MACHINERY CONTROL SYSTEMS

NAVAL EXPERIENCE IN DESIGN AND OPERATION

BY

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The following paper was presented by the Authors to the Scottish Section of the Institute of Marine Engineers in Glasgow on 22nd January, 1964, and is reproduced by permission of the Institute.

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 (FIG. 1). Operation 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 we assume that measurement devices exist and our 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 we shall 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 we will discuss. 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 we will discuss

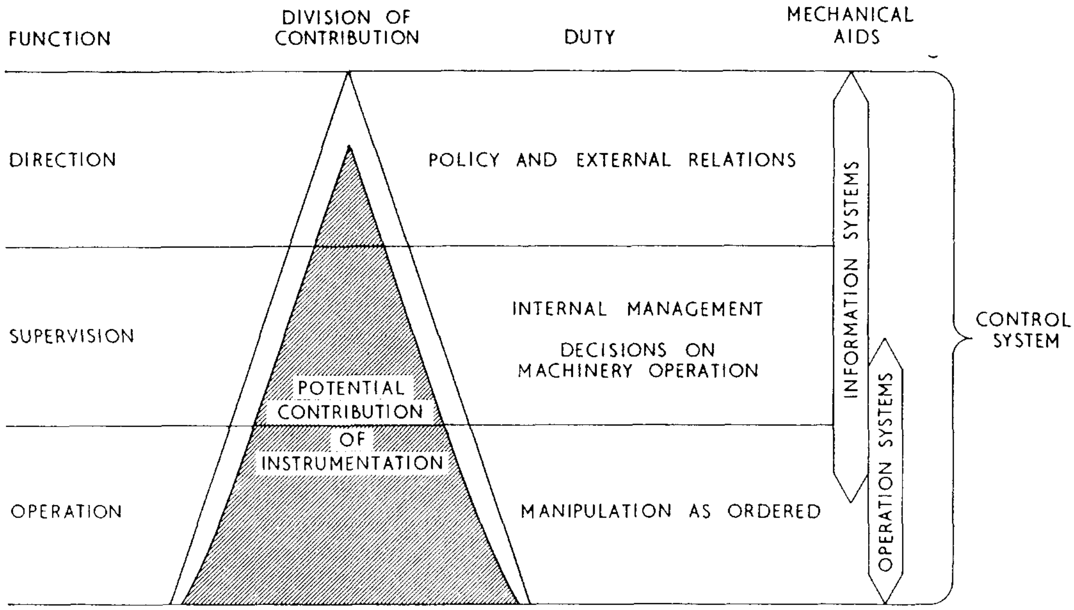


FIG. 1—LEVELS OF DECISION AND CONTROL

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 stand-by 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 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.

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 watertight 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 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. We 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, we prefer, 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 man power 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 we have 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 we have the use of several small analogue computers and two digital computers.

This, then, is the background against which our 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 F.D. blower.

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, both as to 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 (FIG. 3). In some cases, however, where it is necessary to accelerate the control action

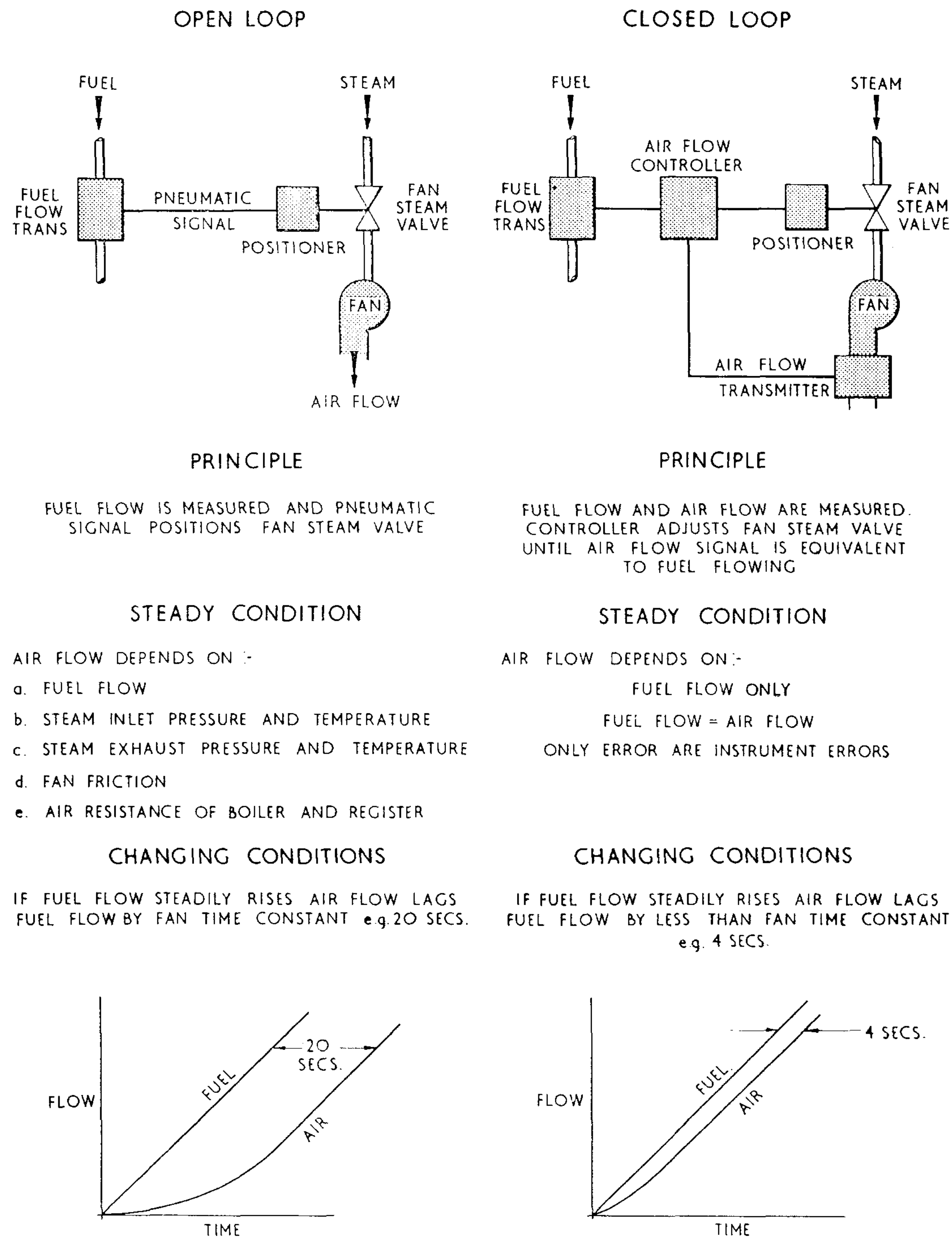


FIG. 2—AIR FLOW CONTROL—OPEN LOOP, CLOSED LOOP

sharply an additional term is used generated by differentiating the error as it develops, i.e. derivative action.

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

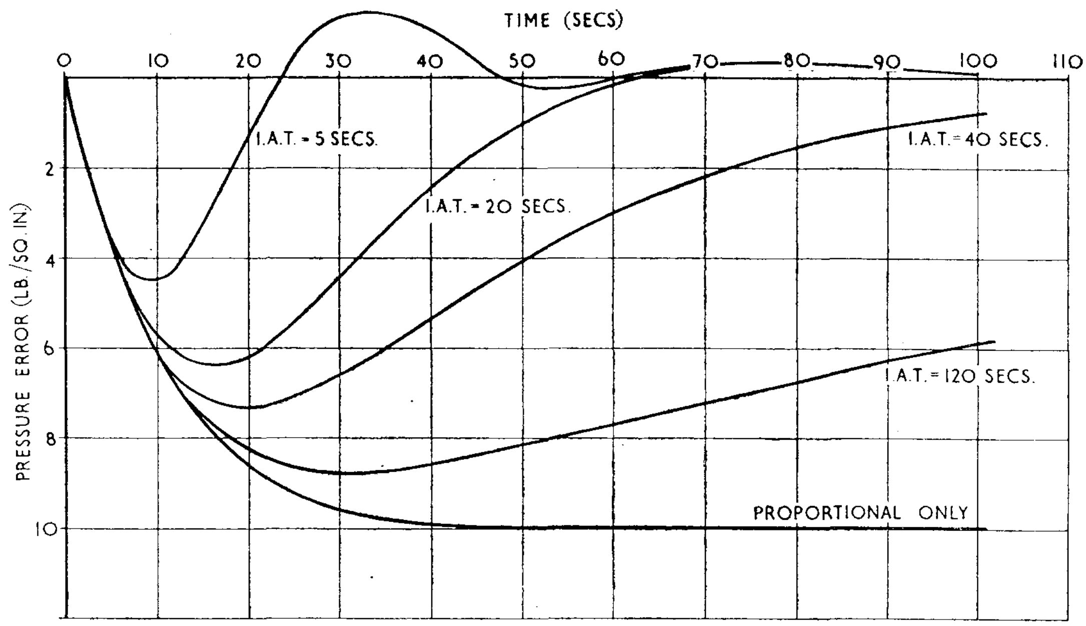


FIG. 3—EFFECT OF INTEGRAL ACTION TIME ON STEAM PRESSURE RESPONSE

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 the 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 we have had 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. While 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 we have been obliged 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.

Our first experience in boiler control system design was in 1958 with H.M.S. *Centaur* (a ship with machinery of pre-war design), which was based on power

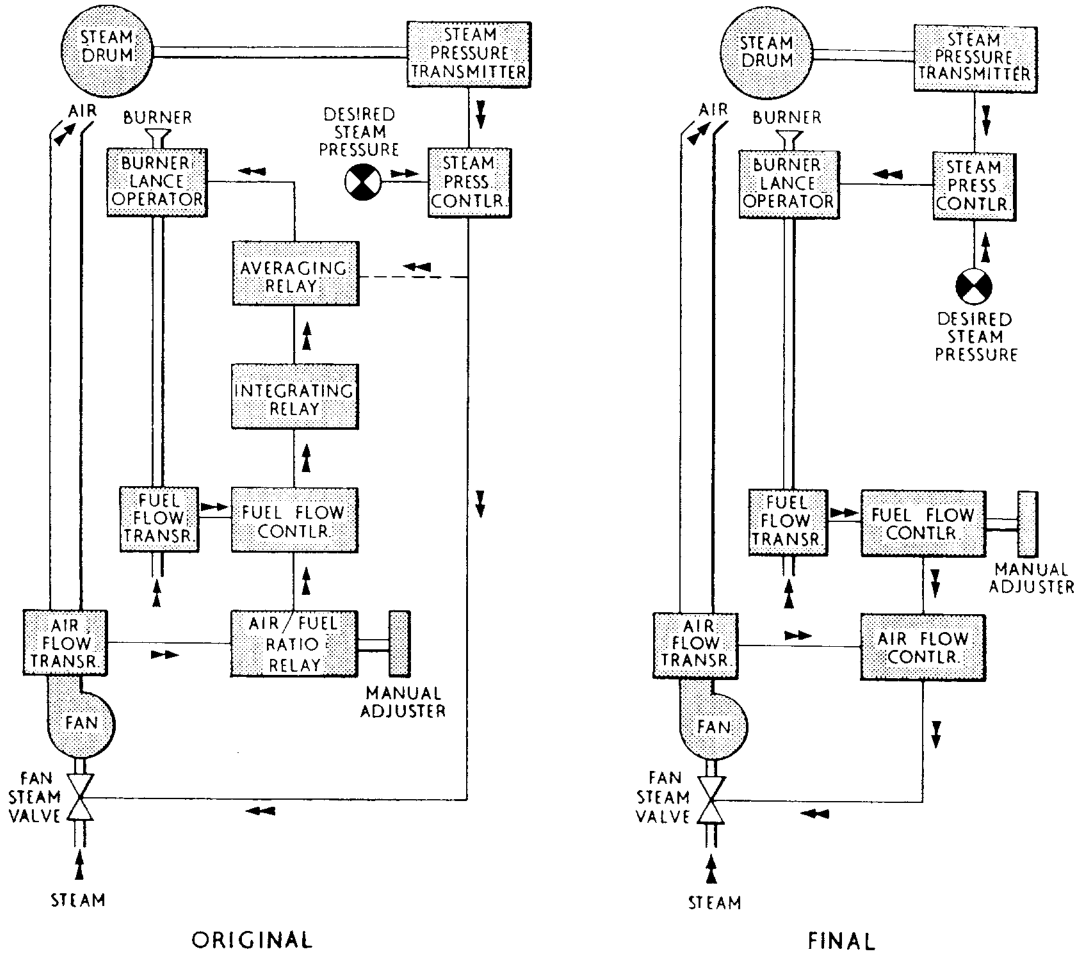


FIG. 4—H.M.S. CENTAUR BOILER CONTROL SYSTEM

station practice (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 to not only 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 15 lb/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 our rapid load change requirements and in subsequent ships we have used spill burners.

(*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 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 connection 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 we are now in the position where we can eliminate some of these larger difficulties at the design stage.

A hard lesson has been that of the correct selection of control valves. While 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 $9\frac{1}{2}$ 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 our 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 imposed 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 we have had to pay much attention (FIG. 5).

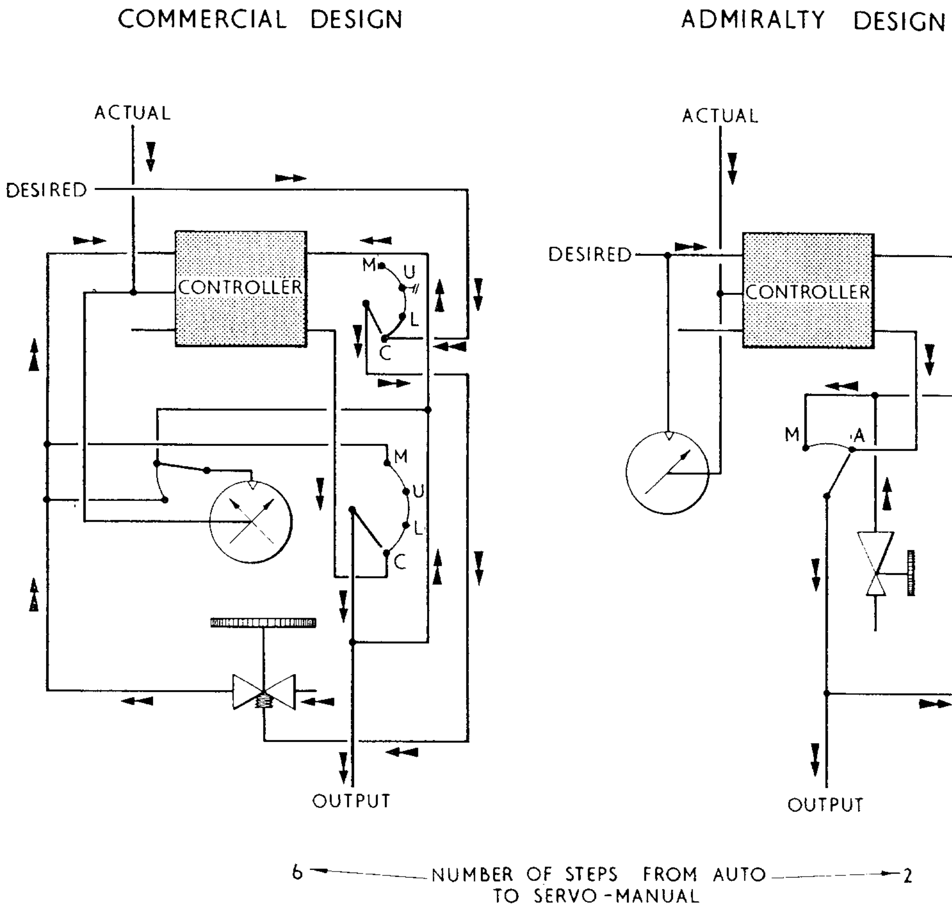


FIG. 5—F.F.O. PRESSURE CONTROL STATION

INSTRUMENTATION MEDIUM

Our 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 our particular requirements for lightness and simplicity in signal processing. Electronics are not well suited to the environment of steam machinery. Pneumatic instrumentation, on the other hand, meets all these requirements, provides us with all the performance we need and has the additional advantage of having been developed in industrial use for thirty years and more. We have, therefore, adopted pneumatic instrumentation for all our major control systems. Although there are several excellent ranges of proprietary pneumatic instruments, we have stuck 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 we have learned the hard way, and so little published guidance available that we have had to produce our own guide book on the subject.

With regard to bringing control systems into use we have found it necessary 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.

We have not yet got performance specifications 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 we have found it 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 we are planning 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. We had the case in one ship of a 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 we failed to take sufficient account of the need for clean, dry and oil-free air, and got into serious trouble with fouling of the small nozzles and restrictors built into pneumatic instruments. A film of oil on the walls of an orifice of 0·005 in. bore 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 connection, 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 get used to it in time. The remoteness of the watchkeepers from the 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

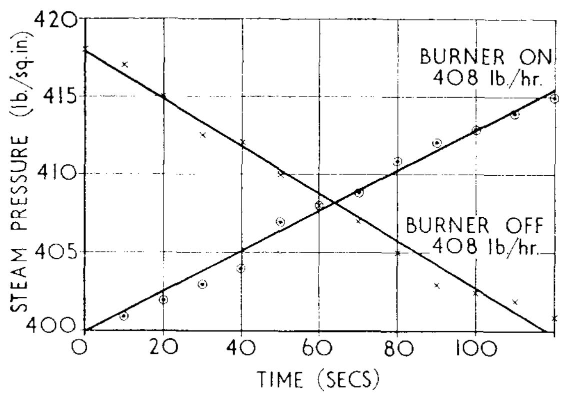


FIG. 6—STOP WATCH RECORD OF BOILER PRESSURE—BURNER—ON/OFF

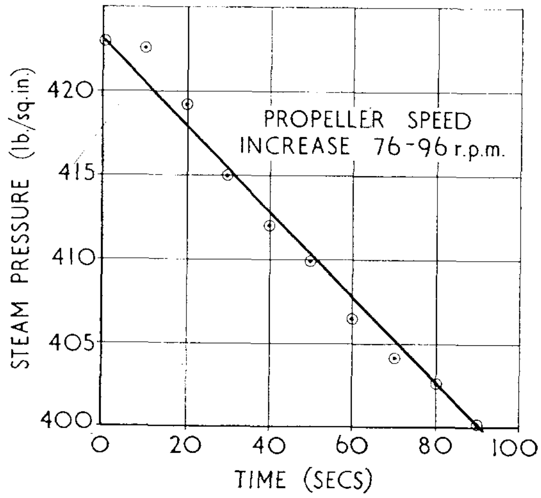


FIG. 7—STOP WATCH RECORD OF BOILER PRESSURE—SUDDEN CHANGE IN MANŒUVRING VALVE

of machinery the control instruments themselves must undergo periodical tests for accuracy and balance. We have, therefore, found it necessary to provide ships with a range of test equipment, of which the most important are those for checking the accuracy of the measuring instruments.

HISTORY OF RESEARCH

Initial trials in H.M.S. *Centaur*, 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. *CENTAUR*

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 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.

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

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 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.

$$\frac{dP}{dt} = K \quad S \qquad (2)$$

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 per second

S Steam off-take as a fraction of S_o

L Latent heat of steam at the operating condition—BTU per lb.

h Sensible heat of water at the operating condition—BTU per lb.

F_o Fuel flow to the boiler at full rated load—lb per hour

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.

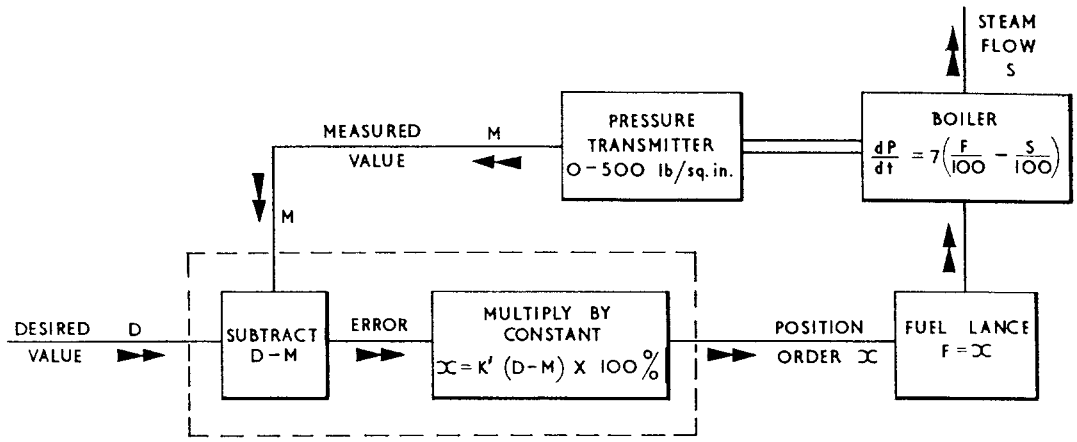
$$S = F \qquad (3)$$

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:

$$L \quad \delta S \quad S_o \qquad \text{BTU per sec}$$

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.

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



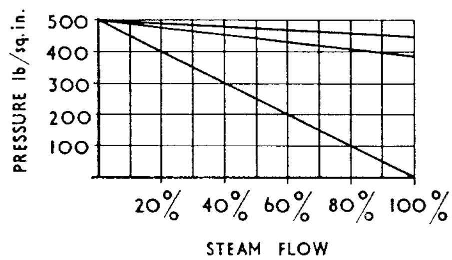


FIG. 8—DEPENDENCE DIAGRAM FOR H.M.S. CENTAUR

or a reduction in pressure of δP given by:

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

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:

$$\frac{dP}{dt} \quad = \quad \frac{-L \; S_o}{M} \quad \frac{(\partial P)}{(\partial h)} \quad \delta S \qquad (6)$$

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

$$\frac{dP}{dt} \quad = \quad \frac{L \; S_o}{M} \quad \frac{(\partial P)}{(\partial h)} \quad \delta F \qquad (7)$$

Equations 6 and 7 may be combined to give the more complete Equation 8:

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

where

$$K \quad = \quad \frac{L \; S_o}{M} \quad \frac{(\partial P)}{(\partial h)} \qquad (9)$$

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 6·3 lb/sq in./sec.

K from steam flow change 7·6 lb/sq in./sec.

K from boiler design data 7·2 lb/sq in./sec.

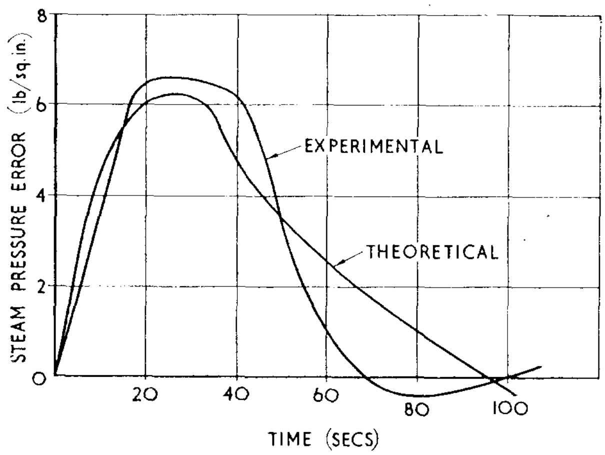


FIG. 9—COMPARISON OF THEORETICAL AND EXPERIMENTAL VALUES OF PRESSURE RESPONSE—H.M.S. CENTAUR

Steam Pressure Response in Automatic Control

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

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

and the fuel flow response by Equation 11.

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

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.

Lessons Learnt from Initial Trials in H.M.S. 'Centaur'

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 whys and wherefores 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 the A.F.E.S.

Experience upon other ships broadly bore out the accuracy of the boiler representation established in H.M.S. *Centaur*. 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 shown in FIG. 10, where the effect of fuel flow variation upon steam pressure is shown. The most significant thing about FIG. 10 is that it indicates that the 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 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.

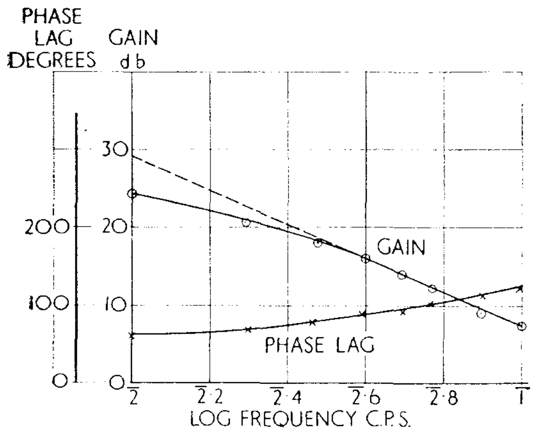


FIG. 10—STEAM PRESSURE HARMONIC RESPONSE TO FUEL FLOW

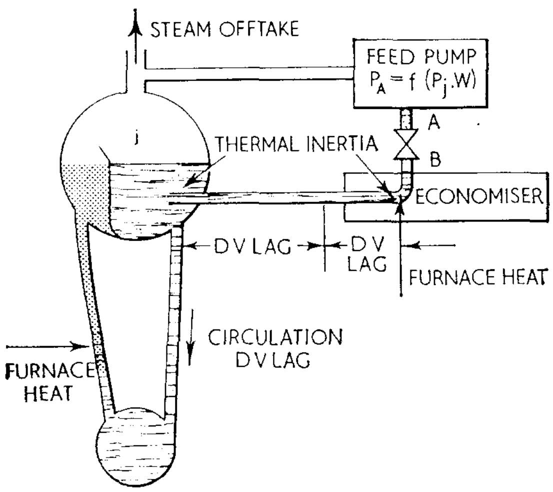


FIG. 11—PHYSICAL MODEL OF D TYPE BOILER

BLOWERS

Initial trials in H.M.S. *Centaur* 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 for the blowers. 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.

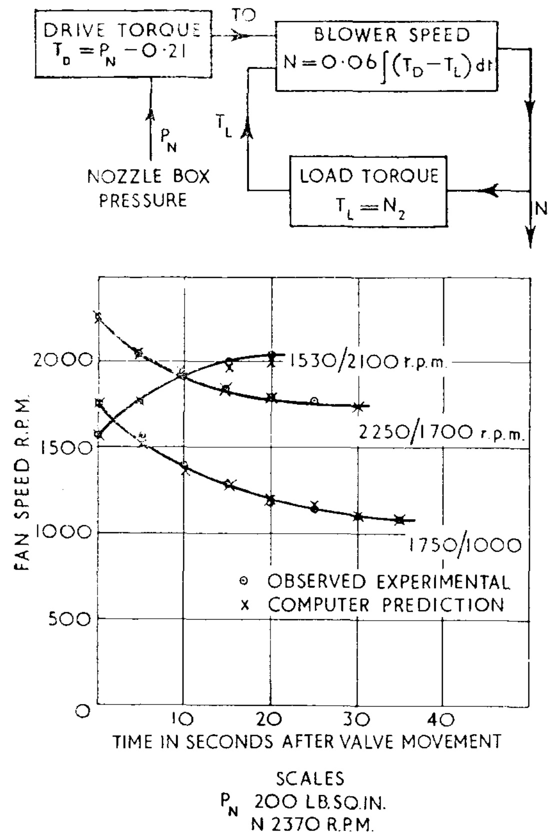


FIG. 12—STOP WATCH RECORDS OF BLOWER SPEED AND MATHEMATICAL MODEL

Calculation of Blower Time Constant

An attempt was made to calculate the blower response on a linearised basis of small changes of speed and torque. If the blower is discharging into a constant aero-dynamic 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 ft²

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:

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

The ultimate increase in speed δN will be:

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

Therefore the time constant will be:

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

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

Mathematical Model to Account for Initial Records in H.M.S. 'Centaur'

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 in H.M.S. 'Centaur'

Encouraged by results using simple analogue computer techniques, a full

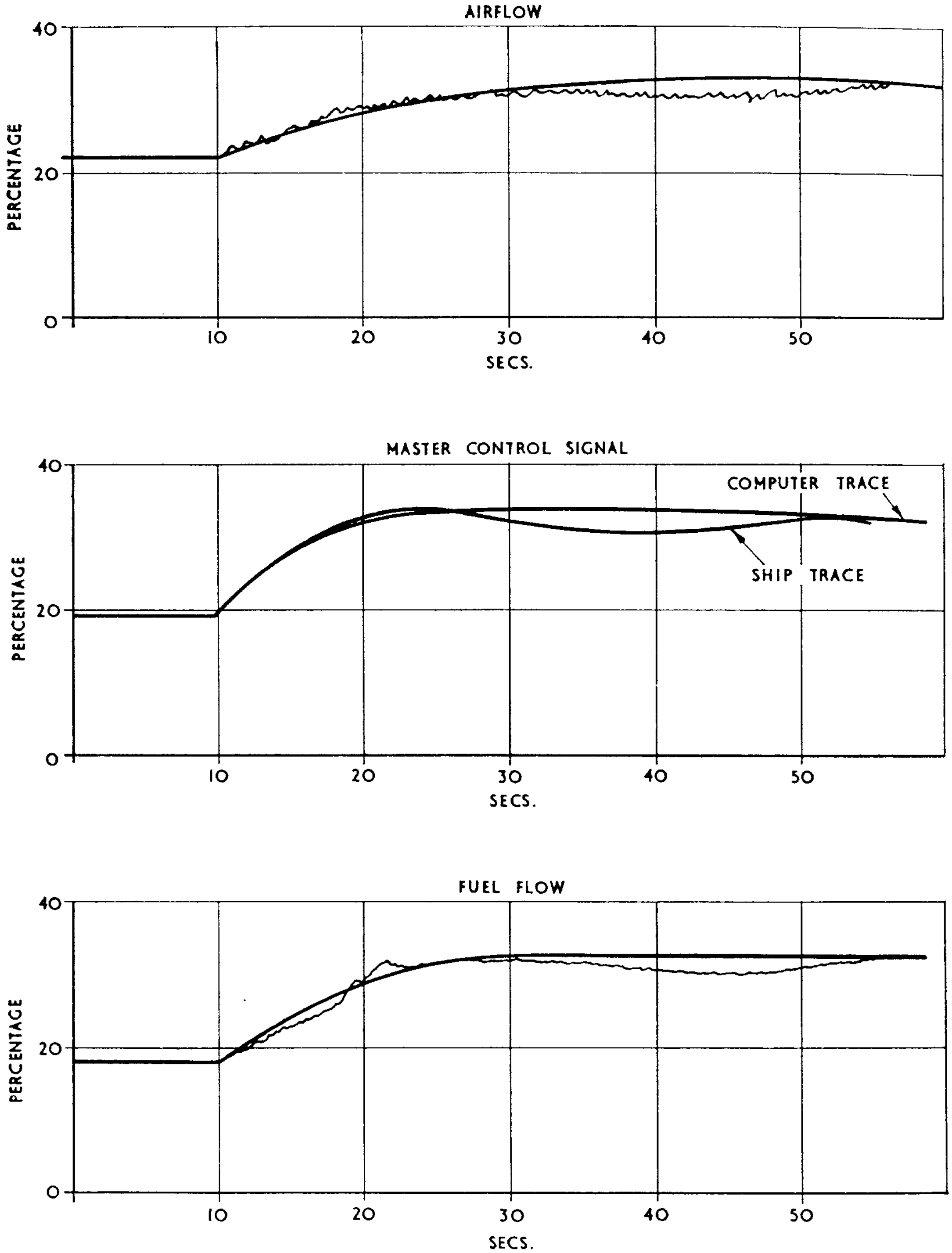


FIG. 13—SHIP AND COMPUTER PEN RECORDS, AIR FLOW—FUEL FLOW

scale effort was made to simulate in detail the air flow system in H.M.S. *Centaur* 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 were 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

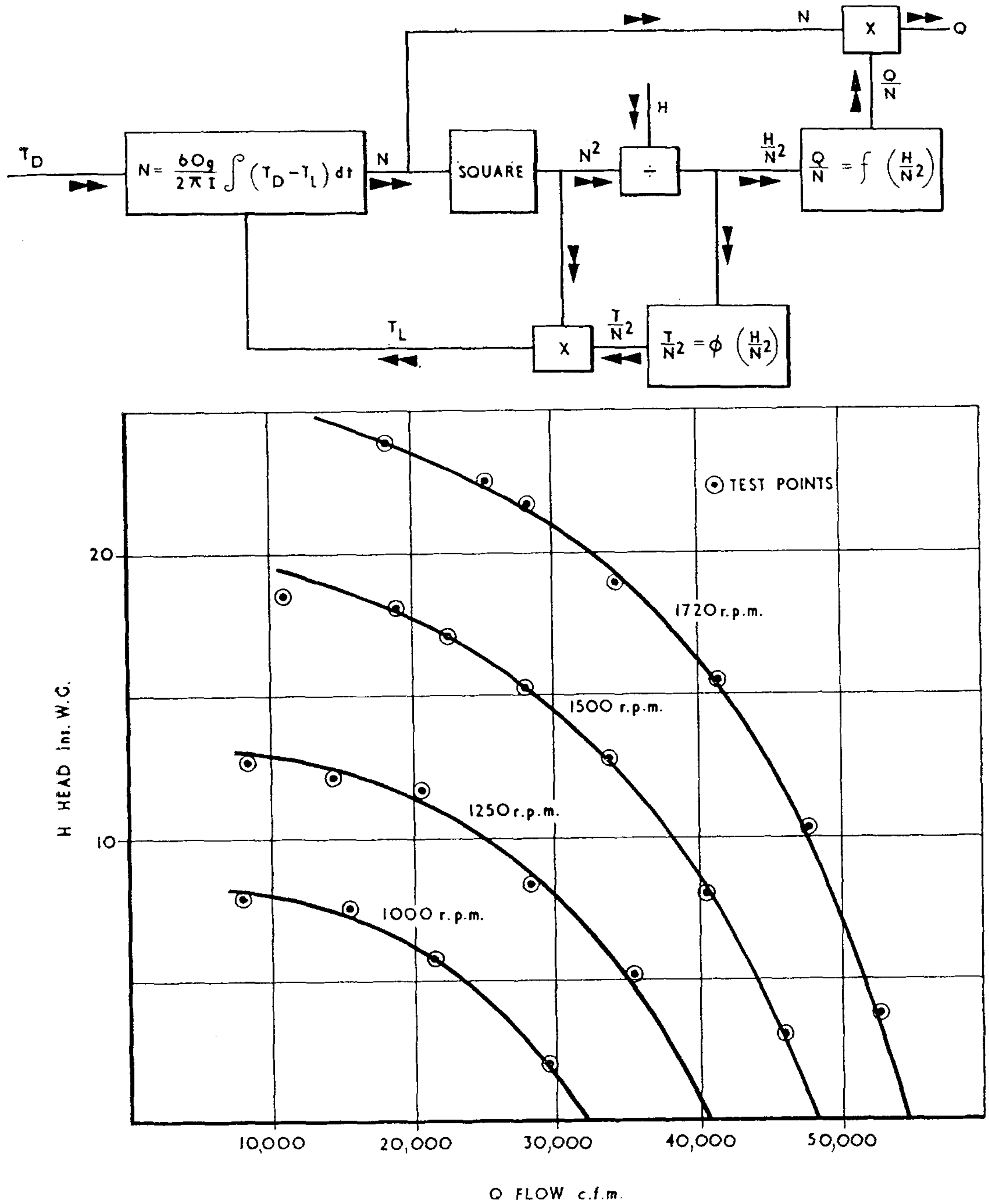


FIG. 14—GENERAL MATHEMATICAL MODEL OF BLOWER AND COMPUTED CHARACTERISTICS

ship system followed the computer prediction. Finally a step change of load was applied to one boiler on the ship and pen 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 judgment 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 the above 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 in FIG. 14.

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

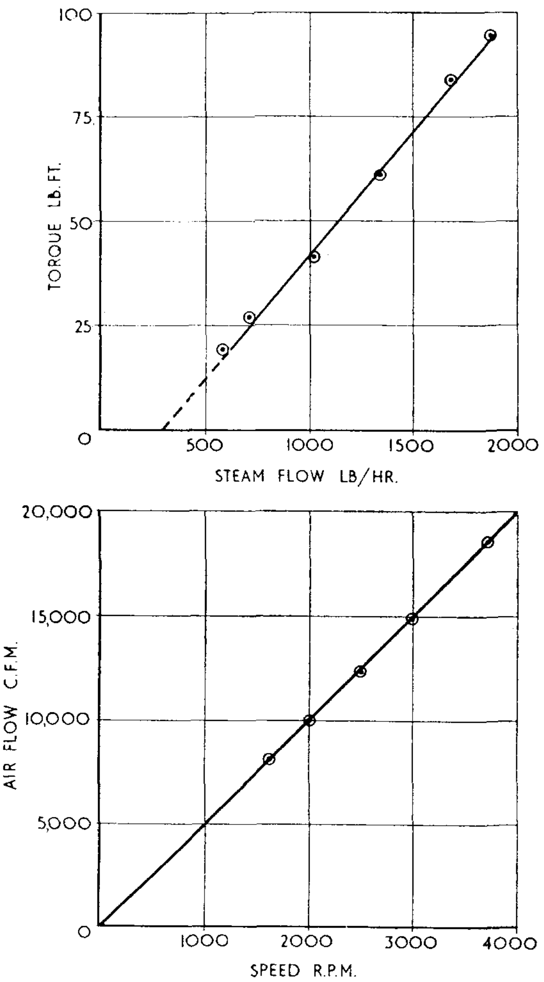


FIG. 15—SIMPLIFIED BLOWER CHARACTERISTICS

Simplified Turbo-Blower Characteristics

In many instances, steam turbo-blowers 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 turbo-blower, 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—1,000 lb/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:

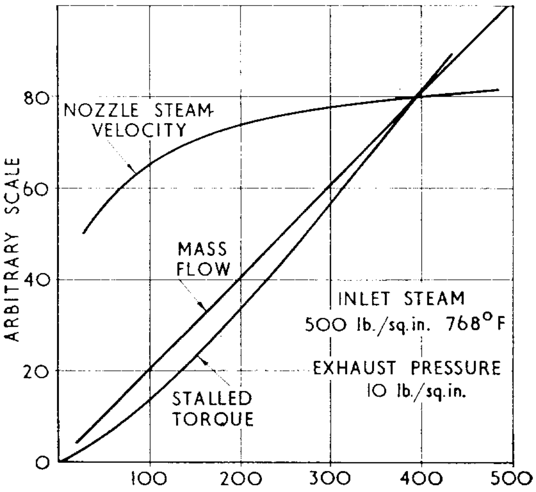


FIG. 16—CALCULATED CHARACTERISTICS FOR A THROTTLE-CONTROLLED IMPULSE TURBINE

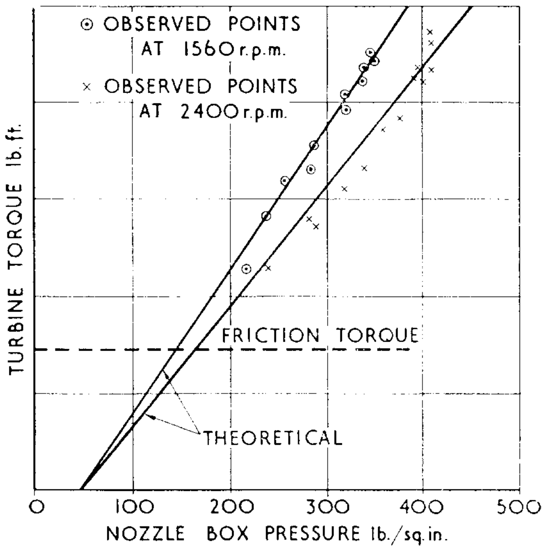


FIG. 17—EXPERIMENTAL TURBINE CHARACTERISTICS

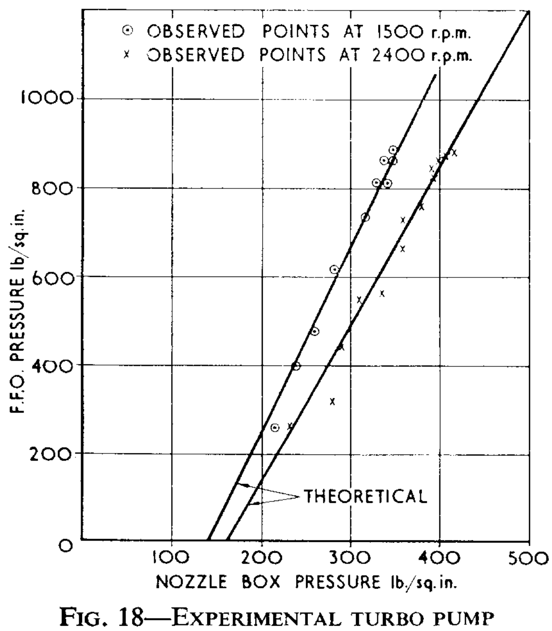


FIG. 18—EXPERIMENTAL TURBO PUMP CHARACTERISTICS

(*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

(*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 150/500 lb/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 to 40 per cent in a three-stage turbine. Over a 150/500 lb/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 plus or minus, 2·4 per cent. At a constant speed, torque should be linearly related to nozzle box pressure. The experimental results are shown in FIGS. 17 and 18.

For this particular machine it was found that the steady state characteristics could be expressed by the following set of equations:—

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

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

for the pump, and

$$T = \left(0{\cdot}365X - 16\right)\left(1 - 0{\cdot}35\frac{N}{2400}\right) \qquad (18)$$

for the turbine

Where Q is the pump output flow in gallons per hour

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 gallons per hour.

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 T is suddenly applied the initial acceleration will be:

$$\frac{dN}{dt} = \frac{\delta T\ 60g}{2\ \pi\ I} \text{ r.p.m. per second} \qquad (19)$$

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

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

The time constant will be

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

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 per hour, pressure in lb/sq in. and speed in r.p.m. when the time constant becomes, (for oil SG 0·9)

$$0{\cdot}239\ \frac{IN^2}{FP} \text{ seconds} \qquad (22)$$

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, notably in Reference 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 in Ref. 1 has been simulated. The following equations were found to represent the burner characteristics:

$$\text{For simplex operation, Burnt Flow} = 97{\cdot}7\sqrt{P_1}\ \text{lb per hour} \qquad (23)$$

$$\text{For spill operation, Burnt Flow} = 27\frac{P_2}{\sqrt{P_1 - P_2}}\ \text{lb per hour} \qquad (24)$$

$$\text{and Supply Flow} = 298\sqrt{P_1 - 0{\cdot}95\,P_2}\ \text{lb per hour} \qquad (25)$$

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

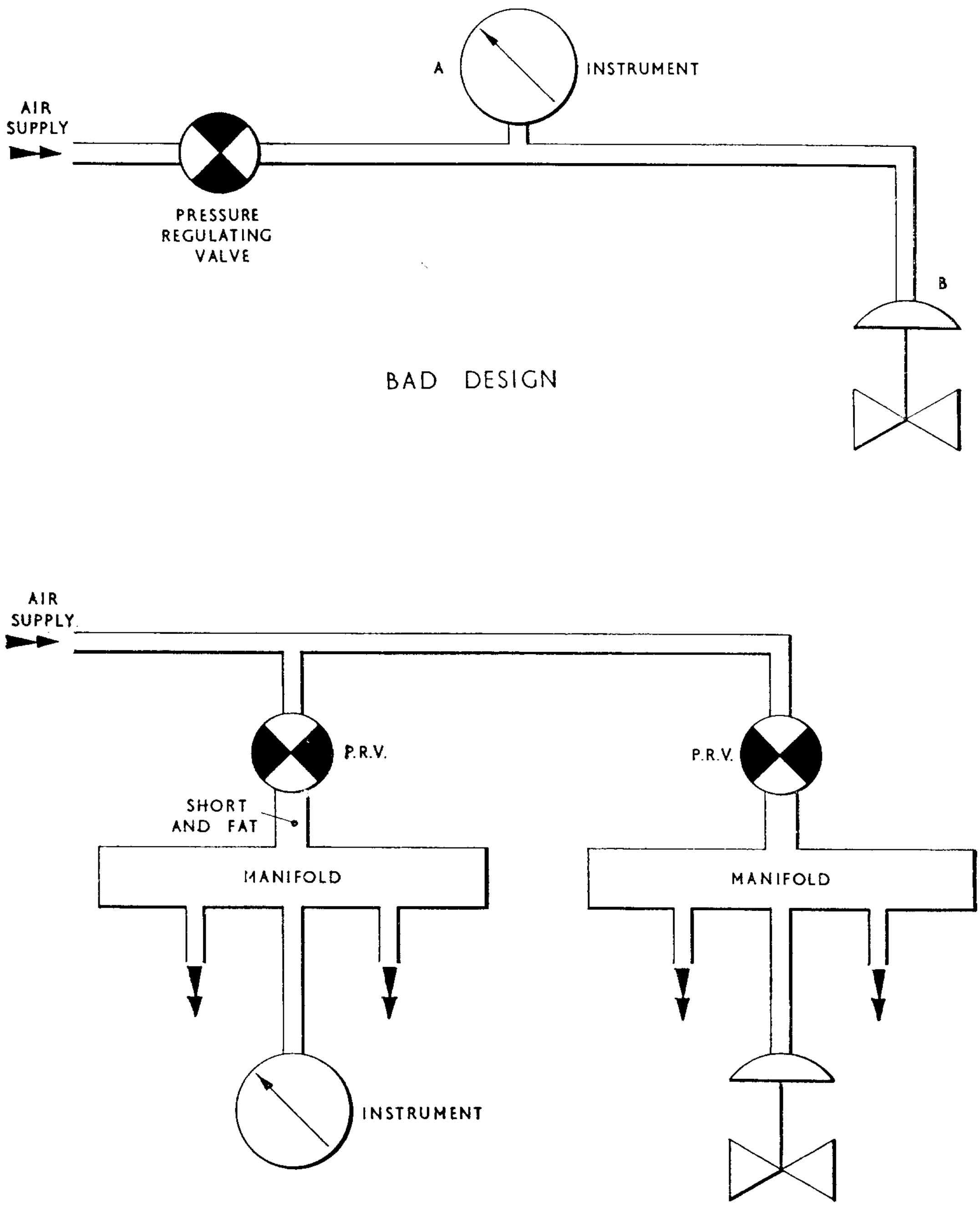


FIG. 19—SERVO AIR DISTRIBUTION

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, 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 propulsive plants such as deaerators, 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.

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 of 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 P.R.V. with a 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 P.R.V.s 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*) Don't 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 connection with sensing points are not to sense steam pressure in such a way that water can be sloshing 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

Our hopes and plans are considerable in extent.

From the all-up plant simulation we hope to find out how to specify figures for optimum plant acceleration to give maximum ship acceleration with less wear and tear on machinery. To aid this we aim 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, we hope 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.

We have looked at the fashionable subject of remote engine control from the bridge and feel that there may be something in it for us. Most such arrangements constitute the remote setting of a throttle rather than the remote control of shaft revolutions. We 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 the higher levels of machinery management we have done 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. We feel 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 is an obviously advantageous facility, and if both these facilities are provided we question the value of routine data logging. The possibilities in this field are very considerable and a firm effort is needed to keep one's feet on the ground, especially as we have not yet got such an information system at sea.

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.

Acknowledgments

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 staff of a number of H.M. ships.

The views expressed are those held by the authors and must not be construed as necessarily representing those of the Admiralty.

Reference:

1. Brown, Cdr. J. P. H. and Thomas, Lt.-Cdr. W. J. R. 'The Automatic Control of Naval Boilers'. Trans. I. Mar. E. Nov. 1960.