at any chosen operating point:

$$T = K N^2 \qquad (12)$$

If a small increment of torque δT is applied, the resulting initial acceleration will be:

$$\frac{dN}{dt} = \frac{60g}{2\pi I}\,\delta T \text{ r.p.m./sec.} \qquad (13)$$

The ultimate increase in speed δN will be:

$$\delta N = \frac{\delta T}{2KN} \qquad (14)$$

Therefore the time constant will be:

$$\frac{\text{change of speed}}{\text{initial acceleration}} = \frac{2\pi I}{2\,KN\,60g} \text{ sec.} \qquad (15)$$

The time constant is inversely proportional to speed, thus accounting for sluggish response at low speeds.

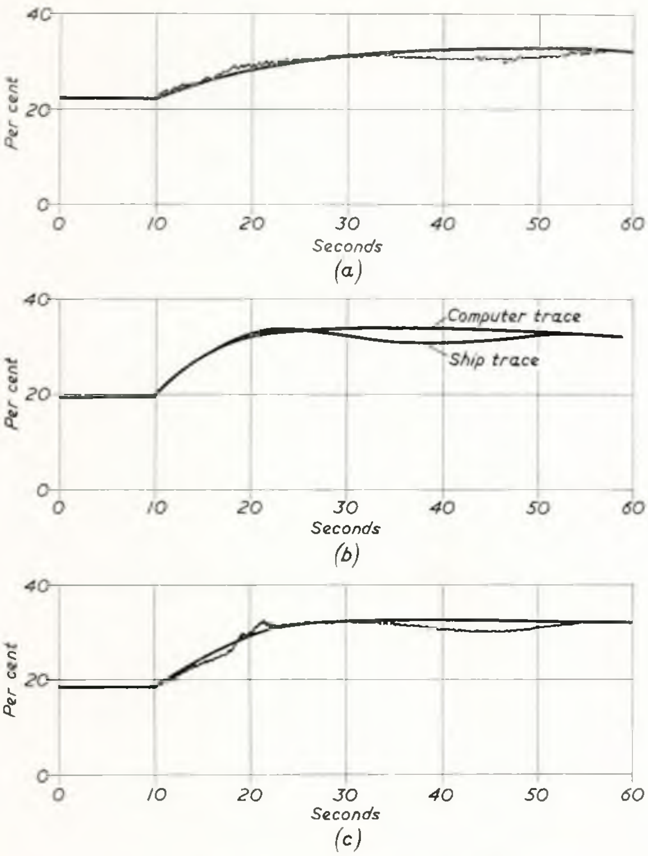
Mathematical Model to Account for Initial Records on H.M.S. Anon

The mathematical analysis given in the previous section indicates that blower characteristics are essentially non-linear, and therefore are not amenable to analysis by pen and pencil calculations. However, analogue computer techniques may be readily employed. In Fig. 12 a mathematical model is built up for the blower and the computer is used to sort out the required relation between the speed and time following step disturbances in turbine torque. The results of this comparison between experiment and theory are shown in Fig. 12.

More Advanced Trials on H.M.S. Anon

Encouraged by results using simple analogue computer techniques, a full scale effort was made to simulate in detail the air flow system on H.M.S. *Anon,* and to compare computed with experimental responses. The air flow control system was simulated in detail. Steam pressure control and fuel flow control systems were simulated rather less accurately. A number of tests was carried out on the computer to clear up outstanding problems, such as the ranging of the fan steam valve, the effect of limited speed of the fan steam valve and system stability under extreme operating conditions.

As a result of this initial computer study, controller settings were derived to give the best all round performance of the system. These settings were applied to the ship and a number of spot checks were carried out to see whether the ship system followed the computer prediction. Finally a step change of load was applied to one boiler on the ship and pen



a) *Air flow* b) *Master control signal* c) *Fuel flow*

FIG. 13—*Ship and computer pen records—Air flow-fuel flow*

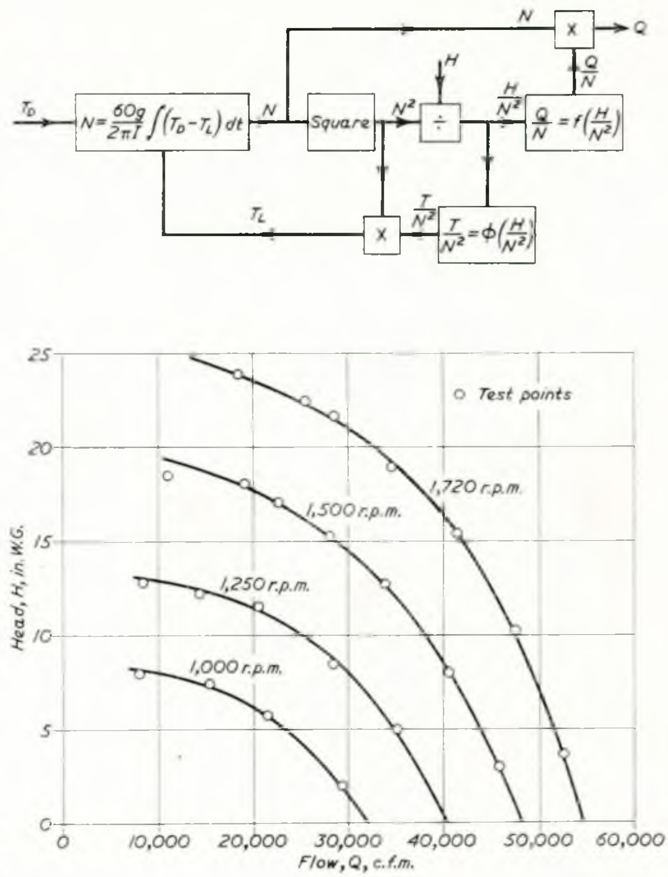


FIG. 14—*General mathematical model of boiler and computed characteristics*

records were obtained of air flow, fuel flow and master control signal. Considerable care was taken to exclude unwanted variables such as changing the number of burners, changing fuel pump steam supply, etc. The computer was adjusted to the conditions which prevailed during the ship's trials and the records given by the computer were compared with those taken on the ship. The comparison is shown in Fig. 13.

One or two useful lessons were learnt from this simulation. Firstly, it is physically impossible to include every detail in the simulation, therefore, the answers obtained depend on the judgement exercised in selecting the significant variables. Secondly, the vast amount of detailed numerical work required to prepare the computer programme, gives a very real insight into the physical problems associated with any particular installation. Really bad design features are highlighted at an early stage and quantitative precision can be given to design decisions.

General Mathematical Model for a Blower

Analogue computer techniques give a method of describing fairly completely and concisely the whole range of static and dynamic characteristics of a low head blower. For any particular flow pattern within the blower, the following proportional relationships hold:

Flow (Q) is proportional to speed (N).
Head (H) is proportional to square of speed (N^2).
Torque (T) is proportional to square of speed (N^2).

Therefore, the ratios $\frac{H}{N^2}$, $\frac{Q}{N}$ and $\frac{T}{N^2}$ will each be single value functions of the flow pattern and, therefore, of each other. For a blower it is only necessary to establish two of these ratios in terms of the third. This can usually be done from the published characteristics of Q/H and T/H. These relations, combined with a knowledge of inertia, enable the blower characteristics to be completely written into the computer. Any particular relation required may be extracted at will. As an example, the normal flow/head characteristics at constant speed have been written out by the computer and are given in Fig. 14.

It is suggested that a similar technique should be applicable to centrifugal pumps, such as feed pumps.

Simplified Turboblower Characteristics

In many instances, steam turboblowers do discharge through a fairly constant aerodynamic resistance and simpler relations may be used to relate steam flow, torque, head, air flow and speed over the working range. Fig. 15 shows such characteristics for a typical small turboblower, and it is seen that most of the characteristics are conveniently straight lines, which makes the simulation very easy. There is some loss of accuracy due to blade speed ratio effects which occur if the blower is accelerating very violently.

FURNACE FUEL OIL PUMPS

Turbo-driven positive displacement pumps, of the screw or gear type, are used to supply hot furnace fuel oil to burners at pressures ranging from 250-1,000lb./sq. in. Closed-loop control of pressure is effected by varying steam flow to turbine.

A fairly thorough analysis of the steady state characteristics of a particular turbo-pump was made. The analysis showed the following properties for the pump:

a) the pump can be regarded as producing a total displacement flow proportional to speed, a portion of the flow being absorbed by internal leakage within the pump;
b) the internal leakage flow is similar to that through a fixed orifice connected between pump outlet and inlet, and is proportional to the square root of the pump outlet pressure;

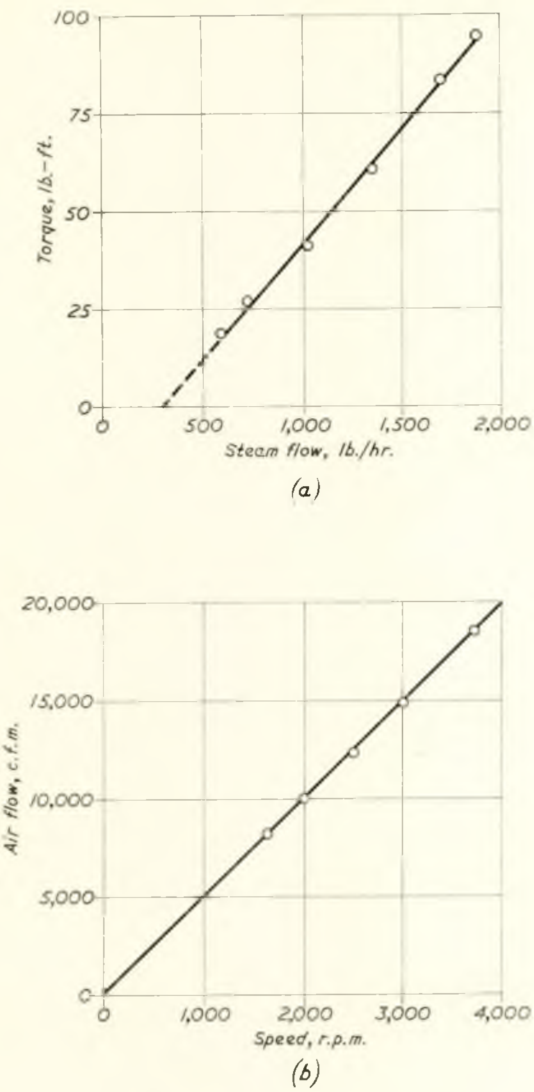


FIG. 15—*Simplified blower characteristics*

c) the hydraulic power absorbed by the pump is equal to its displacement flow multiplied by its outlet pressure;

d) friction torque was constant in this particular pump, independent of speed, probably because of an additional load imposed by a separate supply pump.

In the case of the turbine, which was a three-stage velocity compounded machine, operating over a comparatively narrow range of steam flows, the characteristics were reduced to fairly simple equations based upon nozzle box pressure. The stalled torque, mass flow and nozzle steam velocity may be calculated for such a turbine; typical characteristics are shown in Fig. 16. If the operating range of nozzle box pressure is restricted to 150/500lb./sq. in. the stalled torque is linearly related to nozzle box pressure. As the turbine speeds up from standstill, its torque will fall by a factor depending upon the ratio of blade speed to steam speed at nozzle exit. At full rated speed, which is normally the most efficient speed, torque will be reduced by some 30-40 per cent in a three-stage turbine. Over a 150/500lb./sq. in. range of nozzle box pressures the variation in nozzle steam velocity is 16 per cent. If the variation in initial steam velocity is ignored as far as the blade speed ratio is concerned, the running torque will depend only on the stalled torque and on turbine speed to an accuracy of $\pm$ 2·4 per cent. At a constant speed, torque should be linearly

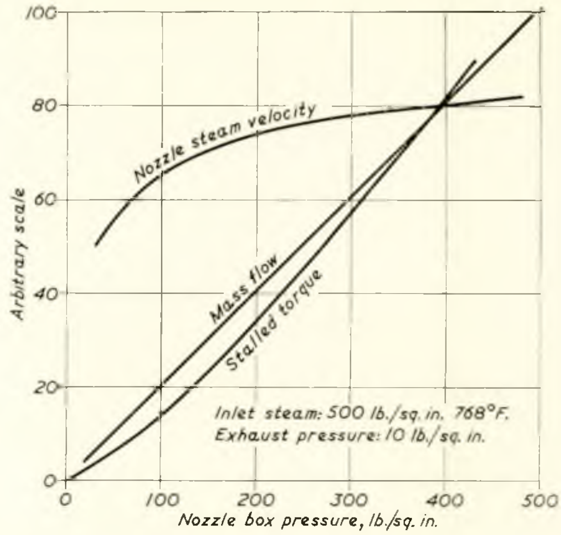


FIG. 16—*Calculated characteristics for a throttle controlled impulse turbine*

related to nozzle box pressure. Experimental results shown in Fig. 17 confirm this linear relationship. For this particular machine it was found that the steady state characteristics could be expressed by the following set of equations:

$$Q = 1{\cdot}10N - 26{\cdot}5\sqrt{P} \qquad (16)$$

$$T = 29 + 0{\cdot}0672P \qquad (17)$$

for the pump, and

$$T = (0{\cdot}365X - 16)\left(1 - 0{\cdot}35\frac{N}{2{,}400}\right) \qquad (18)$$

for the turbine:

Where Q is the pump output flow in gal./hr.
N is the pump speed in r.p.m.
P is the F.F.O. pressure at pump outlet in lb./sq. in.
T is the total load torque on the turbine rotor referred to the pump shaft in lb. ft.
X is the nozzle box pressure in lb./sq. in.
F is the displacement flow in gal./hr.

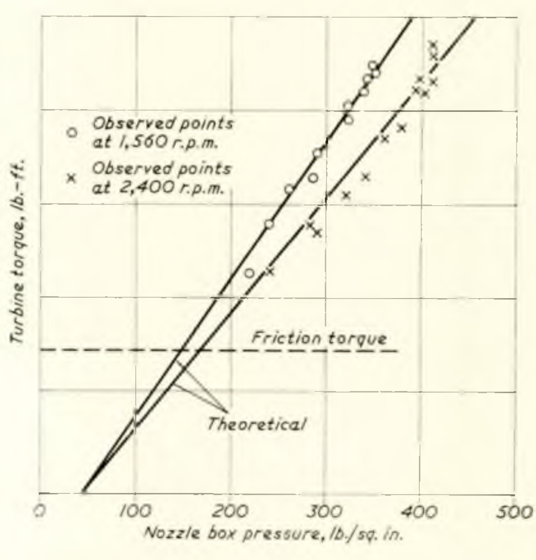


FIG. 17—*Experimental turbine characteristics*

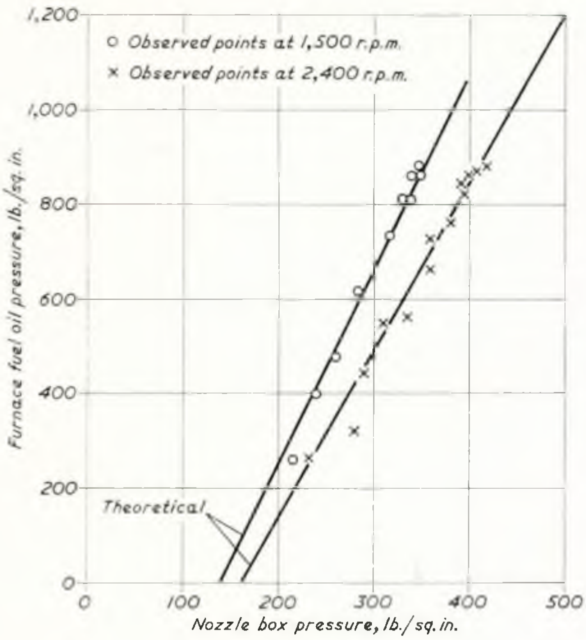


FIG. 18—*Experimental turbo-pump characteristics*

Calculation of Pump Time Constant

Because of the non-linear relation between pressure and flow, the time constant can only be calculated for small changes using linear approximations. If the pump is running at speed N with a displacement flow F, at a pressure P, and a small increment of torque, δT, is suddenly applied the initial acceleration will be:

$$\frac{dN}{dt} = \frac{\delta T 60g}{2\pi I} \text{ r.p.m./sec.} \quad (19)$$

The pump characteristics give the following proportionalities:

$$N \propto F$$
$$T \propto P$$

and the discharge orifice (including pump leakage), gives:

$$F \propto \sqrt{P}$$

therefore, the ultimate change in speed will be given by

$$\frac{\delta N}{N} = \frac{1}{2}\frac{\delta T}{T} \quad (20)$$

The time constant will be:

$$\frac{\text{change of speed}}{\text{initial acceleration}} = \frac{N}{2T}\frac{2\pi I}{60g} \quad (21)$$

Notice that for a given pressure (or torque), the time constant is proportional to speed, that is, the pump is more sluggish at high speeds.

For quick calculations it is often more convenient to express the time constant in terms of oil flow expressed in lb./hr., pressure in lb./sq. in. and speed in r.p.m. when the time constant becomes (for oil SG 0·9):

$$0{\cdot}239 \frac{IN^2}{FP} \text{ sec.} \quad (22)$$

A harmonic response test on a pump confirmed the accuracy of this formula.

SUMMARY OF SYSTEM ANALYSES

The preceding notes on boilers, blowers and pumps form a small sample of some early research and draw attention to the nub of any effective control system, that is, a quantitative knowledge of the machinery characteristics. Since that time, system analysis has been refined and expanded, particularly at the Yarrow-Admiralty Research Department, the Admiralty Fuel Experimental Station and at the Admiralty Engineering Laboratory. Some of the progress made has been reported elsewhere[1]. As many of the system studies are large in scope they could not adequately be covered in a paper of this sort; each could form a paper in its own right. The following notes are an attempt to outline the systems which have been studied in some detail giving close correlation between experimental, and theoretical responses derived by analogue computer.

Steam Superheater

A damper control superheater system has been analysed. By changing the mechanical drive linkages to give a linear relation between the control action and steam temperature, accuracy was improved by a factor of about twelve. A computer simulation of the control system was established which agreed with experimental characteristics measured by step and harmonic response techniques.

Spill Burner Fuel System

The burner system described by Brown and Thomas[1] has been simulated. The following equations were found to represent the burner characteristics:

For simplex operation: burnt flow $= 97{\cdot}7\sqrt{P_1}$ lb./hr. (23)

For spill operation: burnt flow $= 27\dfrac{P_2}{\sqrt{P_1 - P_2}}$ lb./hr. (24)

and supply flow $= 298\sqrt{P_1 - 0{\cdot}95\,P_2}$ lb./hr. (25)

where P_1 is the supply pressure in lb./sq. in.
and P_2 is the spill pressure in lb./sq. in.

Different variations of the control system were explored by computer and the performance predicted. A theoretical examination has been made of the performance using advanced types of furnace fuel oil pumps which are still on the drawing board.

Airflow Control Systems

Combustion air flow control systems for various ships have been simulated and optimized at the design stage. A design criterion for the dynamic response of blowers has been established. Detailed problems associated with multiple blower systems, the positioning of valve stops and overload requirements for acceleration purposes have been studied. Experimental work indicates that the tightness of the control loop is limited primarily by signal noise associated with the sensing of air flow. With blown boiler room systems, the time lag associated with filling the boiler room has not been found to be significant in the two cases which have been studied in detail.

Feed Water Control

Simple simulation methods have been established for the conventional three-element system. A comprehensive series of harmonic response tests has been completed in which the effects of feed flow, steam flow and fuel flow on boiler level and upon steam pressure have been studied. Examination of these results is giving an insight into the dynamics of boiler and economizer operation.

Methods have been established for eliminating the effects of roll and pitch of a ship on level indication. A method has been evolved for sensing indirectly the level at the centroid of the water surface. The virtues and snags of different methods of level indication, whether it be conventional or as suggested by Brown and Thomas[1], have been studied.

Overall Propulsive Plant Simulation

An attempt is being made by the Yarrow-Admiralty Research Department to combine all these system analyses into a complete simulation of the propulsive plant of a particular ship. The ultimate end in mind is to obtain the most effective acceleration of the ship through the water with the minimum strain on machinery. To this end, simulations are in an advanced stage for main turbines, condenser, hull and propeller dynamics, and various other items of propulsion plant such as de-aerators, etc. Apart from producing the most effective means of manœuvring the ship, it is hoped that this investigation will highlight any redundancies in control systems. For example, it is hoped that some simplification may be made in the rather tortuous procedure by which water is transferred from the main condenser into the boiler.

SUMMARY OF PRECAUTIONS IN DESIGN AND INSTALLATION

The most effective way of checking whether a system is properly organized at the design stage is by analogue computer. However, good basic design can be wrecked by poor installation or lack of attention to the fundamentals of control engineering.

System Organization

Each control should stand on its own feet. First-order elements (where there is no definite output quantity for each input quantity), must be enclosed within a local loop to reduce them to zero-order elements. The channels of signal flow must be clearly and concisely defined so that they may be readily understood by the operator. The use of open-loop feed-forward techniques to improve performance must be kept to a minimum.

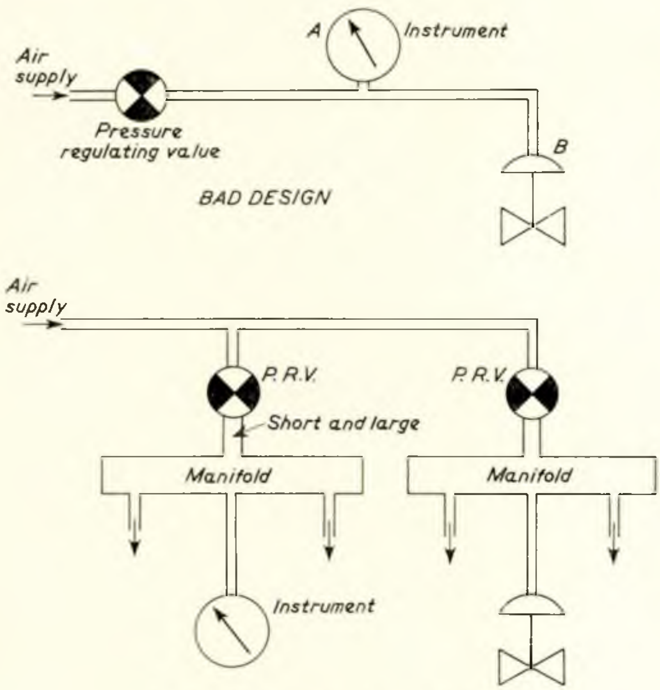


FIG. 19—*Servo air distribution*

Air Distribution

The importance of a correctly designed distribution system cannot be overstressed. A bad system is shown in Fig. 19, where for example, a valve operator at point B drawing a considerable quantity of air will affect the pressure at an instrument at point A, and set up unpredictable cross-effects. The right thing to do is also shown in Fig. 19; use a good pressure regulating valve with short large pipe to the manifold and then tap off from the manifold the various instrument supplies.

Instrument and operator air should never be derived from the same P.R.V. Generally a P.R.V. with a low droop should be used.

Control Valves

The main points to watch are as follows:

a) the valves must be sized for minimum as well as maximum flow;
b) single-beat valves are preferred because they give a tighter shut-off than double-beat valves;
c) do not use a control valve for isolating purposes, unless it is a single-beat valve with a hand jack;
d) the instrumentation system should be organized to require a valve having a standard characteristic such as logarithmic, linear or parabolic; errors in valve sizing are least damaging with the logarithmic and linear valves;
e) use well-designed rod gearing without backlash; where an operator is associated with a positioner, the rod gearing should be excluded from the minor loop; backlash within any control loop is absolutely deadly, except in one or two very unusual circumstances.

Sensing Points

The choice of sensing points is critical and can be difficult. While it is usually best to put the sensing point at the point at which precise information is required, this may be very damaging to the stability and speed of the control system. In fact the choice of sensing points is so intimately bound up with system design that it is usually best to make sure that the exact location is precisely defined at the design stage.

Other points to watch in connexion with sensing points are, not to sense steam pressure in such a way that water can "slosh" about between the sensing point and the transmitter, and not to use long lines to the transmitter when sensing low steam pressures; it takes some time for steam pressure to build up in a cold line. If long distances have to be traversed it is best to put the transmitter close to the point of interest and transmit the signal pneumatically.

Change-over Arrangements

The most important thing to aim at is simplicity in the details of the actual change-over operation. Sophisticated systems requiring a succession of adjustments and readings to change from manual to automatic control can be a nightmare to an operator with a nervous disposition. With care at the design stage, change-over from manual to automatic control may be effected by lining up two pointers on a duplex gauge and pushing a knob.

Saturation Effects

Automatic control systems are often blamed for poor performance when the machinery is flat out. Control is no substitute for unbalanced machinery design. As has been mentioned earlier, a controller having integral action should not feed an operator with mechanical limits, unless similar pneumatic limits are applied to integral action within the controller.

FUTURE PROSPECTS

Future hopes and plans are considerable in extent.

From the all-up plant simulation it is hoped to find out how to specify figures for optimum plant acceleration to give maximum ship acceleration with less wear and tear of machinery. To aid this it is aimed to develop a design technique using an overall dynamic criterion for plant design and hence feed controllability data and specifications to designers of machinery. A parallel result of this should be the simplification of machinery control processes and hence of the instrumentation and equally of the manual operating drill.

By applying the essential commonness in nature of control processes, with decision-making and computation at low energy levels, it is hoped to exploit standard instruments to do many control jobs which presently are done at high energy levels and, therefore, involve special devices in every case.

The fashionable subject of remote engine control from the bridge has been looked at, and this may be very profitable for naval vessels. Most such arrangements constitute the remote setting of a throttle rather than the remote control of shaft revolutions. The authors are more interested in the latter which, however, means closing the control loop around the engine. This, with steam turbine plant, is not altogether straightforward.

In regard to the higher levels of machinery management, there has been much thinking about information systems. This has been in three fields—first, the selection of parameters to be measured and the methods of measurement, second, the refinement and processing of the data gathered, and thirdly the presentation of the final information. It is felt that a fundamental examination of parameters which influence both operation and maintenance is necessary, taking account of new possibilities such as vibration measurement. There may well be a future in providing a facility for computing plant performance, at will. Alarm scanning and presentation are obviously advantageous facilities, and, if both these facilities are provided, the value of routine data logging is questioned. The possibilities in this field are very considerable and a firm effort is needed to keep one's feet on the ground, especially as there is not yet such an information system at sea in the Navy.

THE CONTROL ENGINEER'S PLACE IN MACHINERY DESIGN

Control theory is a body of technology which has only been formalized in the last fifteen or so years. It offers a completely fresh, powerful and cogent way of looking at almost any form of plant. Perhaps, for this reason, its exponents are apt to be regarded either with awe or with suspicion and scepticism. Either view will result in incorrect employment and organization. The simplest way of considering the problem leads to reasonable conclusions. Where manual control is the order of the day the plant designer alone can, and must, write the instructions for operating the installation and each machine. These instructions must be learned by the operator before he uses the machinery. When the operator becomes an instrument, the operating instructions remain as the designer laid down, and it is then the control engineer's job to teach the instrumentation to follow these instructions. However, his life will be a somewhat negative one if he loses any opportunity to suggest, to the plant designer, means by which operating instructions can be simplified These considerations force one to the conclusion that, first, expertise in control engineering theory, by itself, is only a partial qualification and must be accompanied by the ability to understand—or some actual experience of—the type of plant to be controlled. Second, that the control engineer must take an advisory position relative to the plant designer and, conversely, that the latter will be denying himself very worthwhile possibilities of progress if he neglects the extremely powerful tools which the former has to offer.

ACKNOWLEDGEMENTS

This paper includes a necessarily superficial and simplified account of a substantial body of work which has been carried out very largely by the staffs of the Admiralty Engineering Laboratory, the Admiralty Fuel Experimental Station and the Yarrow-Admiralty Research Department of Messrs. Yarrow and Co., with the considerable assistance of the ships' staffs of a number of H.M. Ships.

This paper is published by permission of the Admiralty, but the views expressed are those held by the authors and must not be construed as necessarily representing those of the Admiralty.

REFERENCE

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Service Trials with Prefired Quarl Blocks of High Alumina Content and the Economics of Various Front-wall Installations

W. McCLIMONT, B.Sc. (Member)*

The service performance of prefired quarl blocks of three materials, selected for their resistance to spalling, has been examined by the British Ship Research Association over periods of up to three years. The materials were a sillimanite refractory, a 60 per cent alumina refractory based on calcined bauxite, and a sillimanite-type refractory based on calcined kyanite. The results, when related to the chemical analyses, physical properties, and costs of the materials, were encouraging. The use of prefired quarls in a mouldable front has been shown to permit of easy replacement of the quarls.

Some data have been prepared indicating the relative costs of front-wall installations using different materials and forms of construction and these have confirmed general experience that the higher quality material, generally of higher initial cost, will prove the most economical.

INTRODUCTION

In view of the continued interest of users in prefired quarl blocks which appear to offer advantages in ease and cost of maintenance, further service trials have been carried out by the British Ship Research Association with three materials where resistance to spalling was expected to be better than that of the 38-42 per cent alumina-content firebricks which have been commonly used in the past. An earlier paper[1], concerned principally with the properties and service performance of mouldable and castable materials, gave some information on the performance of a sillimanite brick which was a 60 per cent alumina material with a percentage apparent porosity of 32 and a very high refractoriness (>1,800 deg. C.) This brick had behaved very creditably.

The present paper presents the results after one trial had been in progress for three years; the trial involving the other two materials was terminated after 2¼ years owing to the need for repairs to the boiler casing. Some data have been prepared indicating the relative costs of front-wall installations using different materials and forms of construction.

PROPERTIES OF THE REFRACTORY MATERIALS

The three materials included in the experiment were†:

K : A sillimanite refractory.

L : A 60 per cent alumina refractory based on calcined bauxite.

M : A sillimanite-type material based on calcined kyanite.

The chemical properties are given in Table I.

Table I.—Chemical analyses (per cent)

	K	L	M
SiO_2	33—37	32	40—42
Al_2O_3	61—64	62	55—57
Fe_2O_3	0·5	2·5	1·2

* Principal Assistant Chief Marine Engineer, British Ship Research Association.

† The code letters continue from earlier papers.

A number of physical properties has been obtained. The refractoriness of all three materials was greater than 1,800 deg. C. All three materials passed a prism spalling test of +30 cycles at 1,000 deg. C.

The coefficients of thermal expansion were:

K : $0·41 \times 10^{-5}$

L : $0·54 \times 10^{-5}$

M : $0·52 \times 10^{-5}$

Percentages apparent porosities on 9 × 4½ × 3-in. bricks:

K : 16-19

L : 28-29

M : 20-22

The permanent linear change after 2 hours at 1,500 deg. C. for each material is shown in Table II.

Table II.—Permanent linear change

Firing temperature, deg. C., and time	Percentage contraction (—ve) or expansion (+ve)		
	K	L	M
1,500 deg. C. for 2 hours	Nil to +0·20	+0·45	—0·20

SERVICE QUARL-BLOCK TRIALS

Two sister ships were used for the trials. Both are cargo vessels trading between U.K. and Continental ports and South Africa or India, averaging three voyages a year. Each vessel has two Babcock and Wilcox single-pass header boilers, each served by five Wallsend burners arranged in a straight line. The following data are applicable to all four boilers. The output per boiler is 42,500lb./hr. at service power. The furnace volume is 690 cu. ft. and the projected radiant heating surface 136 sq. ft. Oil is fired at the rate of 2,910lb./hr. at service power. There is forced draught and the air is preheated.

Ship 1—Material K

Four burner positions of the starboard boiler (the access quarl, No. 1, was excluded) were fitted in December 1959 with quarls of material K. The access quarl was of Firebrick A, a 42 per cent alumina-content brick of refractoriness 1,760 deg. C. and percentage apparent porosity in the range 22 to 25. The front wall around the quarls consisted of mouldable material (hereafter designated W) which was essentially a mixture of fireclay and ground firebrick classed as high-duty, containing some 38 per cent alumina, with a recommended service hot-face temperature of 1,450 deg. C. The thickness of the mouldable wall was about 9in. which is considered to be about 2½in. too thick.

During earlier quarl-block trials in this ship it had been found possible to take temperature measurements in the quarl blocks around the second burner in the port boiler (i.e. next to the access burner), and the maximum temperature recorded during trial coastal runs was 1,260 deg. C. with temperature fluctuations of up to 30 deg. C./min. These data are probably reasonably indicative of the general conditions during the present tests.

After one voyage to South African ports, the boiler was inspected in May 1960 and all the sillimanite quarls were found to be in excellent condition. The ship made three further voyages to South African ports during which the average forcing rate was 17 and did not exceed 19; the normal maximum capacity of the boiler is 21·4 with a maximum safe overload for short periods of 22·0. The boiler was inspected in April 1961, after the fourth voyage. The test quarls were found free from spalling and in good condition; the cone corners were sharp and there was no poker wear. There was a minor degree of surface attack over the bottom left segment of No. 2 quarl. The conical surface of No. 3 quarl was corrugated by surface attack, which was severe on the bottom segments and was thought to be the effect of unburnt fuel flaking off. There were suspicious surface fractures near the joint at 9 o'clock on No. 4 quarl and also about 1in. back from the furnace face of the bottom right quarl (3 to 4.30 o'clock); there was also slight surface attack on this quarl. No. 5 quarl was in particularly good condition with very slight surface attack between 2 and 4 o'clock. There had been spalling of some areas of the mouldable material between and above the quarls until the wall face had become flush with the faces of the quarl blocks. The boiler continued under similar service conditions and was inspected in March and May 1962. The sillimanite quarls were free from spalls and fractures except the bottom right segment of No. 4 quarl which was cracked about 1in. back from the furnace face, the crack running for 5in. from the horizontal joint; a suspicious surface crack had been noted at this position in April 1961. Although this deeper crack was observed in March, it had not developed further in May. There was a light build-up of slag all over the quarls, up to ⅛-in. thick; this increased to ¼-in. on both lower segments of No. 2 quarl, where the surface was rough and furrowed. Although no further opportunity for inspection had occurred, these quarls were reported in December 1962 to be in good condition. Their life at this point was three years.

Ship 2—Materials L and M

One quarl was installed at the No. 2 burner position of the starboard boiler, consisting of four segments of material L, and another at No. 4 burner position of the same boiler consisting of material M. The front wall around the quarls consisted of mouldable W, as in Ship 1. The access quarl aperture and the quarls at the two remaining burner positions were integral with the moulded wall. The thickness of the mouldable material was about 6½in. applied direct to the air-cooled steel casing although this is not considered good practice. A ½-in. thickness of asbestos millboard was used as a cushion for the test quarls at Nos. 2 and 4 positions. A two-course brick footing was used; there was a further part course of tiles below each prefired quarl. Surface cuts were not made in the rammed material. The installation was completed in May 1960.

In this ship also it had been found possible during earlier quarl-block trials to take temperature measurements in one of the quarl blocks. The maximum temperature recorded was a little higher at 1,340 deg. C. and the temperature fluctuations very similar.

Some details of the operating conditions during the first year of the test are given in Table III. Thereafter the boiler continued under similar service conditions, although precise details are not available as the ship was not sailing under the British flag for the next year.

Material L

At the first examination on 21st August 1960, this quarl was in sound condition except that the two lower segments were losing a surface layer about ⅛-in. thick. At the second inspection on 16th September 1960, this phenomenon was more extensive. About 20 per cent of the conical faces and 5 per cent of the furnace faces of each of the two lower segments had sustained a skin loss which was associated with crazy surface cracking. No

TABLE III.—TYPICAL OPERATING CONDITIONS OF SHIP 2.

Period	Days steamed at significant load	Weighted mean forcing rate	Maximum individual value of mean forcing rate	Number of times boiler lit from cold	Number of sailings and arrivals	Time boiler banked
15.5.60—21.8.60	48	16·5	18·2 (25 hrs)	15	15	9 days 5 hrs
30.8.60—15.9.60	Coastwise transits	16·4	17·8	3	4	Nil
22.9.60—27.12.60	50	17·3	19·9 (10 hrs) 19·6 (22 hrs) 20·7 (5 days) 21·6 (3 days 3 hrs) 19·1 (8 days 8 hrs)	9	17	12 days
6.1.61—28.1.61	Coastwise transits	17·2	19·5	7	6	5 hrs
6.2.61—10.5.61	49	17·8	21·3 (4 hrs)	11	16	10½ days

surface wash had been given to these segments; it was noted that the inception area coincided with a vitrified surface where bleeding had occurred of the compound used for bonding at the segment joints. At the third inspection on 3rd February 1961, it was found that about 80 per cent of the surface of the complete quarl had been affected by peeling of a layer about $\frac{1}{8}$-in. thick and by 12th May 1961, almost the entire surface had been affected. No further peeling developed after the first layer had been shed.

At the third inspection on 3rd February 1961, it was noted that cracks had developed which appeared to be of the nature that had been found in other materials to be associated with spalling type fractures. These were mainly located at the 12 o'clock and 6 o'clock positions. At the fourth inspection on 12th May 1961, a shallow spall about 1-in. deep had occurred at the furnace face in way of the butt joint of the two top segments. Spalling type fractures were developing about $\frac{1}{2}$ to $1\frac{1}{2}$-in. back from the furnace face at the bottom butt joint and in the cone face of the top left-hand quarl. At the fifth inspection on 20th February 1962, the fractures at the lower butt joint had caused a loss of material across the furnace face of the two lower quarls. This loss was not sufficiently serious to warrant replacement of the segments and the test was allowed to continue. The condition of this quarl was unchanged at the sixth inspection on 22nd May 1962, but in September 1962 when the test had to be terminated to allow repairs to the boiler casing it was considered that this quarl had reached the end of its useful life, as there were many fractures in it, and it would probably have been considered unwise to let it continue in service even if circumstances had not called for its replacement.

The millboard fitted behind the segments had disintegrated leaving a gap of nearly 1in. There were no indications that this had a deleterious effect.

Material M

The first inspection on 21st August 1960, showed the quarl to be in excellent condition. At the second inspection on 16th September 1960, a similar surface effect to that observed on material L was observed, again on the lower two segments, but was less extensive in area. Several small chips had been lost from the conical face. By the third inspection on 3rd February 1961, about 60 per cent of the surface had sustained a face peel about $\frac{1}{16}$-in. thick, and at the fourth inspection on 12th May 1961, face peeling had extended over the whole surface of the two bottom segments and was progressing on the two top segments. The whole of the cone surface had peeled by 20th February 1962, and the surface at this time was generally covered with a coating of adherent slag about $\frac{1}{8}$-in. thick. This thin peeling was not serious and no further peeling developed after the first layer had been shed.

Slackness of the lower left segment (viewed from inside boiler) was observed at the second inspection on 16th September 1961; this was due to release of the Wildish bolt, which was successfully retightened.

At all inspections up to 20th February 1962, the segments of this quarl were otherwise in excellent condition with the shape well maintained, the edges sharp, and no indications of spalling likely to develop.

On 22nd May 1962, the condition of the two right-hand segments was exactly as on 20th February 1962. The lower left-hand segment was loose, the Wildish bolt having failed, and the segment had moved forward about 2in. The upper left-hand segment was broken in four pieces, although none of the pieces had been dislodged. The major plane of fracture was parallel to the front face, through the bolt-head recess. The appearance of the fracture faces, which were examined in the laboratory, suggested purely mechanical damage. On removal of these two loose segments, the front of the casing was found to be distorted outwards over a length of some 4ft.; the casing plate showed no evidence of overheating. Subsequent examination of the pan of the furnace showed a similar distortion of the area adjoining the damaged portion of the front casing. The recent history of the operation of this boiler is a little obscure, because for the previous year the ship had not been sailing under the British flag, but the indications were that there had been a minor furnace explosion originating at the junction of the front casing and the pan behind the refractory. The loose lower left segment was refitted in position and a replacement segment of Firebrick A was fitted in the upper left position.

As already mentioned, tests in this boiler had to be discontinued in September 1962 to allow repairs to the boiler casing. Some trouble had again been experienced with the Wildish bolt in the lower left segment and it was considered that the recess in the segment was not well formed. In view of this and of the availability of only two other segments, it was decided not to incorporate this material in the rebuilt wall and to discontinue the test. At this point material M had been in service for 2 years 4 months and deterioration due to normal operation over this period had been so small that it was not possible to extrapolate a prediction of life for this material.

Remainder of Front Wall

The mouldable wall developed a large number of contraction cracks; in spite of the fractures the wall remained firm until the inspection on 22nd May 1962, when it was found to be loose in the area around Nos. 4 and 5 quarls where the casing was buckled. The condition of the wall contributed largely to the decision to repair the casing and rebuild the front wall completely in September 1962.

At the third inspection on 3rd February 1961, it was found that a surface skin about $\frac{1}{8}$-in. thick was peeling from the surface of the wall and by 12th May 1961, most of the wall surface had been affected in this way. Gross wastage was limited to the areas between the end burners and the side walls and by September 1962 the thickness in these areas had been reduced from $6\frac{1}{2}$ to 5in.

The moulded opening in the access position was between 1 and 2in. larger than the quarl ring attached to the firing door, and it is thought that the excessive clearance had contributed to the severe burning of the steel shroud noted at the first examination and which recurred several times. At the third examination it was found that two fractures had developed through the wall around the access opening at 5 and 7 o'clock and that the mouldable material was friable in the vicinity of the fractures.

At the second examination there were three contraction fractures across the conical face of No. 3 quarl and erosion by poker damage had occurred to a depth of about $\frac{1}{2}$in. at 5 and 7 o'clock positions in the throat. By the third examination, mechanical abrasion, probably due to clearing carbon, had caused some loss of shape from 3 to 7 o'clock and at 11 o'clock. At the fourth inspection it was noted that there was a marked loss of shape in the lower half of the quarl, and a face peel about $\frac{1}{8}$-in. deep had been sustained over the remainder of the cone surface. At the seventh inspection on 22nd May 1962, five contraction cracks were noted; there was poker damage up to $\frac{3}{4}$-in. deep in the throat from 5 to 7.30 o'clock and the shape was badly deteriorated at 3 o'clock. The cone surfaces were friable below a coating of fuel slag.

Three contraction fractures across the conical face of No. 5 quarl were noted at the first examination, together with some minor poker damage on the throat. On 3rd February 1961, it was found that there had been some general wastage of this quarl, the conical section being markedly concave from 1 to 10 o'clock and there was an increase of mechanical damage between 6 and 7 o'clock. The deterioration of shape continued and by 22nd May 1962, it was advanced, with heavy poker damage, up to $1\frac{1}{2}$-in. deep, from 4 to 8 o'clock.

DISCUSSION OF SERVICE PERFORMANCE

The surface peeling observed on both prefired materials L and M was an interesting phenomenon which may be related to an inevitable slight difference in texture between the surface and the interior of the blocks. It had been observed before in some high-grade materials, particularly mouldable, but during the present tests peeling was also observed from the surface of the mouldable material used in the front wall, which would not be regarded as a particularly high-grade material. There was no evidence with any of the materials tested of a second peel developing; second and sometimes third peels after very long periods have been reported by some observers. Peeling was not observed on quarls of material K, though some may have occurred on the lower segments and been obscured by rather heavy slagging.

This form of construction (prefired quarls in a mouldable front) has been shown by experience on Ship 2 to permit easy replacement of the quarls.

The conclusion should not be drawn that the higher the alumina content of a material the better it will be. In quarl blocks, spalling resistance is the most important criterion of quality and this property does not necessarily depend on alumina content. Recent investigations on thermal shock resistance have shown that it may be entirely independent of alumina content.

RELATIVE COSTS OF DIFFERENT CONSTRUCTIONS

With the assistance of Mr. J. Hall of British Wheeler Process Ltd. and Mr. E. F. Barton of British and Commonwealth Shipping Co. Ltd., some data have been prepared indicating the relative costs of front-wall installations using different materials and forms of construction. The labour costs quoted are considered realistic, but may be conservative and may well be higher in many ports.

The analysis has been done first for the front wall of a Babcock and Wilcox header-type boiler with five burners, and covers the installations in Ships 1 and 2. All costs are compared with the costs for a brick front wall (quality 'Firebrick A') with Firebrick A quarls which was the form of construction applicable when these boilers were designed. The following data have been used:

i) Firebrick A wall with Firebrick A quarls, repaired when quarls spall by replacing segments with same material:

Life of front wall	1·75 years.
Life of a set of quarls	0·875 years.
Cost of set of five quarls	£20.
Cost of material for remainder of wall	£70.
Labour cost for initial building	£85.
Total cost of initial building or of complete reconstruction	£175.
Cost of material to make good wall when fitting new quarls only	£11.
Labour cost for fitting new quarls only and making good wall	£34.
Total cost of fitting new quarls only and making good wall	£65.

ii) High-duty mouldable front wall with integral quarls, repaired when quarls lose shape by removing sections around burners and ramming with high-duty mouldable:

Life of front wall	3·5 years.
Repairs to quarls needed after	1·75 years.
Cost of material for front wall with integral quarls	£158.
Labour cost of initial building or of complete reconstruction	£90.
Total cost of initial building or of complete reconstruction	£248.
Cost of material to make good around burners	£34.
Labour cost for making good around burners	£50.
Total cost of making good around burners	£84.

iii) High-duty mouldable front wall with Firebrick A quarls, repaired when quarls spall by replacing segments with same material:

Life of front wall	3·5 years.
Life of a set of quarls	0·875 years.
Cost of set of five quarls	£20.
Cost of material for remainder of wall	£123.
Labour cost for initial building	£93.
Total cost of initial building or of complete reconstruction	£236.
Labour cost for fitting new quarls only	£29.
Total cost of fitting new quarls only	£49.

iv) High-duty mouldable front wall with prefired quarls costing 2½ times Firebrick A, repaired when quarls spall by replacing segments with same material:

Life of a set of quarls	2·25 years.
Cost of set of five quarls	£50.
Total cost of initial building or of complete reconstruction	£266.
Total cost of fitting new quarls only	£79.

v) High-duty mouldable front wall with prefired quarls costing five times Firebrick A:

Life of a set of quarls	4·5 years
Cost of set of five quarls	£100.
Total cost of initial building or of complete reconstruction	£316.

vi) High-duty mouldable front wall with prefired quarls costing seven times Firebrick A:

Life of a set of quarls	6 years.
Cost of set of five quarls	£140.
Total cost of initial building or of complete reconstruction	£356.

vii) Super-duty mouldable front wall with integral quarls, repaired when quarls lose shape by removing sections around burners and ramming with super-duty mouldable:

Life of front wall	5·25 years.
Repairs to quarls needed after	2·625 years.
Cost of material for front wall with integral quarls	£225.
Labour cost of initial building or of complete reconstruction	£90.
Total cost of initial building or of complete reconstruction	£315.
Cost of material to make good around burners	£49.
Labour cost for making good around burners	£50.
Total cost of making good around burners	£99.

viii) Super-duty mouldable front wall with Firebrick A quarls, repaired when quarls spall by replacing segments with same material:

Life of front wall	5·25 years
Life of a set of quarls	0·875 years.
Cost of set of five quarls	£20.
Cost of material for remainder of wall	£176.
Labour cost for initial building	£93.
Total cost of initial building or of complete reconstruction	£289.
Labour cost for fitting new quarls only	£29.
Total cost of fitting new quarls only	£49.

ix) Super-duty mouldable front wall with prefired quarls costing 2½ times Firebrick A, repaired when quarls spall by replacing segments with same material:

Total cost of initial building or of complete reconstruction	£319.
Total cost of fitting new quarls only	£79.

x) Super-duty mouldable front wall with prefired quarls costing five times Firebrick A, repaired when quarls spall by replacing segments with same material:

Total cost of initial building or of complete reconstruction	£369.
Total cost of fitting new quarls only	£129.

xi) Super-duty mouldable front wall with prefired quarls costing seven times Firebrick A:

Total cost of initial building or of complete reconstruction	£409.

The cumulative costs of these eleven arrangements for the first six years of service are shown in Figs. 1 and 2. Fig. 1

TABLE IV.—BABCOCK AND WILCOX HEADER-TYPE BOILER WITH FIVE BURNERS.

Total cost over 25 years of front-wall refractory, including quarls, with no allowance for loss of earnings on sums expended

Form of construction	Total expenditure	Percentage saving (compared with construction 1)
1. Firebrick A wall with Firebrick A quarls	£3,360	—
2. High-duty mouldable front wall with integral quarls	£2,324	31
3. High-duty mouldable front wall with Firebrick A quarls	£2,681	20
4. High-duty mouldable front wall with prefired quarls costing 2½ times Firebrick A	£2,415	28
5. High-duty mouldable front wall with prefired quarls costing five times Firebrick A	£2,212	34
6. High-duty mouldable front wall with prefired quarls costing seven times Firebrick A	£2,492	26
7. Super-duty mouldable front wall with integral quarls	£2,070	38
8. Super-duty mouldable front wall with Firebrick A quarls	£2,621	22
9. Super-duty mouldable front wall with prefired quarls costing 2½ times Firebrick A	£2,206	34
10. Super-duty mouldable front wall with prefired quarls costing five times Firebrick A	£2,214	34
11. Super-duty mouldable front wall with prefired quarls costing seven times Firebrick A	£2,045	39

TABLE V.—BABCOCK AND WILCOX HEADER-TYPE BOILER WITH FIVE BURNERS

Total cost over 25 years of front-wall refractory, including quarls, with ten per cent allowance for loss of earnings on sums expended

Form of construction	Total expenditure	Percentage saving (compared with construction 1)
1. Firebrick A wall with Firebrick A quarls	£7,807	—
2. High-duty mouldable front wall with integral quarls	£5,603	28
3. High-duty mouldable front wall with Firebrick A quarls	£6,400	18
4. High-duty mouldable front wall with prefired quarls costing 2½ times Firebrick A	£5,814	26
5. High-duty mouldable front wall with prefired quarls costing five times Firebrick A	£5,435	30
6. High-duty mouldable front wall with prefired quarls costing seven times Firebrick A	£6,123	22
7. Super-duty mouldable front wall with integral quarls	£4,932	37
8. Super-duty mouldable front wall with Firebrick A quarls	£6,171	21
9. Super-duty mouldable front wall with prefired quarls costing 2½ times Firebrick A	£5,423	31
10. Super-duty mouldable front wall with prefired quarls costing five times Firebrick A	£5,592	28
11. Super-duty mouldable front wall with prefired quarls costing seven times Firebrick A	£5,010	36

TABLE VI.—FOSTER-WHEELER D-TYPE BOILER WITH FOUR BURNERS

Total cost over 25 years of front-wall refractory, including quarls, with no allowance for loss of earnings on sums expended

(Also applicable approximately to Babcock and Wilcox selectable superheat-type boiler)

Form of construction	Total expenditure	Percentage saving (compared with construction 1)
1. Firebrick A wall with Firebrick A quarls	£7,532	—
7. Super-duty mouldable front wall with integral quarls	£4,900	35
8. Super-duty mouldable front wall with Firebrick A quarls	£5,378	29
9. Super-duty mouldable front wall with prefired quarls costing $2\frac{1}{2}$ times Firebrick A	£5,103	32
10. Super-duty mouldable front wall with prefired quarls costing five times Firebrick A	£5,188	31
11. Super-duty mouldable front wall with prefired quarls costing seven times Firebrick A	£4,860	35

TABLE VII.—FOSTER-WHEELER D-TYPE BOILER WITH FOUR BURNERS

Total cost over 25 years of front-wall refractory, including quarls, with ten per cent allowance for loss of earnings on sums expended

(Also applicable approximately to Babcock and Wilcox selectable superheat-type boiler)

Form of construction	Total expenditure	Percentage savings (compared with construction 1)
1. Firebrick A wall with Firebrick A quarls	£18,696	—
7. Super-duty mouldable front wall with integral quarls	£11,861	37
8. Super-duty mouldable front wall with Firebrick A quarls	£12,923	31
9. Super-duty mouldable front wall with prefired quarls costing $2\frac{1}{2}$ times Firebrick A	£12,381	34
10. Super-duty mouldable front wall with prefired quarls costing five times Firebrick A	£12,663	32
11. Super-duty mouldable front wall with prefired quarls costing seven times Firebrick A	£11,907	36

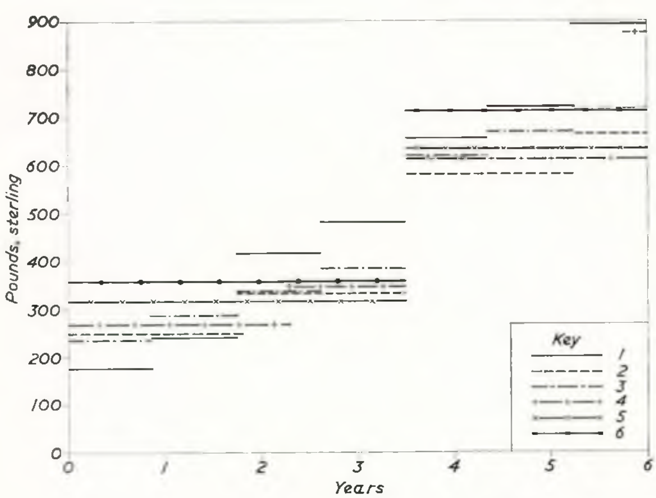

FIG. 1—*Babcock and Wilcox type with five burners. High-duty mouldable wall*

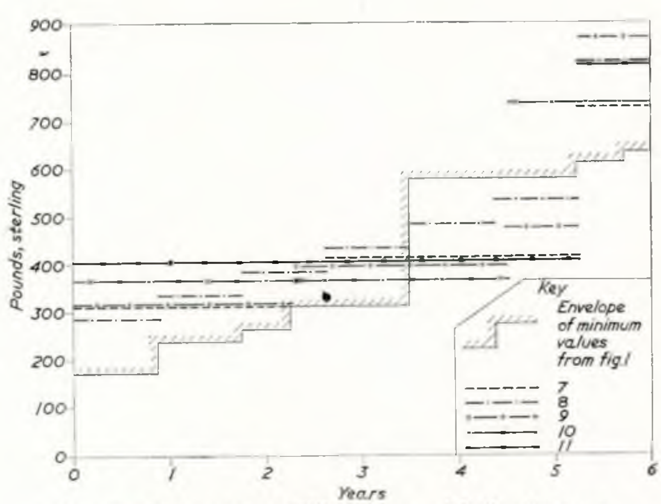

FIG. 2—*Babcock and Wilcox type with five burners.* Super-duty mouldable wall

covers the all-brick wall and the five alternative arrangements using high-duty mouldable material for the body of the wall. Fig. 2 covers the five arrangements incorporating super-duty mouldable. For ease of comparison, an envelope showing the lowest cost value at any time on Fig. 1 has been drawn on Fig. 2. It will be seen that, as the period of service increases, the more expensive materials tend to become the least expensive installations.

The data have been further considered for the life of a boiler, which has been taken as the life of a ship, which has in turn been taken as 25 years. Table IV gives the estimated cost of installing and maintaining in good condition for 25 years the front-wall refractory, including quarls, of a single Babcock and Wilcox header-type boiler with five burners. Once again the cost when using brick of Firebrick A quality throughout has been treated as the basic construction, and in column 4 of the table the percentage savings likely to be achieved by the use of each of the other ten constructions are shown.

It may be argued that, where higher first costs are incurred, allowance should be made for loss of earnings on the money thus tied up. This has been taken into account in Table V, where an allowance of ten per cent has been made for the loss of earnings on all expenditure incurred during the 25 years. It will be seen that it does not have any major effect on the relative costs of the various constructions. Similar data for a Foster Wheeler D-type boiler with four burners are given in Tables VI and VII. These data will also apply approximately to the Babcock and Wilcox Selectable Superheat type boiler.

When considering the use of less expensive materials, it should be noted that the labour charges quoted for repair and replacement are minimum figures based on the assumption that all quarls are attended to at one time and that there is no "waiting time" or overtime. If this assumption is invalid, labour charges may be anything up to twice those quoted. On the other hand, some of the repair work may be carried out by the ship's personnel with a consequent saving in labour costs. The use of more expensive materials introduces the risk of their full life expectancy not being achieved, owing to adventitious damage. It might well be reasonable to consider these factors self-cancelling and to regard these cost estimates as a fair prediction.

ACKNOWLEDGEMENTS

Thanks are due to the Research Council and Director of Research of the British Ship Research Association for permission to publish this paper.

Grateful acknowledgement is made of the facilities and information provided by shipowning companies and ship repairers and the materials provided by refractory manufacturers.

REFERENCE

1) McClimont, W. and Richardson, H. M. 1962. "The Properties and Performance of Quarl-block Materials." Trans.I.Mar.E., Vol. 74, p. 1.

INSTITUTE ACTIVITIES

Section Meetings

Merseyside and North Western

Joint Meeting

A joint meeting with the Barrow Society of Engineers was held by the Section on Thursday, 26th November 1964, in the lecture room of the Vickers-Armstrongs Training School, Barrow-in-Furness, at 7.30 p.m.

Mr. R. McVie, B.Sc. (Corresponding Member, Barrow) was in the Chair and about one hundred members of the Institute and the Society were present.

Mr. J. F. Harrison, O.B.E., presented his paper, entitled "The Development of the Diesel Engine for Rail Traction", which was very interesting and was well received. A lively discussion period followed.

Mr. S. B. Cooper (Member) proposed a vote of thanks to the author, which was greeted with acclamation.

This meeting marked a step forward in the activities of the Institute in this part of the North West, in that it was the first senior meeting held in Barrow.

North East Coast

Works Visit—The Doxford J Type Engine

By kind permission of Wm. Doxford and Sons (Engineers) Ltd., members of the Section were shown the Doxford J9 engine running on full load test at Pallion Works, on Thursday, 26th November 1964, at 3.30 p.m. Members of the North East Coast Institution of Engineers and Shipbuilders had also been invited by the Section. Approximately 150 members and visitors were given full facilities and attention, and were also entertained to tea.

It is regretted that the vast student interest in the works visit could not be met on this occasion. As it was, four times the estimated number were present on the engine test site.

General Meeting

A general meeting of the Section was held on Thursday, 26th November 1964, in Monkswearmouth Hall, Sunderland Technical College, at 6.15 p.m.

Mr. D. H. Sword (Chairman of the Section) was in the Chair and 225 members and visitors were present.

The Chairman introduced Dr. M. Hutton, Principal of the College, to the assembly. Dr. Hutton welcomed this general meeting, which was an inaugural meeting for the new hall, and, in his speech, extended an invitation for further events of this nature.

In reply, the Chairman thanked Dr. Hutton for the excellent facilities and arrangements offered in the magnificent new building. He then went on to thank Wm. Doxford and Sons (Engineers) Ltd. for the unique works visit which had taken place that afternoon. The Chairman then extended a welcome to members of the North East Coast Institution of Engineers and Shipbuilders, who had been invited to attend the meeting.

After seeking and gaining approval for the minutes of the previous meeting, the occasion of the presentation of the Presidential Address of Mr. A. Logan, O.B.E., the Chairman introduced Mr. P. Jackson, M.Sc. (Member) who presented his paper entitled "The Doxford J Type Opposed Piston Marine Oil Engine—Testing Experiences". In his presentation, which lasted for one hour, Mr. Jackson gave the full paper. He then spent a further 40 minutes in dealing with the discussion in seriatim. Mr. G. Yellowley (Member) opened the discussion followed by Mr. Appleby, Dr. F. Ørbeck and Mr. Robertson.

A sincere vote of thanks to Mr. Jackson, proposed by the Chairman, was accorded prolonged applause.

Northern Ireland Panel

A general meeting of the Panel was held on Tuesday, 27th October 1964, in the Millfield Building of the College of Technology, Belfast, at 7.00 p.m., when a paper, entitled "Interesting Investigations" by J. H. Milton (Member of Council), was presented by the author and discussed.

Mr. D. H. Alexander, O.B.E., F.C.G.I., M.Sc., Wh.Sc. (Local Vice-President, Belfast) was in the Chair and about fifty members and visitors attended the meeting.

A vote of thanks to the author was proposed by Mr. J. W. Bull (Member) and was greeted with acclamation.

The meeting closed at 9.00 p.m.

Scottish

A general meeting of the Section was held on Wednesday, 16th December 1964, in the Weir Hall of the Institution of Engineers and Shipbuilders in Scotland, 39 Elmbank Crescent, Glasgow, C.2., at 6.15 p.m.

Commander A. J. H. Goodwin, O.B.E., R.N. (Chairman of the Section) was in the Chair and eighty-seven members and visitors were at the meeting.

After extending a welcome to those present, the Chairman introduced the speaker for the evening, Mr. E. G. Hutchings, B.Sc. (Member) and invited him to present his paper entitled "Modern Boilers".

Mr. Hutchings referred to his previous paper—"The Design and Development of Two-drum Marine Boilers"—which he read before the Institute in London on Tuesday, 13th November 1962. He said that his paper to the Section might be considered as an appendix to that earlier paper. The author dealt with furnace design and combustion which, he said, were of major importance in any boiler design. Emphasis was placed on a number of typical marine furnaces, from the smallest to the largest, currently in use and projected, which revealed that over a whole range, the "oil per square foot" was a function of the rate of heat transfer to furnace tubes within plus or minus five per cent.

Mr. Hutchings went on to elaborate on superheater design and superheat control in relation to current boiler design and development.

He summarized his talk by saying that two-drum boiler design had developed to a point where boilers could be offered which would be free from the troubles experienced in the past, due to bonded deposits on superheater tubes. To do this, attention must be given to the furnace design and the choice of combustion equipment, and long retractable soot blowers should be included. Corrosion of air-heaters could be overcome by the use of vitreous enamel, so that the way was open to use higher feed temperatures and steam pressures. A new approach to the design of drums might also encourage higher steam pressures.

Reheat was likely to prove more acceptable and boiler maintenance would be less.

All these factors contributed to a steam installation which, at 25,000 s.h.p., would show a better economy than a Diesel engine, providing that the cost of lubricating oil and the effect of propeller speed were taken into account. In fact, it was likely that the steam installation of the near future would be able to compete with the present day Diesel in terms of running cost alone, at powers below 22,000 s.h.p., so that the days of the big Diesel might indeed be numbered.

The discussion was opened by Mr. D. S. McFarlane (Member) and there were many speakers, on pertinent points that were dealt with by the author in a very firm and confident manner.

Mr. J. Laing (Member) proposed a vote of thanks to Mr. Hutchings, which was received with acclamation.

The meeting closed at 7.50 p.m.

South East England

A general meeting of the Section was held on Tuesday, 1st December 1964, at the Clarendon Royal Hotel, Gravesend, at 7.30 p.m.

Mr. G. F. Forsdike (Chairman of the Section) was in the Chair and forty-five members and guests were present.

A paper entitled "The Design and Construction of the Trawsfynydd Power Station" was presented by the author, Mr. J. N. Bishop. The lecture started with Mr. Bishop running through the design of the power station and some of the problems which the site itself imposed upon the design, which he illustrated with a series of slides. This was followed by a film covering one year of the construction, during which the twelve heat exchangers and reactors were built and installed. The film illustrated the great thought and planning which had gone into the construction, with shots that over-simplified the task of lifting and placing the 375-ton heat exchangers in position.

The interest in the subject and the manner in which Mr. Bishop presented his paper was evident from the prolific and spontaneous questions put forward. The author, with his wonderful command of the subject, was able to satisfy the interest of all those present.

South Wales

A general meeting of the Section was held on Monday, 30th November 1964, at the South Wales Institute of Engineers, Park Place, Cardiff, at 6.00 p.m.

Mr. T. C. Bishop (Chairman of the Section) presided over the assembly of nearly ninety members and visitors, from Cardiff, Swansea and Newport, among whom were students from the technical colleges in the region covered by the Section.

Mr. P. Jackson, M.Sc. (Member) was the speaker for the evening and presented his paper entitled "The Doxford J Type Opposed Piston Marine Oil Engine—Testing Experiences". Mr. Jackson's excellent lecture was generously illustrated by slides and evidence of the interest aroused was shown during the discussion period which followed the presentation, when the author ably answered twenty-two questions concerning the paper.

Mr. D. Skae (Vice-President) proposed a vote of thanks to the author, which was accorded an enthusiastic response. Mr. A. W. G. Long (Associate Member) proposed a vote of thanks to the Chairman for presiding at the meeting, which then concluded.

West of England

A general meeting of the Section was held on Monday, 14th December 1964, at the City of Bath Technical College, at 7.00 p.m., when a paper entitled "Ship Repair Practice" by L. Scaife (Member) was presented by Mr. F. C. Tottle, M.B.E. (Local Vice-President, Bristol) and discussed.

Mr. J. P. Vickery (Vice-Chairman of the Section) took the Chair at the meeting, which was attended by twenty-eight members and visitors.

The paper, which was written from a practical point of view, gave a short summary of the practice of shipbuilding and ship repairing in United Kingdom and Continental ports, and dealt with all aspects of shipyard work.

The paper stressed the need for co-ordination of all sections of the shipyard towards a single purpose and also mentioned the many functions of the works manager, on whose judgement and good guidance the success of the shipyard depended.

The lecture was most interesting and enlightening, and many questions were asked, all of which were ably answered by the speaker.

A vote of thanks to Mr. Tottle was proposed by the Chairman and the meeting ended at 9.00 p.m.

Election of Members

Elected on 16th December 1964

MEMBERS

George Harold Baker
John Burgess, Lt. Cdr., R.N.
Alan Craig
Charles Nicol Crockett
Thomas Darling
Stanley Lidsay Foreman, Lt. Cdr., R.C.N.
John H. Govan, B.A.Sc. (Toronto)
Roy Frederick Charles Gregson
John William Dodd MacGregor
Fred Pennington
James Hamilton Robertson
Dennis Scruton
Jawhar Kishinchand Sippy
Robert Stewart
Frederick William Tonkin
Sidney Reynolds Williams, Eng. Lieut., R.N.

ASSOCIATE MEMBERS

Joseph Young Allman
Norman Vivian Almy
Alan John Bacon
Peter Evelyn Barratt
Edward Barritt
Balram Singh Bedi
Arnold Bennett
Ivan Desmond Chapman
David Frederick Coath, Eng. Lieut., R.N.
Joseph Peter Daily
Edward Ronald Evans
Alan James Garlinge
Robert Jordison Hayton
Alan Hockey
Goronwy Owen Hughes
Chandra Kumar
Kenneth Lea
Frederick Terence Lindley
John Livingstone
William Albert Mason
William Todd Nicol
David Mellor Payne
Evan Edward Farr Ralph, Eng. Lieut., R.N.
John Hylton Rennison
Roy Ian Scaife
Albert Christopher Smith
Venkataraman Srinivasan
Terrence Andrew Scully, Eng. Lieut., R.M.N.
Eric Stubbs
Mailvaganam Robert Tharmaseelan, Lieut. (E), R.Cey.N. B.Sc. (Eng.) (London).
Kenneth Varlow
Barrie Elsworth Welbourn
John Boris Williams
John Elwyn Williams
Frederick Edward Wood, Eng. Lieut., R.N.

ASSOCIATES

Kenneth George Finlayson

John Graham Martin
Muhammed Meeran Mohiuddeen
Horace Leslie Perkins

GRADUATES

Stewart Marr Adamson, B.Sc. (Hons) (Newcastle)
Terence Douglas Bedford
Peter John Cain
Douglas Dudley Grayling
Michael Haynes
George William Hotson
John Martin Jones
Mashooque Ali Khan
Kwong Kwok Yui
James Murray Meldrum
John Michael Howard Saint, Lieut., R.N.
Anton Hugh Singarayer
John Rendle Smith, Lieut., R.N.
Brian Alan Wickens, Lieut., R.N.
Gilbert Wilson
Edward Wood

STUDENTS

Josephus Johannes Baghurst
Lionel Peter Bateman
David John Bissland
Frederick Ronald Braithwaite
Richard Anthony Chappell
Chan Che Ching
John Patrick Cowan
David Ronnie Gregory
David Wood Hargroves
Patrick Colm Keogh
Edward John Larcombe, B.A. (Cantab.)
Robert Ward

PROBATIONER STUDENTS

John George Atkinson
Colin Richard Bleach
Jeremy Brian Callon
Richard John Clark
Alan Geoffrey Clarke
Philip John Daniels
Paul Arthur Efford
David Sargeant Evans
John Caswell Flaherty
Alan William George
Peter Robert Goodfellow
Phillip Stewart Grant
Michael Wyndham Harries
Ian Irvine
William James Hobbs
Denis Keith
Gerald Campbell Kirton
Terence John Kirton
Allan David Lynch
John Alexander MacDougall
Carl Sandford Gustaveson Mayl
Keith Purvis
David James Ramsay
Peter Read
James Alexander Reynolds
Robert Clark Robertson
John Austin Saddington
Brian Siddall
Malcolm Douglas Branson Start
Colin McInnes Styles
Martin Donald Tulloch
Emlyn John Williams
Gerald Robert Williamson

TRANSFERRED FROM ASSOCIATE MEMBER TO MEMBER

Kenneth Arthur Craven
William Sidney Ellis
Charles George Edward Gudge
Norman Harrison
Arthur John Orpe
David Bryce Stables
John Summersgill
Frank Henry Trenchard
Sydney Charles Wells
Ivan William Charles Westaway
Jack Wetherell

TRANSFERRED FROM ASSOCIATE TO MEMBER

James Fulthorpe
John Frederick Jones
Charles Douglas Kersey
Derek John Smart

TRANSFERRED FROM GRADUATE TO MEMBER

Stirling Macneill Ross, Lt. Cdr., R.C.N.

TRANSFERRED FROM ASSOCIATE TO ASSOCIATE MEMBER

Charles Robert Flack

TRANSFERRED FROM GRADUATE TO ASSOCIATE MEMBER

James John Dobie
Peter James Foster
Nitte Prasanna Kumar Hegde, Lieut. (E), I.N., B.E. (Mech.), Madras
Dermot Charles Hogan
Rajendra Kumar Gupta, Lieut., I.N.
John Alexander Innes
Duncan Ramsay McKellar, B.A. (Cantab.)
Colin Strangwood Mason
John Albert Mathews
Walter Henry Maxwell, Lieut., R.N.
Kenneth James Overton
Puttichanda Poovaiah Vijay
Harold Swincoe Scott
Peter Barrie Walker

TRANSFERRED FROM STUDENT TO ASSOCIATE MEMBER

Michael Bulkeley Smalley
Alfred Roy Croft

TRANSFERRED FROM PROBATIONER STUDENT TO ASSOCIATE MEMBER

Michael Raymond Temple

TRANSFERRED FROM STUDENT TO GRADUATE

Richard Bruce Bellchambers
Edward Michael Davis
Sateesh Bhalchandra Lele, B.Sc. (Hons) (Durham)
Thomas Anthony Machell
Anthony John Rainbow
John James Waugh

TRANSFERRED FROM PROBATIONER STUDENT TO GRADUATE

Norman Lowrey Bell
Roger Keith Gibbs

TRANSFERRED FROM PROBATIONER STUDENT TO STUDENT

Colin Michael Attwood
Graham Dadd
Colin Charles Knill
Peter John Matthews

OBITUARY

Daniel Mackay Bain (Member 8588) was born on 25th August 1908. After serving an apprenticeship with John Brown and Co. Ltd., from 1918 to 1923, he went to sea as an engineer with the Orient Steam Navigation Co. Ltd.; from 1928-1937 he served in Royal Fleet Auxiliary vessels. During his sea service he gained a First Class Board of Trade Certificate. In 1937 he became a shift engineer at the Ford Motor Co. power station at Dagenham and later was made assistant foreman there. He accepted an appointment as an engineer surveyor with the Cornhill Insurance Co. Ltd., in 1940, and remained in the service of that company until the time of his death, on 12th August 1964. He served in the Newcastle upon Tyne area, also in Northern Ireland, and, since 1947, had been working in Southern England. He was well esteemed by his business colleagues and all others who knew him.

Mr. Bain was elected a Member of the Institute on 7th February 1938. He leaves a widow.

Denis Charles Crump (Member 9747) was born on 12th February 1903. He served his apprenticeship with Swan Hunter and Wigham Richardson Ltd. A few years later he had achieved the position of assistant outside manager with the Middledock and Engineering Co. Ltd., after which he was assistant manager to Camps and Co., for three years.

From 1933 to 1940, he was a consulting marine surveyor, and in the latter year became a partner in the firm of Walker, Crump and Co. with offices in London. He continued in the business after his partner had retired in 1955.

Mr. Crump, whose death occured on 18th September 1964, was elected a Member of the Institute on 7th December 1943. He was also a Fellow of the Society of Consulting Marine Engineers and Ship Surveyors, a Member of the Royal Institution of Naval Architects, and an Associate Member of the North East Coast Institution of Engineers and Shipbuilders.

He leaves a widow.

Robert Henry Gummer (Member 7008) died in hospital on 25th August 1964 at the age of eighty.

Mr. Gummer, who served his apprenticeship with Humphrey Tennant and Co. Ltd., from 1899-1906, was the holder of a First Class Board of Trade Certificate. He was, at one time, an electrical engineer with the London County Council. He was also works manager and chief engineer to Taylor Bros. and Co. Ltd. for five years from 1919.

In 1925, he joined International Combustion Ltd. as assistant general manager and became a director and contracts manager in 1940. He retired from executive duties in 1954, after nearly thirty years of service with the company.

Mr. Gummer was elected a Member of this Institute on 14th March 1932. He had also been a Vice-President and Treasurer of the Institute of Fuel.

Donald Gordon MacDougall (Member 6363) died suddenly in April 1964, leaving a widow and a son. He was elected an Associate Member of the Institute in 1929 and transferred to full membership on 21st March 1944. He was also an Associate Member of the Institution of Mechanical Engineers.

Born on 11th June 1898, Mr. MacDougall was educated, to matriculation standard, in Glasgow and, subsequently, took a five-year engineering diploma course at Ipswich Technical College, Queensland, Australia. He served an apprenticeship, from 1916 to 1921 at the Railway Works at Ipswich, Queensland.

On completing his training, he took up employment as a draughtsman, first with a large general engineering concern, then with a firm of consulting engineers and, lastly at an armament works. In 1929, he went to sea with the Commonwealth and Dominion Line as second refrigeration engineer, joining Shaw, Savill and Albion and Co. Ltd. three years later as fifth engineer. From 1934 to 1937, he was with the Aberdeen and Commonwealth Line as sixth and fifth engineer. During his seagoing career he gained a First Class Board of Trade Certificate with Motor Endorsement. In 1938, having left the sea, he was engineer-in-charge with a firm of power plant engineers, and, two years later, had joined the Air Ministry Works Directorate as a mechanical and electrical engineering assistant. He remained with the Air Ministry until 1945, when he became associated with an engineering concern as assistant works engineer, later as assistant engineer. In 1952, he joined a firm of mechanical handling engineers as a designing engineer draughtsman.

Cyril Wright Parris (Member 14377) died suddenly on 26th September 1964 in his sixty-first year. He leaves a widow.

He was apprenticed, from 1919 to 1923, to E. Green and Son Ltd., and, during this time, studied engineering at Wakefield Technical College and Leeds University. After completing his indentures, he remained in the employment of the company, being engaged on various power station building projects in Great Britain. During the next twenty years he was successively, a draughtsman engaged on special tool design, a section leader and contracts engineer on power station contracts, and chief designer for marine and power station projects. In 1943, he became manager of the company's power station and marine division. At the time of his death, he had been with the company for forty-five years and held the position of technical manager; he was also an associate director. From 1926 to 1943, he had lectured in higher mathematics, heat engines and strength of materials at Wakefield Technical College.

Mr. Parris was elected a Member of the Institute on 11th May 1953 and rendered service as a member of the Committee of the North Midlands Section, to which he was elected in 1960. He was also a Member of the Institution of Mechanical Engineers and of the Institute of Fuel.

Lieutenant-Commander Francis Henry Peck, M.B.E., S.A.N.F. (Associate Member 7772), an Associate Member of the Institute since 1st October 1934, died in September 1963.

He served his apprenticeship with Smith's Dock Co. Ltd., after which he went to sea, serving in tankers. He was at one time fourth engineer in m.v. *British Honour.* He gained a First Class Motor Certificate, in January 1938, and a First Class Steam Endorsement in the following month.

During the Second World War he joined the South African Naval Force; in 1944 he held the rank of Eng. Lieutenant, S.A.N.F., and had been awarded an M.B.E. He was later promoted to Lieutenant-Commander (E), S.A.N.F.

Of recent years he had lived and worked in Tripoli.