AUTOMATIC CONTROL BY PNEUMATIC DEVICES

BY

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Introduction

During the last two decades, while naval engineers have struggled to maintain a reasonably constant closed exhaust pressure by means of the 'Automatic Closed Exhaust to Condenser' valves hitherto fitted, a vast industry has been steadily growing to deal with just such problems in land engineering. Here, where control problems are complex to a degree so far unimagined by the marine engineer, reliable and accurate operation over long periods with a minimum of maintenance is essential, if the comparatively costly gear is to recoup the initial outlay.

The rapid spread from across the Atlantic of the industry and technique of 'Process Control', bears testimony to the way in which its methods have satisfied the most exacting customers. The writer ventures to offer these few elementary notes on its principles, and a brief guide to the way in which these principles may be put into practice.

Pneumatic controls are now being fitted in H.M. ships, primarily for the automatic operation of closed exhaust supplementing and rejection valves, so that naval engineers will soon be embroiled in yet another sideline of their calling. At least let us assimilate the first principles ; the authentic jargon, if nothing more, will then follow the more easily.

The Approach

In a short article such as this, which attempts to cover the fundamentals of a huge subject, even a discussion of the bare principles must be shorn of all refinements, the need for many of which will be apparent to the discerning reader. In particular, the actual controllers as produced bear little resemblance to the sketches on which the simple mathematical analyses are based. The 'flapper,' which is so convenient for descriptive purposes, has been almost completely superseded by improved methods of construction ; the simple bleed from the control line by means of the flapper-controlled nozzle is itself an over-simplification, although reference to more precise methods is made ; and many other interesting side issues are necessarily by-passed. But the basic principles remain unchanged, or nearly enough so for our immediate purposes.

On the practical side, no attempt has been made to compete with the manufacturers' publications which will be available to those who find such equipment in their charge. To the writer however, there has appeared to be a large gap, lying between such publications and the highly mathematical textbooks on control of all types. If the following notes do anything to fill this gap, they will have served their purpose.

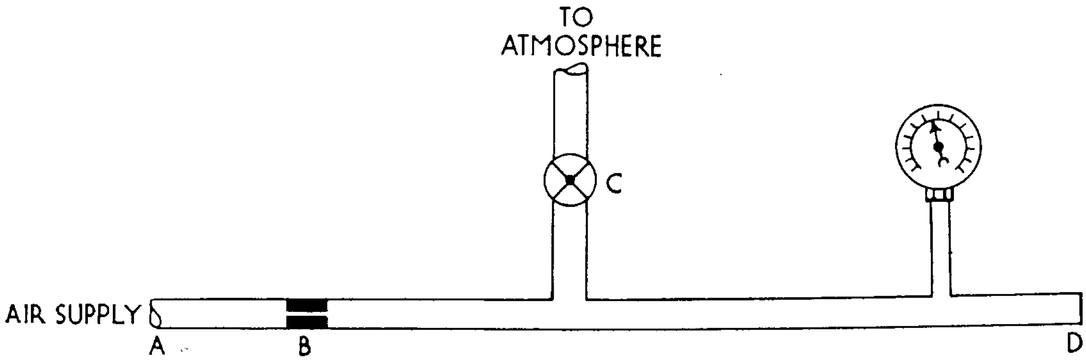

FIG. 1—THE BASIC PRINCIPLE

Why Pneumatic ?

It is no part of these notes to offer an apology for the use of air to such as may be disposed towards, for instance, hydraulics. Suffice it to say that its cheapness, cleanness, chemical inertness and freedom from fire risk enable the pneumatic designer to take full advantage of the versatility of his chosen medium.

In any case, this large industry of pneumatic Process Control has come into existence, and it would be foolish to ignore the knowledge it has to offer.

The Function

In general, and in the simplest case, the ultimate function of the control system is so to change the position of a valve that a pressure, or temperature, is maintained at a set value (which may usually be altered at will).

(*a*) The valve is called a ' Control Valve '. (Not to be confused with a Controller.)

(*b*) The pressure or temperature to be controlled is called the ' Process Variable '.

(*c*) The point at which it is wished to maintain the pressure or temperature is called the ' Desired Value '.

The function of a control system may now be restated as follows :—

' To so adjust the control valve that the deviation (or error) between the value of the process variable and the desired value is always zero '.

This deviation, without which the simple control system has no indication that remedial action is required, is commonly designated θ.

The General Principle

The control valve, and all the pneumatic gear in the control system, are operated from an air supply which need be no more than 30 lb/sq in. The air must be clean, and reasonably dry. A separating column and a filter, often of lambs' wool, are ample to ensure suitable air, without recourse to chemical drying.

If this constant supply pressure is connected to end A of the pipe shown in FIG. 1, and passes through a fixed restriction B, then, if air velocities in the pipes are low enough for frictional losses to be negligible, the pressure at the end D will depend on the degree of opening of the valve C.

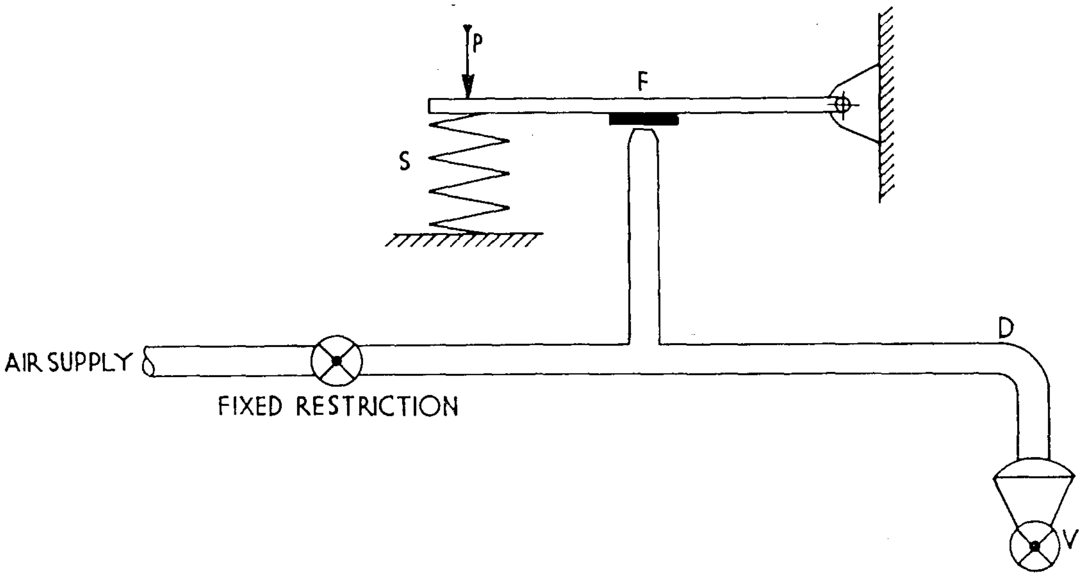

FIG. 2—THE FIRST DEVELOPMENT

Suppose now (FIG. 2) that the end D is connected to the diaphragm of an air-operated control valve V, and that the restriction C is made variable, for instance, by a flapper F over a nozzle ; the position of the control valve, in which the diaphragm load is opposed by a spring, is then controllable by changing the flapper position.

It is clear that if the flapper is hinged at one end, a variation in the force P, acting in opposition to spring S, will achieve this effect. We will thus have an elementary form of controller if we provide the force P by means of a bellows B1 connected to the process variable, the desired value being determined by the setting of the spring S.

Often it is inconvenient to set the desired value at the controller itself. In such cases, which arise particularly with centralized control of a number of processes, where the controllers may be remote from the control position, the spring S is replaced by another bellows B2, into which is fed an air pressure proportional to the desired value.

We thus come to an arrangement as shown in FIG. 3. It will be seen that a deviation θ between process variable and desired value will cause a movement of the flapper away from its equilibrium position.

Proportional Band

Before discussing the limitations of the simple form of controller developed above, it is necessary to define the term ' Proportional Band ', which will be in constant use, and which is frequently misunderstood.

The proportional band of a controller is the percentage change in the value of the process variable which will cause the control valve to operate over its full range. The importance of this concept is such that a numerical example is warranted.

Suppose that we are controlling the pressure in a vessel at 100 lb/sq in by means of a valve admitting (say) steam to the vessel, and that, for a given leak-off from the vessel, equilibrium conditions demand that the control valve should be in mid-position. Then the following table gives the responses for different proportional bands when the pressure in the vessel is altered by a change in leak-off.

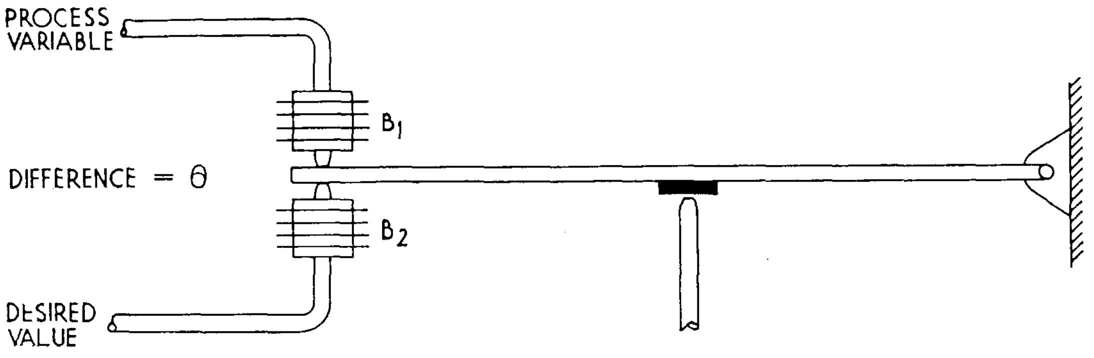

FIG. 3—THE SECOND DEVELOPMENT

Proportional Band per cent.	*Pressure in vessel when Control Valve is :—* *Full Open*	*Full Shut*
2	99	101
10	95	105
50	75	125
100	50	150
200	0	200

It will be seen that a low proportional band indicates a high ' sensitivity ', and a high proportional band a low ' sensitivity '. The term ' sensitivity ' is not however used, since an entirely different meaning is assigned to it in the terminology of Process Control.

The Two-Step Controller

This form of controller was sometimes described as ' on-off ' or ' open and shut '. These terms are now obsolete.

The action is best compared with that of a domestic hot water tank thermostat switch, which has no intermediate position between ' on ' and ' off '. Such positive action is unobtainable in pneumatic instruments, but clearly a very low proportional band approximates very closely to it.

The two-step controller is not without its uses in Process Control, but it should be evident that, in a process with any lag due to capacity or other causes (and there are few without such lag), the extremely narrow proportional band makes oscillation inevitable. Thus, for stable control, we must be able to widen the proportional band at will, to an extent dependent on the process being controlled.

Proportional Controller

Once we admit any departure from two-step action, by widening the proportional band from a virtual zero, we see that we must initiate proportional action between the deviation signal and the controller output signal which decides the control valve movement. The simple controller in FIG. 3 gives a proportional action, for if y is the deviation of the flapper from equilibrium position, at which the gap between nozzle and flapper is Y, then :—

$$y = k \,.\, \theta.$$

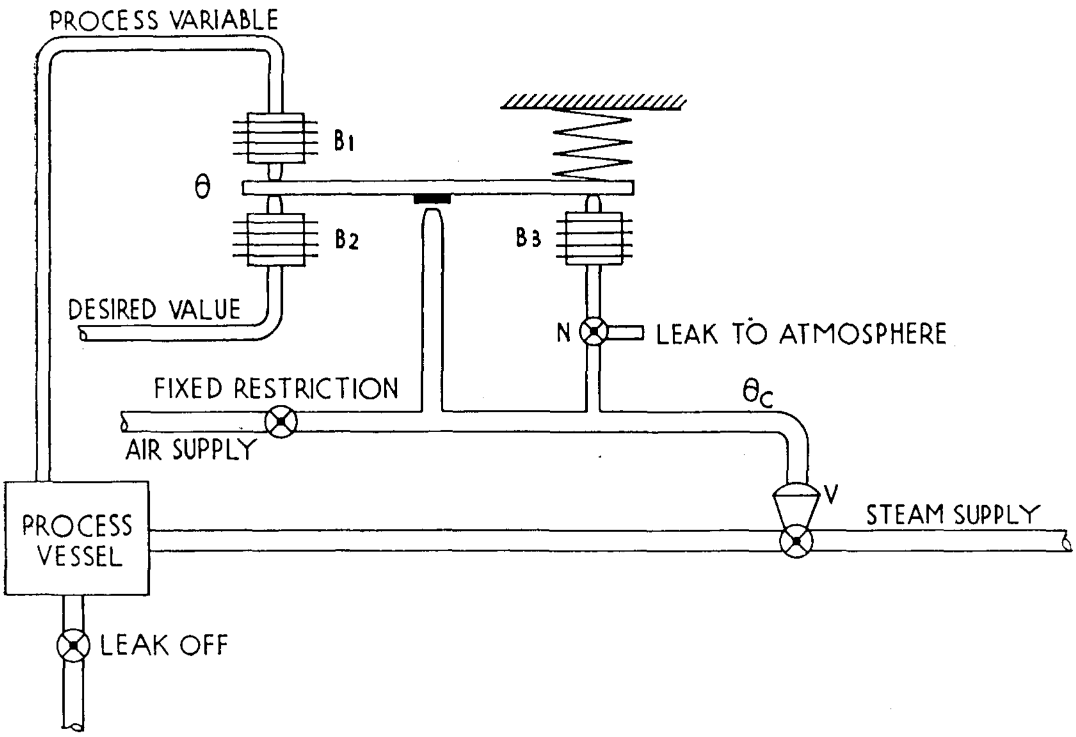

FIG. 4—THE PROPORTIONAL CONTROLLER

In practice however, because we must keep Y very small to obtain any measure of proportionality between y and nozzle pressure, only a very small value of θ is required to give a large proportional change in Y. Thus the proportional band is very narrow, and the controller is in effect a two-step controller. This disadvantage could be overcome, in theory, by altering the mechanical advantage of the flapper arrangement. This would introduce undesirable mechanical complications, particularly if, as we have seen to be desirable, the proportional band is to be easily adjustable over a wide range.

We can obtain the same effect much more easily by introducing a variable pneumatic negative feedback onto the flapper by the use of another bellows, as in FIG. 4. With this arrangement, it is seen that an increase in the process variable leading to a deviation θ causes an increase θ_C in the control signal pressure. If the full θ_C is fed back to B3, the equilibrium position is at once restored, and θ_C becomes zero. The proportional band is thus infinite, and the controller is useless. But if, in a valve N, we bleed a proportion of θ_C to atmosphere, and allow only the remainder to act in B3, we can alter the proportional band at will, by altering the proportion of θ_C fed back. The controller equation then becomes :—

$$y = k \,.\, \theta - k_N \theta_C$$

where k and k_N are proportionality coefficients.

It can be seen that if y is small in relation to the other terms of the equation (and this is arranged to be the case), a close approximation to linear proportionality is obtained, since then

$$\theta_C = \frac{k}{k_N} \cdot \theta$$

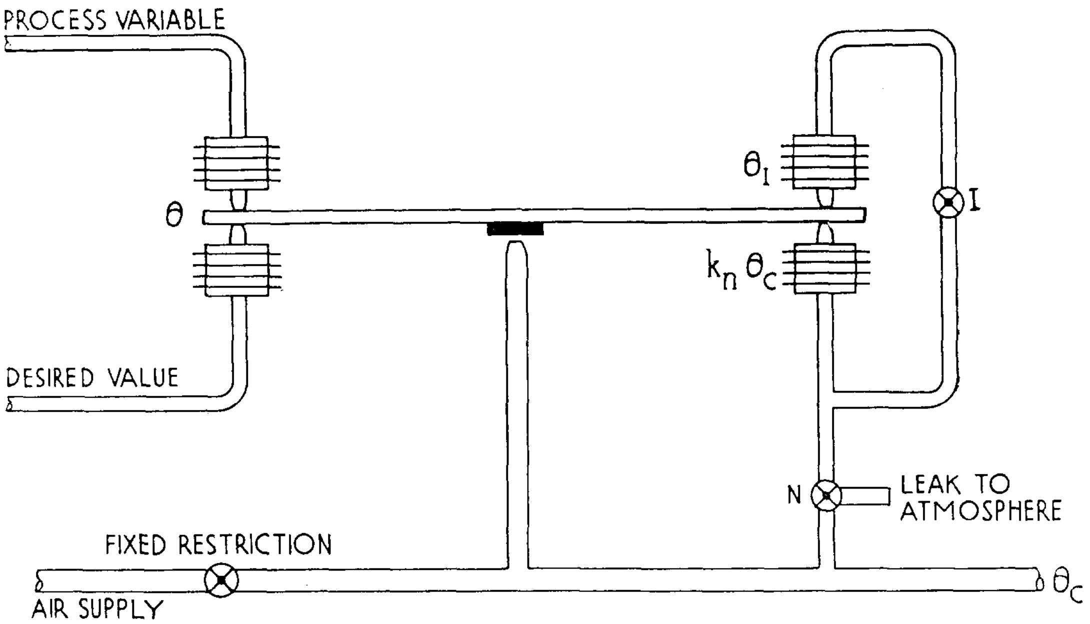

FIG. 5—THE PROPORTIONAL + INTEGRAL CONTROLLER

Limitations of the Proportional Controller

We have seen how too narrow a proportional band will lead to oscillation in a control circuit containing capacity. As the proportional band is widened, more and more stable control is obtained. A very rough rule for the optimum proportional band is that this is twice that at which the amplitude ratio of the oscillation is unity. The penalty for obtaining increased stability by widening the proportional band is the increase in ' offset '. (Often incorrectly termed ' droop ' or ' drift ').

The reason for offset is plain to see. Suppose a load change occurs, in our simple case, by an alteration in the leak-off from the pressure vessel. To maintain the same pressure in the vessel, the control valve must assume a different position, which involves a change in diaphragm pressure. This in turn necessitates a change in the gap between nozzle and flapper in the equilibrium condition ; and the only way this can be obtained, if the desired value setting remains unchanged, is by a change in the process variable, which is the vessel pressure.

If offset is unacceptable, and it often is, we must introduce a further term into the controller, known as the ' Integral Term ', and develop a ' Proportional plus Integral ' instrument.

Proportional plus Integral Controller

A possible form is shown in FIG. 5. Taking the mathematical analysis first, on similar lines to those used in the section on the Proportional Controller :—

$$y = k\,.\,\theta - k_N\theta_C + \theta_I$$

$$= k\,.\,\theta - k_N\left(\theta_C - \frac{\theta_I}{k_N}\right)$$

Assuming that y is negligible in relation to the other movements, we obtain

$$\left(\theta_C - \frac{\theta_I}{k_N}\right) = \frac{k}{k_N} \cdot \theta \text{ approx.} \qquad \ldots \quad \ldots \quad (1)$$

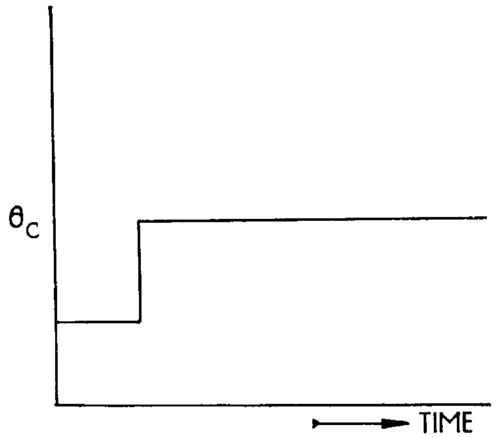

FIG. 6—PROPORTIONAL RESPONSE

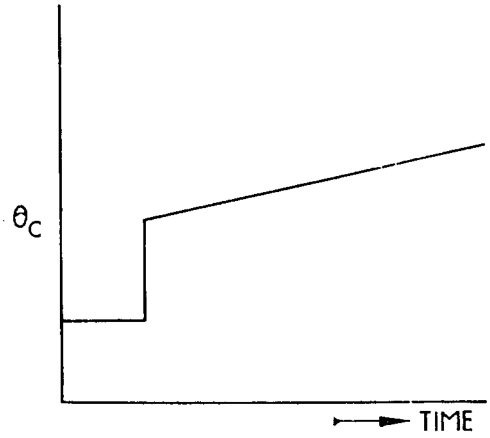

FIG. 7—PROPORTIONAL + INTEGRAL RESPONSE

If we assume that the air flow through I is proportional to the pressure drop across it :—

$$\frac{d\theta_I}{dt} = \frac{1}{\tau_I}(k_N\theta_C - \theta_I)$$

where τ_I is the time constant of the integral unit.

or

$$\frac{d\theta_I}{dt} = \frac{k_N}{\tau_I}\left(\theta_C - \frac{\theta_I}{k_N}\right)$$

Substituting in (1) :—

$$\frac{\tau_I}{k_N} \cdot \frac{d\theta_I}{dt} = \frac{k}{k_N} \cdot \theta \qquad \text{approx.}$$

or

$$\theta_I = \frac{k}{\tau_I}\int \theta \, dt \qquad \text{approx.}$$

Hence, from (1) :—

$$\theta_C = \frac{k}{k_N}\theta + \frac{k}{k_N\tau_I}\int \theta \,.\, dt \quad \text{approx.}$$

$$\therefore \theta_C = \frac{k}{k_N}\left[\theta + \frac{1}{\tau_I}\int \theta dt\right] \quad \text{approx.}$$

The effect of the integral term can best be seen graphically. If FIG. 6 shows the response of a pure proportional controller to a step change in process variable, FIG. 7 shows the response of a proportional plus integral controller to the same change.

The effect of the integral term is thus seen to be to provide, superimposed on the proportional response, a gradually increasing remedial signal which continues until such time as equilibrium is restored with the process variable exactly equal to the desired value. It can be said to ‘ reset ’ the controller on change of load, and this term is frequently used in place of the more correct ‘ integral ’.

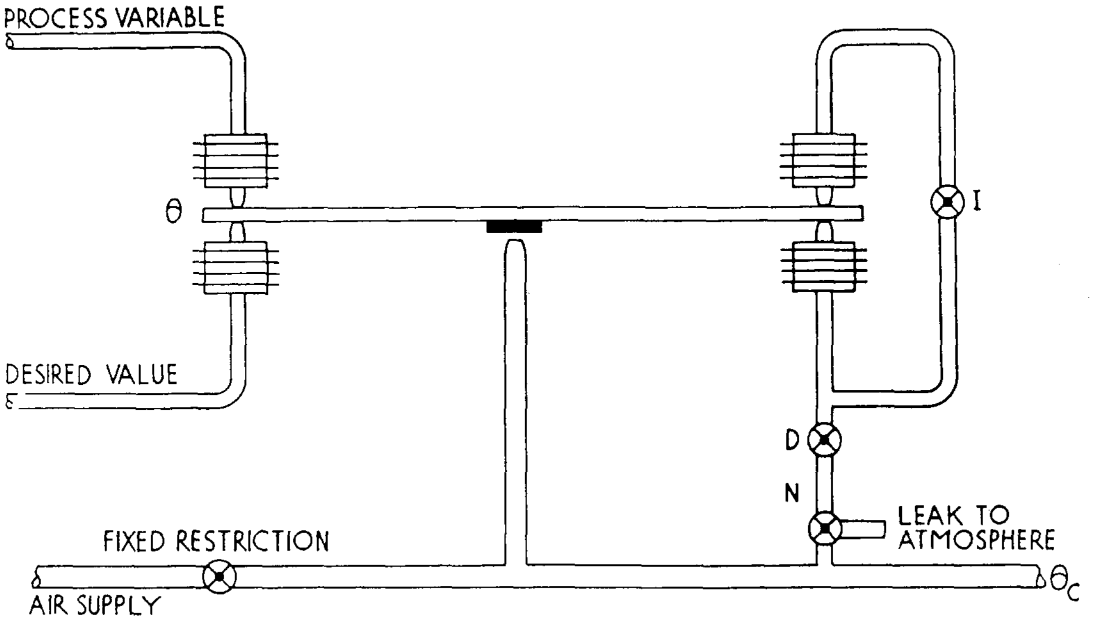

FIG. 8—PROPORTIONAL + INTEGRAL + DERIVATIVE CONTROLLER

Adjustment of the valve I clearly alters the rate of integral action by altering τ_I, and hence the gradient of the curve in FIG. 7. To standardize controller calibration, the term ' Integral Action Time ' is used, which is defined as ' the time interval, in minutes, in which the integral action increases by an amount equal to the proportional action, when the deviation is unchanging '.

Empirical rules for setting proportional plus integral controllers, which have been found to give optimum control, are :—

Proportional Band $= 2{\cdot}2 \times$ that at which the amplitude ratio of the oscillation is unity.

Integral Action Time $= 0{\cdot}8 \times$ the period of oscillation when the amplitude ratio is unity.

Such controllers are commonly designated ' two term ' instruments.

Lags in Processes and Measurement

Let us assume now that we are trying to control the oil outlet temperature from an oil heater, by controlling the supply of heating steam to it. On an increase in oil flow, if the outlet temperature is not to fall below the desired value, a corresponding increase in steam supply is required. But a controller whose action is initiated by a deviation between the measured temperature and the desired value suffers from three separate sources of delay before it receives the first intimation that it should act :—

(*a*) The delay due to the heat capacity of the heater and its contents.

(*b*) The time taken by the cooler oil to flow to the measuring point.

(*c*) The time taken for the measuring instrument to respond to the lower oil temperature.

(*b*) can be minimized by placing the measuring point as close as possible to the outflow, but, for a given design, nothing can be done to improve (*a*) or (*c*).

Methods of providing a true form of anticipation in such a case will be briefly discussed later. In the meantime however, it would clearly be advantageous if the controller could be persuaded, when it finally does receive an

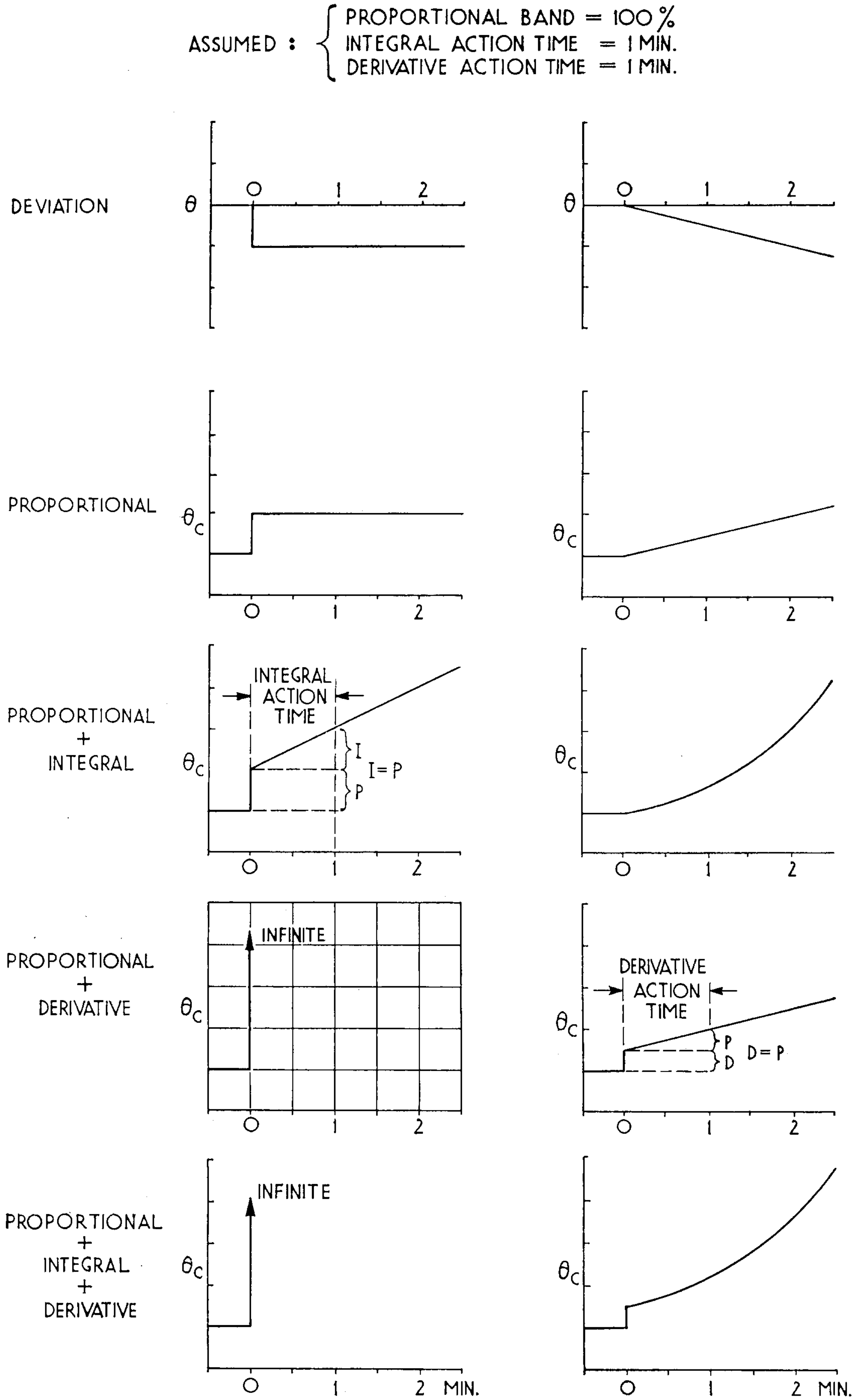

Fig. 9—Theoretical Response of Controllers

intimation that remedial action is required, to initiate such action with a ' kick ', in an endeavour to make up for lost time.

Such a result could be obtained by an initial, and temporary, large reduction in the proportional band. In fact, however, it is easier to achieve the same effect by the introduction of a derivative term. This superimposes on the inevitable proportional term (and the integral term, if employed) a signal proportional to the rate of change of θ.

Proportional plus Integral plus Derivative Controller

We find that a derivative term can be added to the proportional plus integral controller shown in FIG. 5 by the introduction of one more throttling needle valve, D, as shown in FIG. 8.

The derivation of the mathematical relationship is more tedious, though no more complex, than that given in the section on the Proportional plus Integral Controller. It involves as a preliminary, the analysis of two interdependent RC stages. It will therefore be stated without proof that the following equation results :—

$$\theta_C = \frac{k}{k_N}\left[\tau_D \frac{d\theta}{dt} + \left(1 + \frac{2\tau_D}{\tau_I}\right)\theta + \frac{1}{\tau_I}\int \theta \, . \, dt\right] \quad \text{approx.}$$

From which it will be seen that, approximately, we have obtained proportional, integral, and derivative terms in the controller output.

For instrument calibration, ' Derivative Action Time ' is used. This is defined as ' the time interval in which the proportional action, in a controller with proportional and derivative action, increases by an amount equal to the derivative action, when the deviation is changing at a constant rate.'

Similar empirical rules to those given in the section on the Proportional plus Integral Controller have been devised for the optimum settings of ' three-term ' controllers. These are :—

Proportional band $= 1{\cdot}6 \times$ that at which the amplitude ratio of the oscillation is unity.

Integral action time $= 0{\cdot}5 \times$ the period of oscillation when the amplitude ratio is unity.

Derivative action time $= \frac{1}{8} \times$ the period of oscillation when the amplitude ratio is unity.

It will be noted that the recommended proportional band has been decreased in the ratio of $1{\cdot}6 : 2{\cdot}2$ from that suggested for the proportional plus integral controller. Thus increased accuracy of response without increased instability is one of the rewards for the added complication of the derivative term.

Summary of Results

The practical effect of the derivative term is again best shown graphically, and a summary of the *theoretical* responses of the various controllers discussed is shown in FIG. 9. The infinite response to a step-function deviation when the derivative term is included is, of course, a practical impossibility. However, since, in practice, a step-function deviation is itself impossible, it is only important to note the general trend of response.

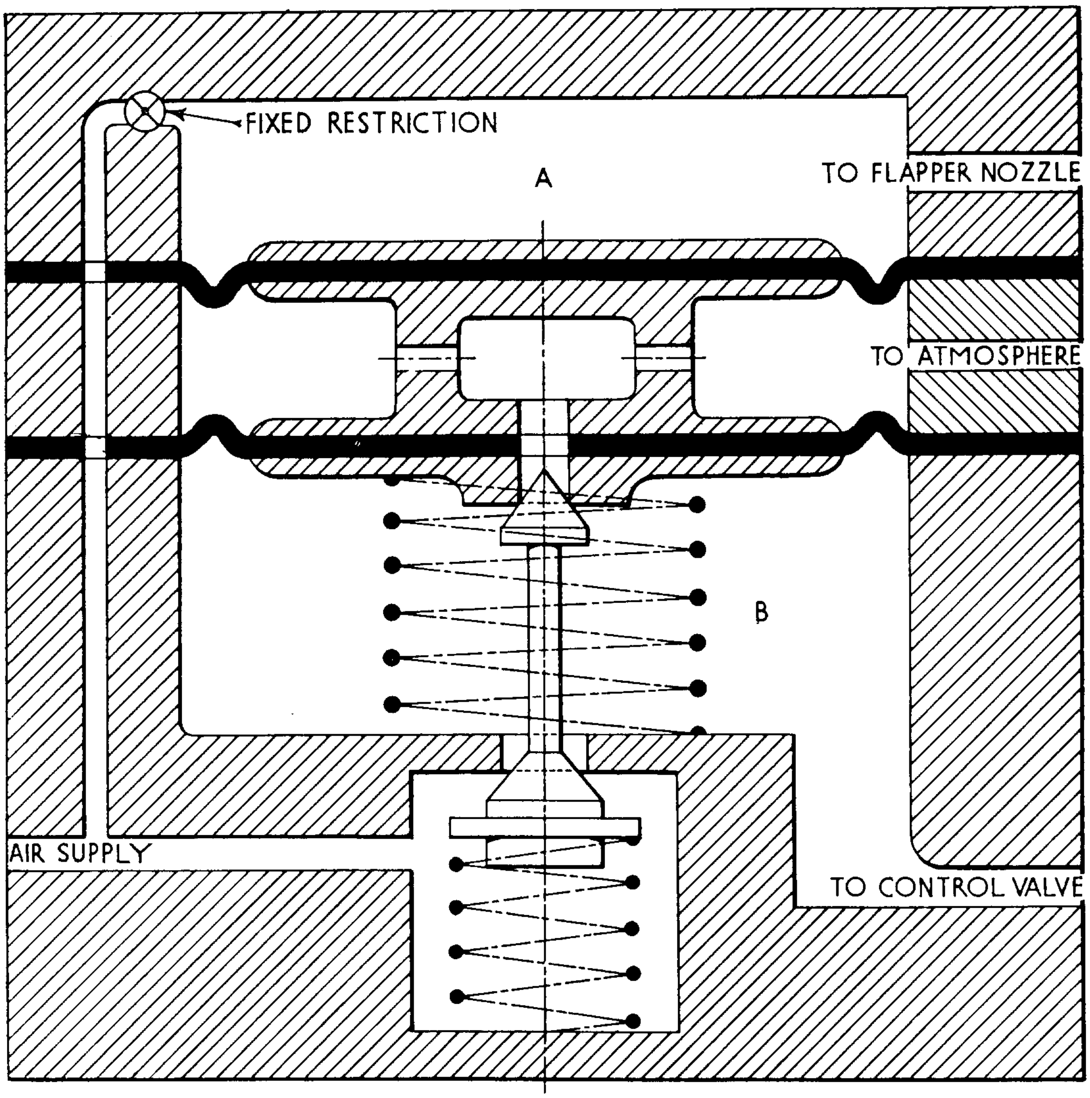

FIG. 10—BOOSTER PILOT VALVE

The Booster Pilot Valve

As was stated in the section on the Proportional Controller, the gap between the nozzle and flapper must be kept extremely small to obtain proportionality between flapper movement and nozzle pressure. In one high quality controller for instance, the equilibrium clearance is ·002 inch. And so the simple device of connecting the flapper nozzle directly into the control line becomes an absurdity if any speed of control valve response is to be obtained.

If however we use the principle shown in FIG. 1 on a very small scale, with only very small air flows involved, a small flapper disturbance from a small equilibrium clearance will provide the full range of pressure change in the nozzle line. This range can then be used to operate a valve in a direct air supply line to the control valve diaphragm.

The booster pilot valve, which, in one form or another, is incorporated in all pneumatic controllers, performs this function. A typical unit is shown in diagrammatic form in FIG. 10. It will be seen that flapper action changes the pressure in chamber A. The diaphragm unit will then move in such a way that equilibrium is only re-established when the pressure in A equals the pressure in B plus the pressure equivalent of the two light springs.

Further refinements incorporated in some booster pilot valves to improve still further the linearity of response may be mentioned without details being given of the comparatively complicated designs entailed. These are :—

(*a*) Means to ensure that the pressure drop across the nozzle-flapper orifice is always constant.

(*b*) Automatic control of the first (normally fixed) restriction, to ensure accuracy of action in spite of possible supply pressure changes.

Generation of the Process Variable Signal

In these notes it has so far been assumed that the process variable signal is led directly to the appropriate bellows. But we have seen that the desired value signal led to the opposing bellows is generated from an air supply which is never more than 30 lb/sq in. Unless therefore the bellows are so proportioned that balance is obtained in the controller, how can higher or lower pressures be dealt with ? The answer is found in the pneumatic transmitter, which generates a pneumatic signal, usually over the range 3–15 lb/sq in, which is proportional to the process variable.

It may be argued that since a range of transmitters will then be required, it would be as well to provide the range in the controller. There are three answers to this argument :—

(*a*) That controllers as manufactured do not lend themselves to such measures.

(*b*) That it is highly desirable to keep, for instance, high temperature steam away from a bellows which is actually in a controller.

(*c*) That so many forms of variables have continually to be dealt with in industry (pressure, temperature, flow, level and specific gravity to mention the most common), that a wide range of transmitters is in any case inevitable.

The Pressure Transmitter

To illustrate the principle of operation of the pneumatic transmitter, we will consider such a device designed to measure pressures up to 60 lb/sq in.

Clearly the process pressure could be connected to a Bourdon tube, the end of which could be made to operate an air pilot valve. Such transmitters are in fact in common use, particularly for high pressures. They suffer however from lack of what the American catalogues invariably term ‘ ruggedness ’, and also from mechanical friction in the unavoidable linkages.

A far more elegant form of transmitter will therefore be briefly considered, based on the pneumatic null balance principle. A semi-diagrammatic sketch is given in FIG. 11. The process pressure is led to the bellows ; this is opposed by the transmitted pressure built up under the diaphragm by the air flowing in through the fixed restriction. Clearly equilibrium will only be established when the transmitted pressure times the effective area of the diaphragm equals the process pressure times the effective area of the bellows. The inflow of air will then be equal to the leak-off through the nozzle. If in this particular case we choose a ratio of areas of 4 : 1, the maximum process pressure of 60 lb/sq in will correspond to the maximum transmitted pressure of 15 lb/sq in, and the bottom of the range of transmitted pressures, 3 lb/sq in, will be obtained when the process pressure is 12 lb/sq in.

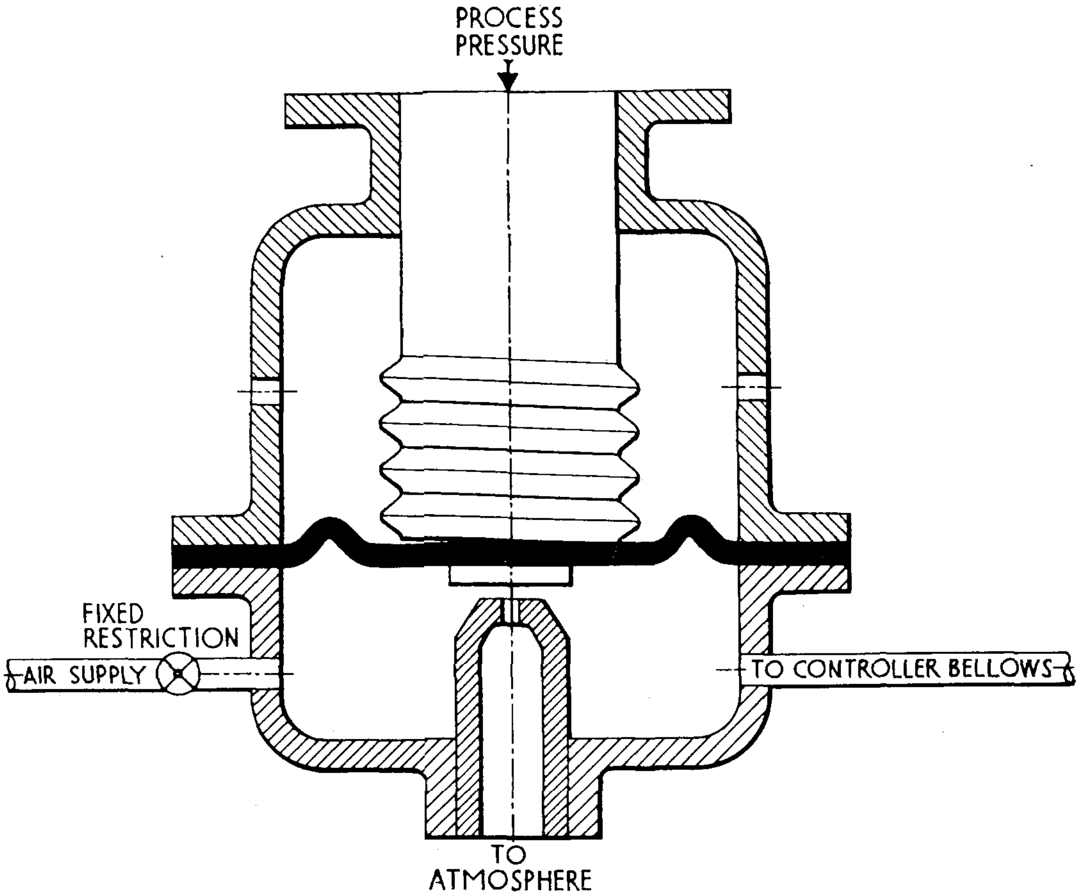

FIG. 11—ELEMENTARY NULL-BALANCE PRESSURE TRANSMITTER

In practice, for reasons connected with the need to obtain accurate proportionality of response, the design of transmitter is slightly changed, and it is used in conjunction with a booster pilot valve similar to that shown in FIG. 10. The fundamental principle however remains unchanged.

The Valve Positioner

A brief mention must be made of this device, which will be found fitted to all but the smallest control valves. We have seen in the section on the Booster Pilot Valve that the pilot valve associated with every controller is used to enable a larger volume of air to be supplied to the control valve than would be the case were the flapper-nozzle to be fitted directly into the control line. But suppose the control valve has friction, due to a tight gland or other causes. The valve movement will no longer be proportional to θ_C, as we have so far assumed to be the case.

θ_C is therefore led, not directly to the control valve diaphragm, but to a bellows in the valve positioner. Here it operates a further pilot valve, fed with its own air supply, which controls the supply of air to the valve diaphragm. Mechanical feedback from the valve stem is incorporated to restore the pilot valve to its equilibrium position when the valve movement required by θ_C has taken place.

It is perhaps worth quoting the claim of one manufacturer of valve positioners : ' diaphragm control valves of up to 1 inch stroke, and having friction as high as 15 per cent of full diaphragm power, are positioned to within $\pm$ ·001 inch.'

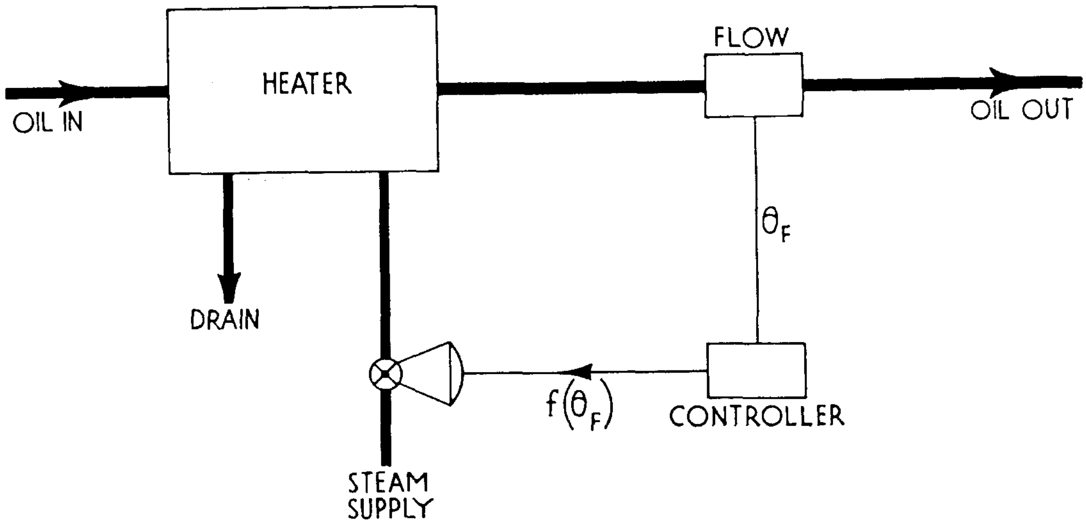

FIG. 12—OPEN-LOOP CONTROL

The Control Station

This is the name commonly given to the unit, usually fitted in a control panel, which transmits the desired value to the controller. In its simplest form, it need be no more than a piloted reducing valve with manual control of output pressure, and with an extremely level output pressure characteristic for all values of flow within its capacity. A pressure gauge is normally connected to its output, calibrated in the units, and over the working range, of the process variable, although it will be clear that the Bourdon tube operating the pointer is connected not to the process variable but to the desired value transmission line.

Developments of this simple form of control station often include the following refinements :—

(*a*) A second Bourdon tube and pointer on the same dial, to give a direct indication of the process variable. Coincidence of the two pointers then indicates satisfactory operation of the automatic control system.

(*b*) Pneumatic switches to cut out the controller, and switch the control station output pressure direct to the diaphragm (or valve positioner, if fitted) of the control valve. The process variable pointer continues to give a direct reading of the process variable pressure ; but the pointer which previously indicated the desired value now gives a direct indication of the control valve diaphragm pressure, and hence of the control valve position, and coincidence of pointers no longer follows.

The need to obtain ' bumpless ' change-over from auto to manual and vice versa usually necessitates a greater amount of control piping between the control station and the controller than would be expected from the foregoing remarks. In any particular application, however, it should not be difficult to establish the function of such connections, given full information on the precise construction of the controller used.

Anticipatory Control

It is hoped that the foregoing notes will have enabled the reader to envisage, in broad outline, the way in which a single controller and its associated gear operates to control a simple process.

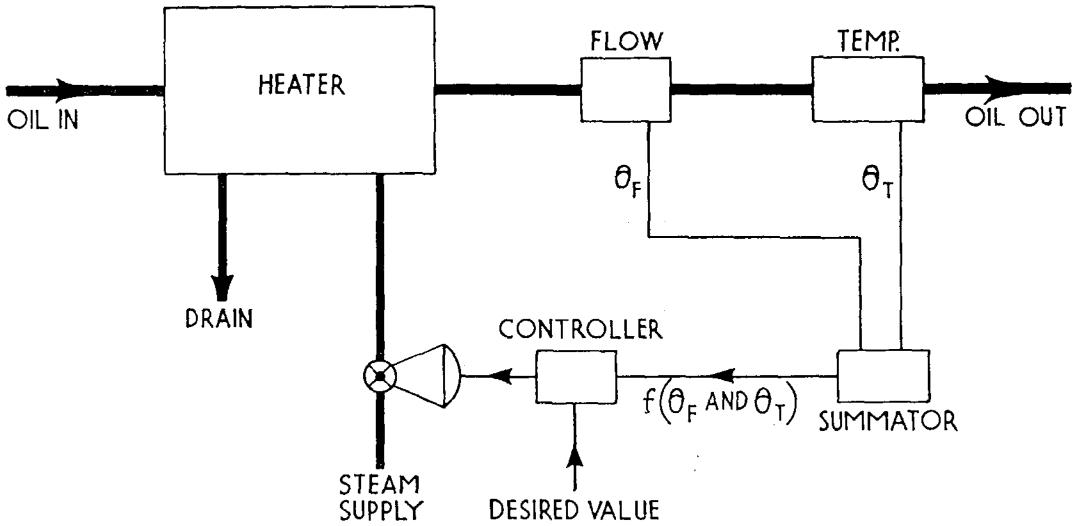

Fig. 13—Multiple Control by Summation

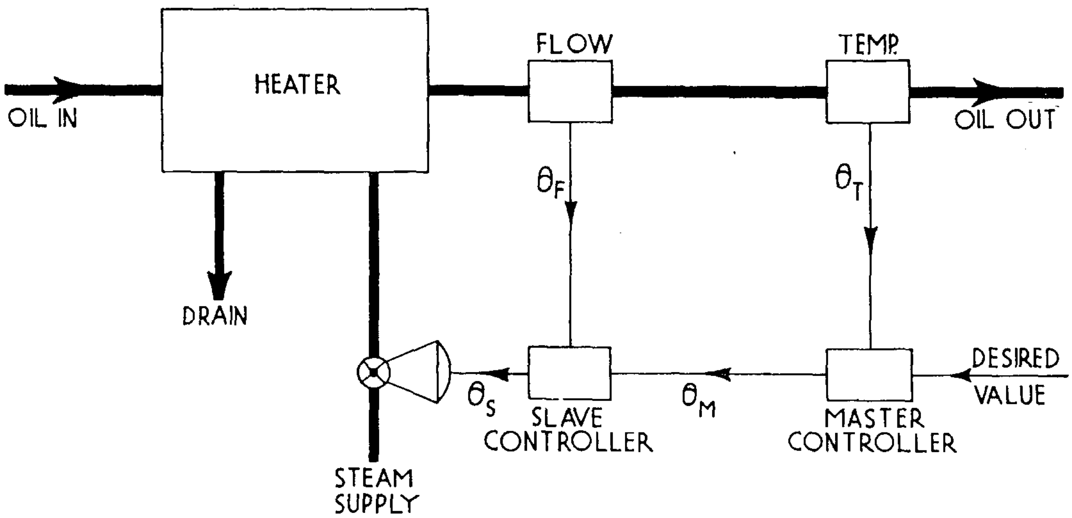

Fig. 14—Multiple Control by Cascade

Mention must now be made of true anticipation, which the derivative term dealt with in previous sections aims at, but can never attain. Consider once more the oil heating system mentioned in the section on Lags in Processes and Measurement, which is typical of many control problems in which a direct measurement of the process variable can never give really close control.

If we assume for the moment that :—

(*a*) The quality of the steam supplied is invariable, and

(*b*) That load changes do not affect heat transfer coefficients, radiation losses, etc.,

we see that, for a given oil outlet temperature, the quantity of steam to be supplied is a function of the oil flow alone.

Theoretically, we could therefore provide satisfactory control by an open loop control system, without feedback, as shown in FIG. 12. Since, however, the assumptions made above are obviously unreasonable, it is necessary to monitor a system as shown in FIG. 12 by a measurement of the process variable, and so obtain once more a closed loop system. So we introduce a temperature measurement, and obtain both a signal of flow θ_F, and one of temperature, θ_T.

There are two ways, closely related, in which θ_F and θ_T can be combined, and both are in common use in industrial control systems. They will be considered in turn, but without mathematical analysis, since in such cases both the amplitude and the phase relationships of θ_F and θ_T must be taken into account, involving the use of controller vector diagrams.

Multiple Control by Summation

To avoid further side issues, it must be taken as read that means are readily available by which signals can be added, averaged, or a weighted mean obtained. The system therefore becomes as shown in FIG. 13.

Multiple Control by Cascade

In this system, the deviation between desired value and process variable is detected in the 'Master Controller'. The output from this controller, $\theta_M = f(\theta_T)$, is fed, as a 'desired value', into the slave controller, where it is compared with θ_F. The slave controller output, θ_S, which is a function of θ_F and θ_M, is then used to operate the control valve. The arrangement is shown in FIG. 14.

Little valid comparison is possible between summation and cascade systems without carrying out a full analysis for a particular application. From a practical point of view however, the latter offers the maximum scope for design in the drawing office and adjustment in the field, and is therefore the purist's instinctive approach to problems in which multiple control is necessary.