automatically, as each movement of the actuator unit is commenced and completed.

Thus initial movement of the handle in either direction starts the pump motor, the direction of movement determines which of the two outlets will be pressurized and the degree of handle movement governs the rate of fluid flow to the actuator. When released the handle returns to the mid-position and switches off the pump motor.

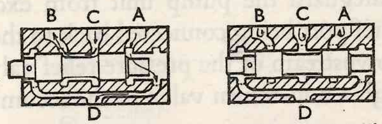


FIG. 8.19. Hydraulic four-way control valve.

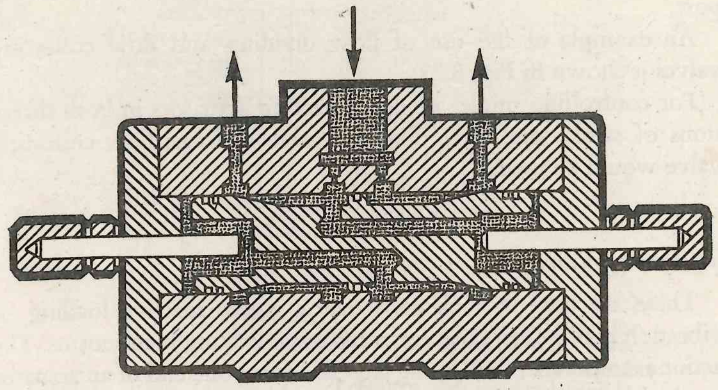
(c) *Flow Dividing and Flow Collecting Valves*

A flow dividing valve will accurately divide flows to within, say 2 per cent of the full rated flow for the unit. The flow collecting valve will collect two flows. They are used where two units must work in unison, e.g. two damper units, the two halves of a hatch cover or two changeover valves in a pipeline.

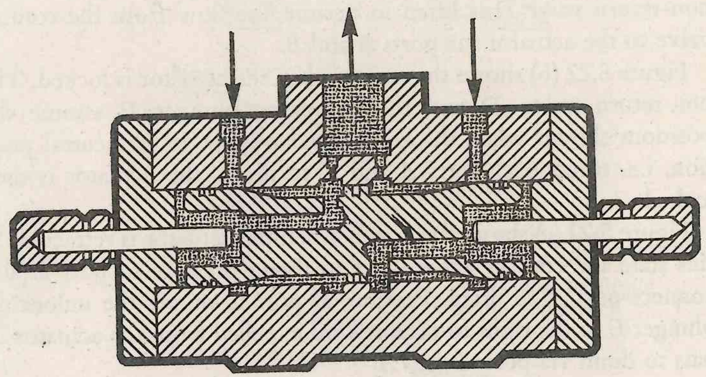
The principle of operation is shown in Fig. 8.20.

Figure 8.20 (a) shows a flow dividing valve. Fluid entering a central inlet port passes through a pair of fixed orifices. The natural pressure drops between the two ensuing flows are compared and utilized to move a spool equalizing the flows of the two supply ports. In this way accuracy of division is not affected by differing or varying loads on the units which the flows are actuating.

The principle of the flow collecting valve, Fig. 8.20 (b), is similar to that of the flow dividing valve—the fluid entering through the



(a) Flow Dividing Valve



(b) Flow Collecting Valve

FIG. 8.20. Hydraulic flow dividing and flow collecting valves.

two inlet ports passes through two orifices to the central outlet port.

An example of the use of flow dividing and flow collecting valves is shown in Fig. 8.21.

For controlling single- or double-acting actuators in both directions of stroke one flow dividing valve and one flow collecting valve would be used.

(d) *Locking Valves*

These are used to lock an actuator where reactive loading or vibration may be too great for mechanical locking mechanisms. The action is to prevent fluid being expelled from one end of an actuator until operating fluid pressure is applied to the other end. Thus positive locking is provided, at any position of the stroke. When movement of the piston is desired the hydraulic lock is freed as soon as fluid is applied by the operator.

The principle of operation is shown in Fig. 8.22 (a), (b) and (c).

Figure 8.22 (a) shows the action of the locking valve during the extension of a single-acting linear type actuator. In this position the non-return valve *D* is lifted to permit free flow from the control valve to the actuator via ports *A* and *B*.

Figure 8.22 (b) shows the action when the actuator is locked. The non-return valve *D* and the unlocking plunger *E* assume the positions shown when the control valve is set in its neutral position, i.e. to connect ports *A* and *C* to drain, the actuator is then locked.

Figure 8.22 (c) shows the action when the actuator is retracted. In this state the control valve is set to direct pressure to port *C* and connect port *A* to drain. Pressure at port *C* moves the unlocking plunger *E* to the right to permit fluid expelled from the actuator to pass to drain via ports *B* and *A*.

Figure 8.23 shows the locking valve piped in circuit with a single-acting actuator. Retraction of the actuator piston cannot take place until fluid pressure is applied to port *C*.

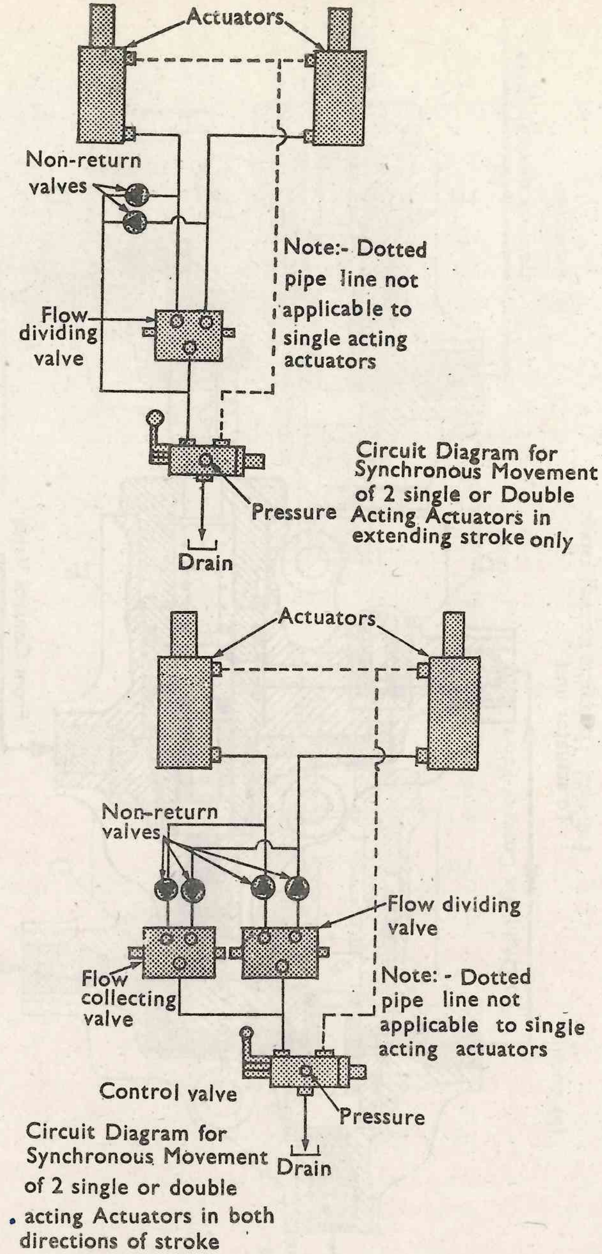


FIG. 8.21. Application of flow dividing and flow collecting valves.

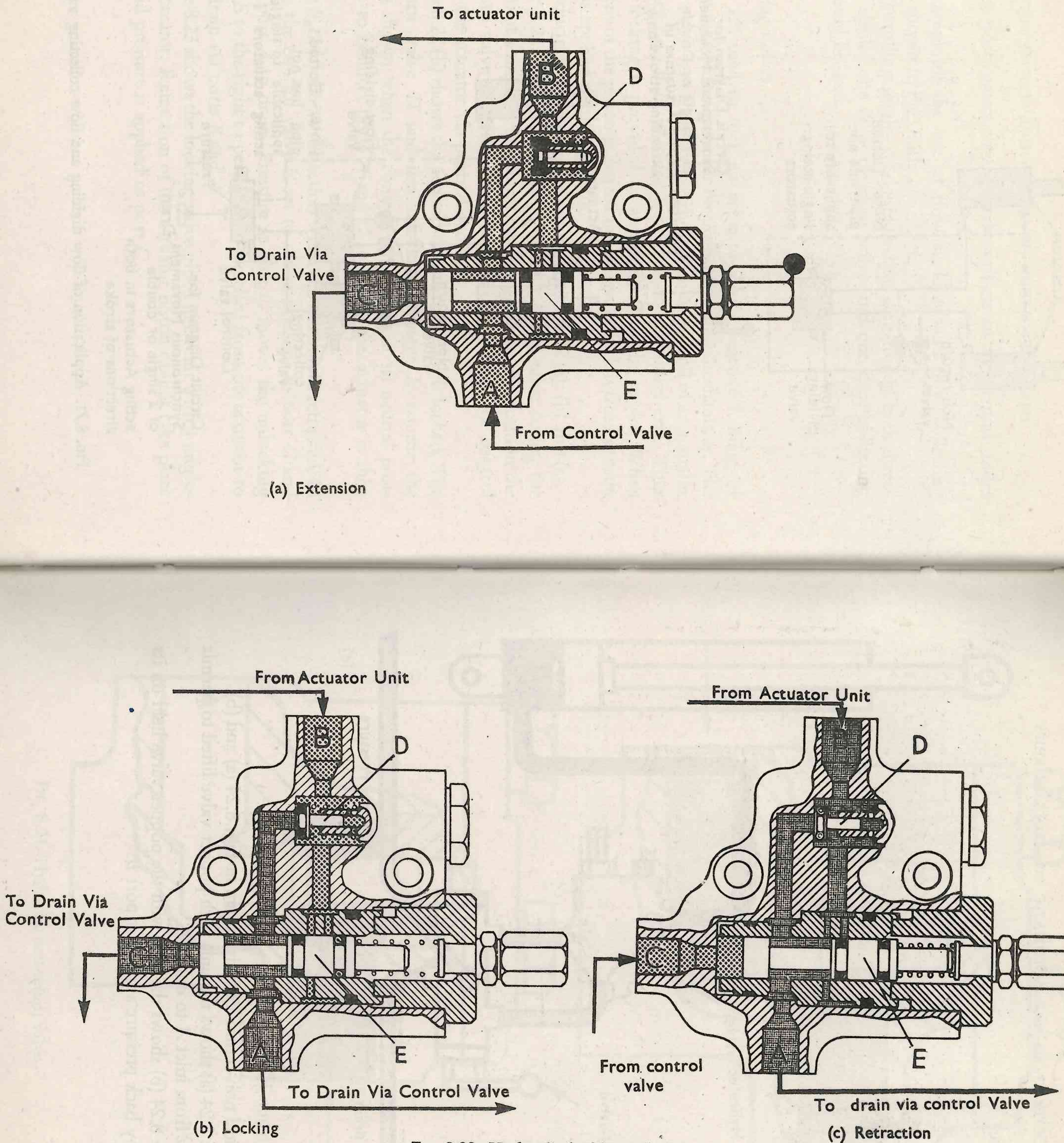


FIG. 8.22. Hydraulic locking valve.

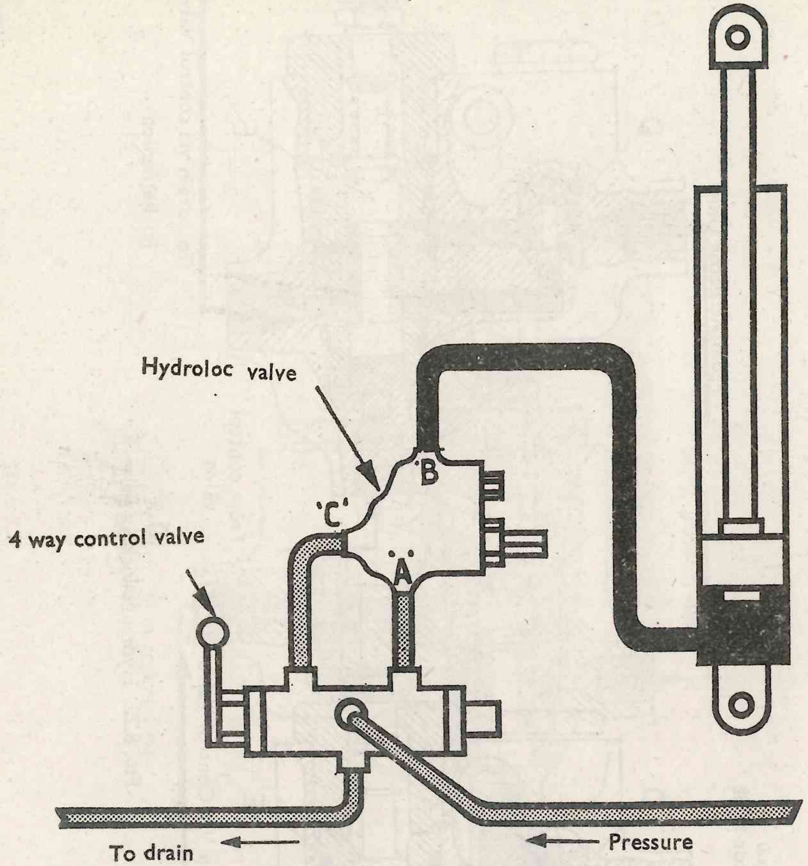


FIG. 8.23. Application of locking valve.

(e) *Non-return Valves*

A typical non-return valve is shown in Fig. 8.24 (a) and (b).

Figure 8.24 (a) shows the unit with poppet valve lifted to permit free flow from inlet *A* to outlet *B*.

Figure 8.24 (b) shows the unit with the poppet valve held on its seating by back pressure applied at port *B*.

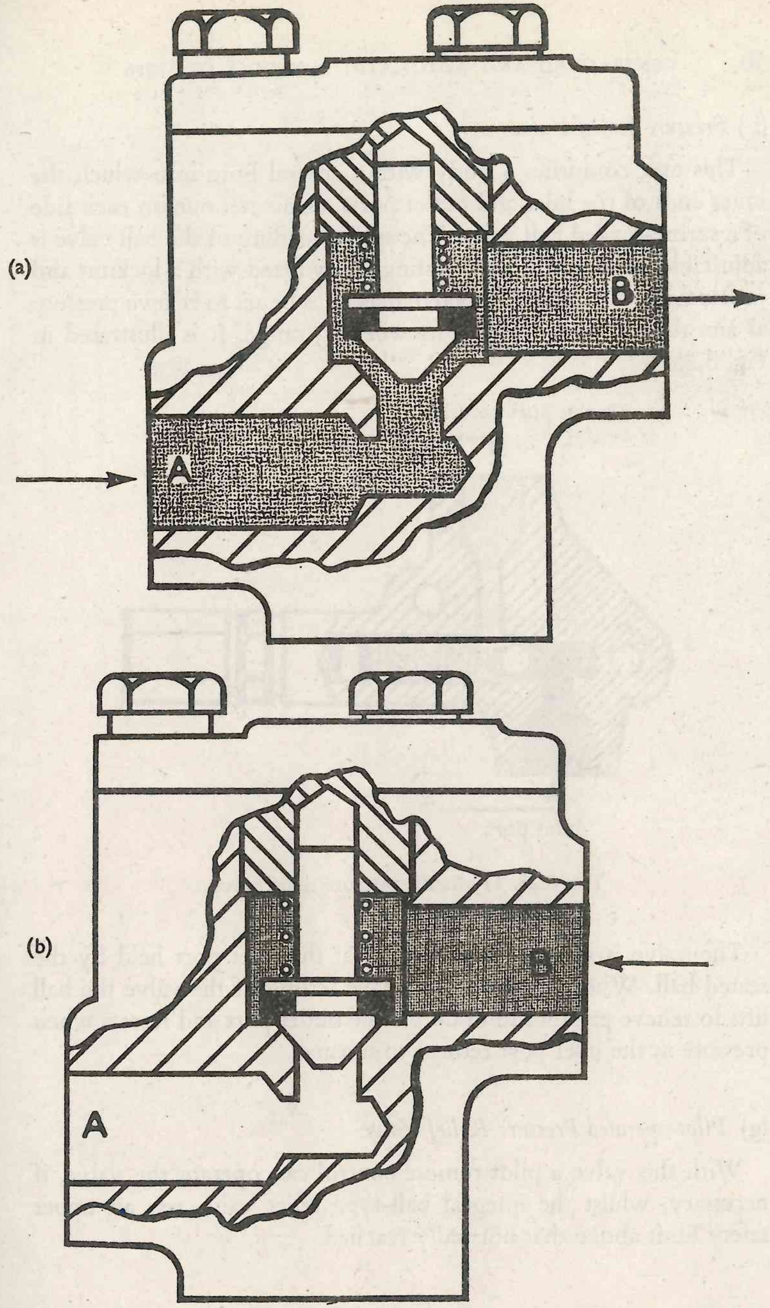


FIG. 8.24. Hydraulic non-return valve.

(f) *Pressure Relief Valves*

This unit comprises a body with a central bore into which the inner ends of the inlet and outlet ports terminate, one on each side of a spring-loaded ball valve. The spring loading of the ball valve is adjustable by means of an adjusting screw fitted with a locknut and protected by a capnut. The valve may thus be set to relieve pressure at any desired figure within its working range. It is illustrated in Fig. 8.25.

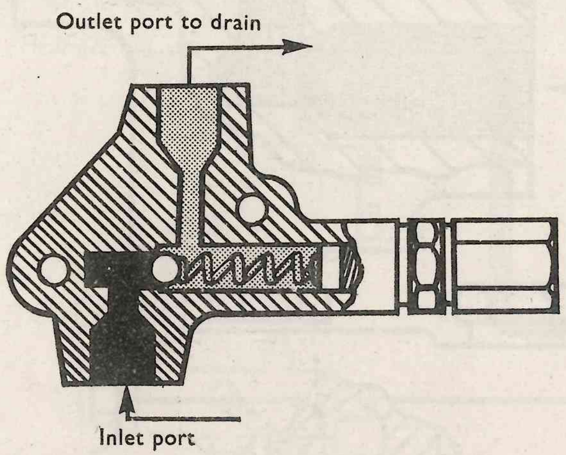


Fig. 8.25. Hydraulic pressure relief valve.

The valve is shown with pressure at the inlet port held by the seated ball. When pressure rises to the setting of the valve the ball lifts to relieve pressure to drain via the outlet port and reseats when pressure at the inlet port returns to normal.

(g) *Pilot-operated Pressure Relief Valve*

With this valve a pilot remote control can operate the valve, if necessary, whilst the integral ball-type relief valve sets an upper safety limit above that normally reached.

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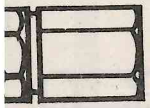

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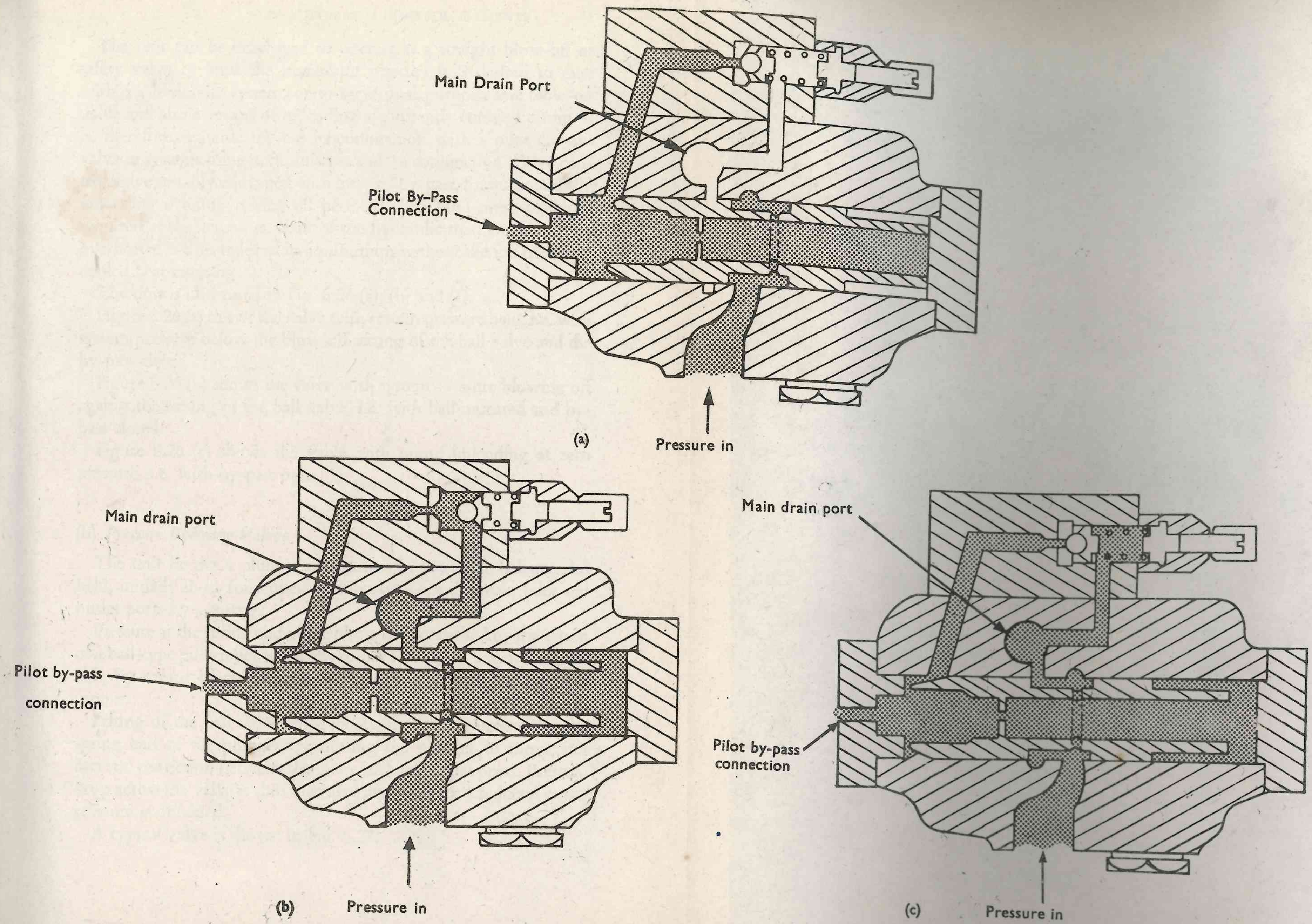


Fig. 8.26. Hydraulic pilot operated pressure relief valve.

The unit can be employed to operate as a straight blow-off or safety valve to limit the maximum pressure it is desired to raise within a hydraulic system, or to serve dual purposes as a blow-off valve and also a means of unloading a constantly running pump. It is, therefore, suitable for use in conjunction with a pilot cut-out valve in systems using accumulators and, in conjunction with a control valve suitably equipped with integral by-pass, for automatically unloading a pump during all periods when fluid pressure is not required at the actuators. Risk of the hydraulic medium becoming overheated is thus reduced to a minimum without the use of water-cooled heat exchangers.

The unit is illustrated in Fig. 8.26 (a), (b) and (c).

Figure 8.26 (a) shows the valve with system pressure held, i.e. with system pressure below the blow-off setting of the ball valve and the by-pass closed.

Figure 8.26 (b) shows the valve with system ssure blowing off against the setting of the ball valve, i.e. with ball unseated and by-pass closed.

Figure 8.26 (c) shows the valve with pump unloading at zero pressure, i.e. with by-pass open.

(h) *Pressure Reducing Valves*

The unit houses a plunger maintained in hydraulic balance and held, initially in its fully open position between the main inlet and outlet ports by a spring.

Pressure at the spring end of the plunger is controlled by the action of a ball-type pilot relief valve, the spring loading of which is adjustable by means of an external adjusting screw with locknut and capnut.

Lifting of the ball valve from its seating reduces pressure at the spring end of the plunger, permitting the plunger to move and increase restriction between the main inlet and outlet ports. Pressure drop across the valve is thus increased and a constant reduced outlet pressure is obtained.

A typical valve is shown in Fig. 8.27.

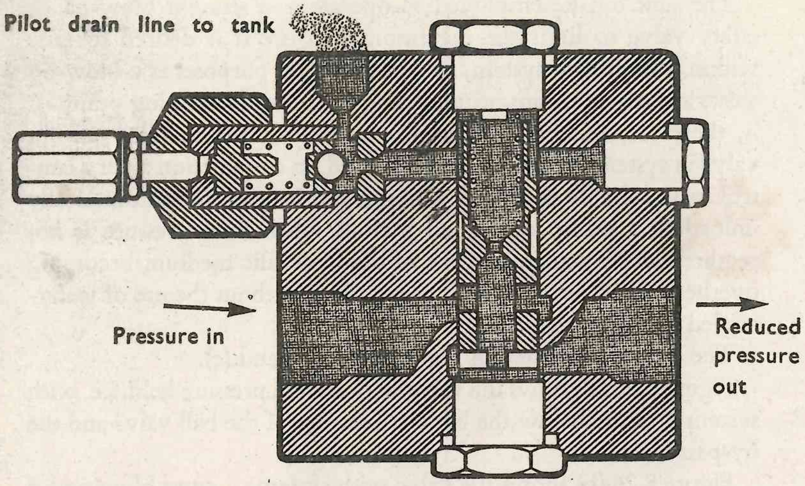


FIG. 8.27. Hydraulic pressure reducing valve.

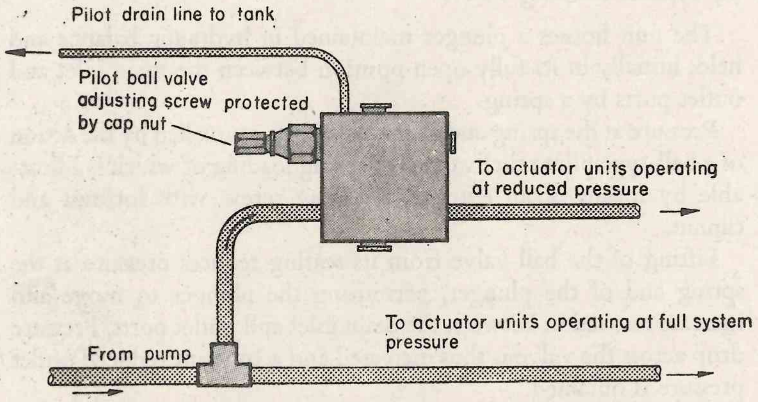


FIG. 8.28. Application of pressure reducing valve.

The pressure reducing valve will permit actuator units of differing maximum permissible working pressures to be operated from the same power source.

The valve can be set, and locked, to limit the maximum pressure to one or more actuator units to, say 1000 psi, whilst other actuator units in the same system operate at the full output capacity of the power source. This is shown in Fig. 8.28.

CHAPTER 9

The Principles of Automatic Control

9.1.

In this chapter the principles and terminology used in the application of automatic controllers will be explained.

(a) *The Uncontrolled Plant*

The example chosen is shown in Fig. 9.1 and consists of an oil-fired boiler feeding several consumers. The object of the control devices is to maintain the temperature of the hot water at a constant temperature. Temperature is thus the *controlled condition*.

As shown in Fig. 9.1 there is no automatic control (i.e. the loop is open) and results are unlikely to be satisfactory because of the following variables each of which will affect the final temperature:

(i) the water supply pressure,
(ii) the mains oil flow,
(iii) the calorific value of the oil,
(iv) the boiler efficiency,
(v) the incoming water temperature,
(vi) the demand of the consumers,
(vii) the ambient temperature.

The first five conditions are known as *supply disturbances*; the next (vi) as a *demand disturbance* and the last (vii) as a *process disturbance*.

The requirement of a controlled process is to balance supply and demand by regulating the one or the other. The choice is determined by the design requirements of the plant. Failure to balance supply and demand upsets the equilibrium and causes the temperature to depart from its desired value. In the case under consideration whilst it would be possible to control temperature by regulating the demand of the consumers obviously this would not be satisfactory to the consumers.

If all the seven variables given above are considered it should be clear that the consumers' demand is bound to be unsatisfactory with

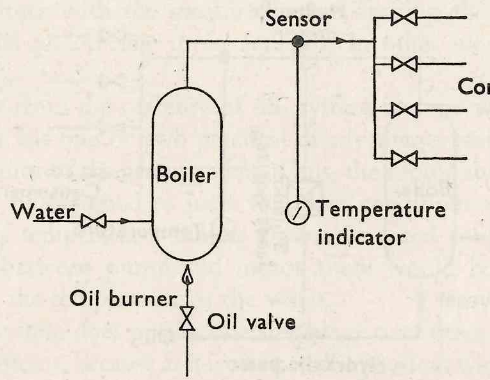


FIG. 9.1. Uncontrolled plant.

an uncontrolled system. However, the conclusion should not be that every process must be fitted with automatic control. Some processes can, by careful design, be substantially free from supply and demand disturbances and in such cases the benefits to be derived from the addition of automatic controls cannot be justified on economic grounds.

In the process under consideration it is possible to either employ a watchkeeper to watch the temperature and regulate the oil flow, or to provide an automatic control system. The various ways of applying automatic control will be outlined in the following sections in order to bring out some of the basic principles.

(b) *Open-loop Control of Fuel–Water Ratio*

The system shown in Fig. 9.2 would give a great improvement in performance over the uncontrolled plant and combines simplicity with reliability and stability.

In the water supply is a positive displacement type hydraulic motor coupled to a similar but smaller pump in the oil fuel line. This arrangement would ensure a fixed volumetric discharge ratio between the water and fuel. Any change in demand for hot water will

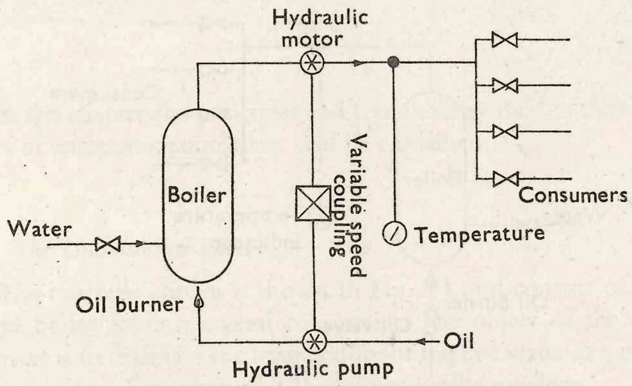


FIG. 9.2. Open-loop control of oil-water ratio.

provide an immediate proportional change in the supply of fuel. By this means the hot water will be fairly closely maintained at a constant temperature. The actual outgoing temperature will be determined, among other variables, by the volumetric discharge ratio of motor and pump. To ensure flexibility a variable speed coupling between the pump and motor would be of advantage.

Such a system would eliminate the following disturbances:

(i) the water supply pressure (supply),
(ii) the mains oil flow (supply),
(vi) the demand of the consumers (demand).

The elimination of these disturbances would give fairly satisfactory results, since these are the main disturbances but, nevertheless, the water temperature would still be unprotected from variations in the following disturbances:

(iii) calorific value of the oil,
(iv) boiler efficiency,
(v) incoming water temperature,
(vii) ambient temperature.

It must be noted at this stage that the system is still of the open-loop type because there is no provision for comparing the desired temperature with the measured value, feeding the answer into a controller and thence to an actuator. In other words there is no feedback.

Apart from the inability of this system to cope with all disturbances, it has one or two practical disadvantages mainly due to the inflexibility of the arrangement. Thus, the motor and pump would have to be designed to meet the plant conditions and the desired value of temperature. Unless a variable speed coupling were included between pump and motor there would be no means of altering the temperature of the water.

The system does possess one advantage over other more sophisticated systems, because as it is an open-loop system with no feedback, there is absolutely no possibility of the type of instability known as "hunting".

(c) *Simple Closed-loop Temperature Control*

In this system which is shown in Fig. 9.3 temperature is measured by a sensor which is connected to a measuring element.

An automatic controller compares the measured value with a signal representing the set value and sends an appropriate regulating signal to a correcting unit (e.g. a diaphragm actuated valve) in the fuel line. Such a system combines economy with flexibility and is the system normally used. It applies corrections for all the seven disturbances described earlier, but the effectiveness with which it suppresses the effects of these disturbances depends on the process characteristics, such as plant lags. For example a sudden increase in the consumers demand would, after a short delay, cause a drop in temperature which, in turn, would cause an increase in fuel flow to take place. However, it might be an appreciable time before the corrective effect of the increase in fuel flow is detected by the sensor.

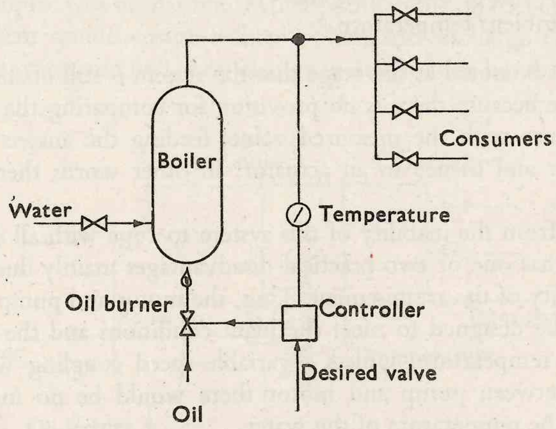


FIG. 9.3. Simple closed-loop control.

During this correcting period there would be a transient deviation in temperature.

Consequently the performance of such a system would be:

(i) the average temperature would be equal to the desired value,
(ii) there would be transient excursions from the desired value whenever a sudden change in any of the seven possible disturbances occurred.

(d) *Closed-loop Control freed from Certain Disturbances*

The transient disturbances suffered by the previous system can be minimized by the addition of extra closed-loop control systems,

which prevent the disturbances from entering the system. This is shown in Fig. 9.4.

The demand disturbance has to be accepted since it is unreasonable to install a flow controller on the hot water supply. However, there is no need to accept disturbances arising from water and oil pressure variations and in consequence two extra control loops have been added, one to control the water supply pressure and the other to control the oil pressure. As pressure controllers are fast in action

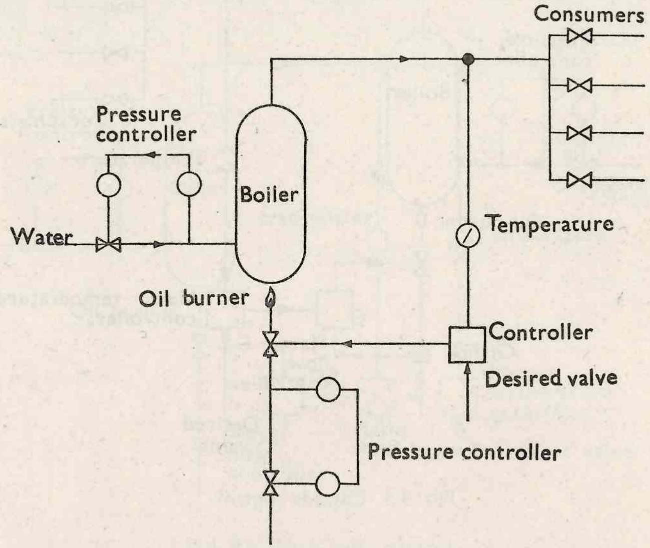


FIG. 9.4. Closed-loop control freed from certain disturbances.

these can be very effective in maintaining constant pressure and, as a result, the temperature controller has a much less disturbed function.

(e) *Cascade Control*

The set value of the controlled condition can be altered manually. It can also be altered by means of another controller. A combination of one controller feeding its output signal to alter the set value of another controller is known as a *cascade system*. The first controller is sometimes referred to as the "master" and the second as the "slave". This is shown in Fig. 9.5.

In this scheme the slave controller is responsible for oil flow and in the absence of a signal from the master temperature controller, it

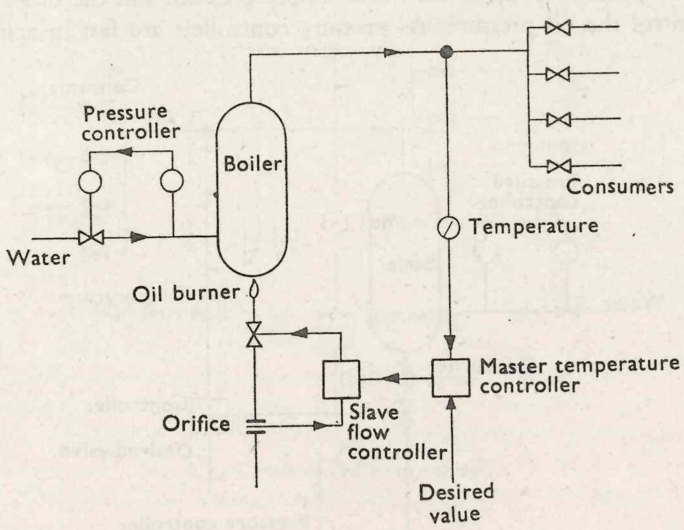


FIG. 9.5. Cascade control.

will maintain a constant flow of oil regardless of oil supply pressure variations. The master temperature controller issues signals to the slave; to the slave controller is delegated the responsibility for carrying out these instructions.

The obvious difference between this scheme and the previous scheme is that only one oil control valve is used, instead of two, which means a saving in cost. Offsetting this, of course, is the additional mechanism for altering the set value of the slave controller in response to the master signal.

The performance of the cascade system is likely to be slightly better than the previous system.

(f) *Flow Ratio Control*

The system shown in Fig. 9.2 incorporated a hydraulic motor and a pump which maintained a constant ratio between water flow and fuel flow. In the absence of a temperature controller this system is

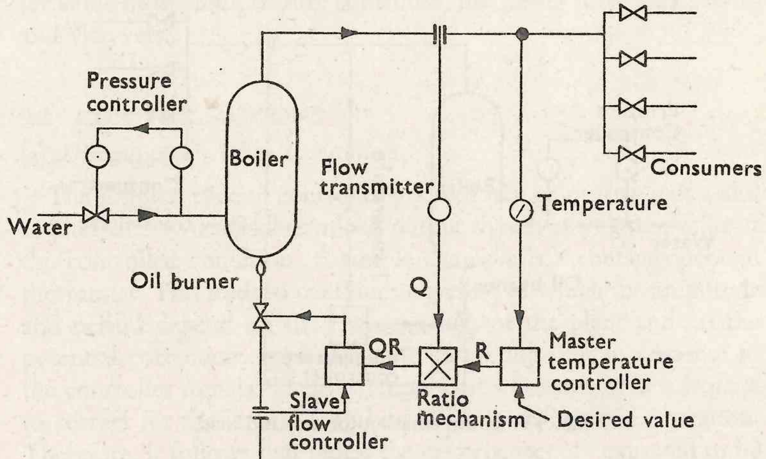


FIG. 9.6. Flow ratio control.

subject to fairly serious temperature deviations as a result of other disturbing influences.

A flow ratio controller such as is shown in Fig. 9.6 has much more flexibility and produces much better control.

The flow ratio controller incorporates a cascade system and consists of a master water flow sensor sending out a signal which is used to adjust the set value of a slave oil flow controller to the required ratio. The output signal from the oil flow controller is then used to control the oil valve.

The signal from the master to the slave can be adjusted by a factor to alter the ratio. Special mechanisms are available so that the output signal from the temperature controller can alter the ratio. In this way the temperature controller superimposes its own demands to maintain a constant temperature.

This system has one important advantage, viz. the demand disturbance is largely neutralized before it affects the temperature because an increase in hot water demand almost immediately increases the fuel flow in proportion, whereas the systems shown in

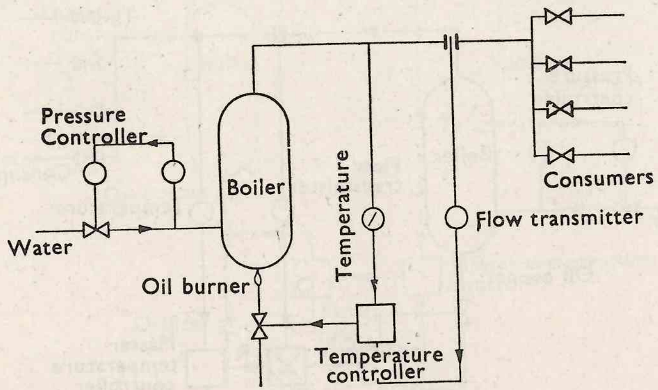


FIG. 9.7. Compensation for demand load.

Figs. 9.3, 9.4 and 9.5 detect only the result of a change in demand as a change in temperature after a time delay (due to time lags in both the plant process and the measurement). The correcting effect of an oil flow change is also delayed by the same time delay and this could be responsible for a serious transient temperature disturbance.

(g) *Compensation for Demand Load*

The next step shown in Fig. 9.7 is really a refinement.

For reasons of control stability the temperature measuring point is best located near to the outlet of the boiler where the time lag in measuring the temperature change resulting from an oil flow change

is at a minimum. However, there is bound to be some heat loss between the boiler exit and the point of usage by the consumer in spite of good pipe lagging.

The real object, of course, is to maintain a constant temperature at the point of consumption. The temperature drop will vary according to the rate of draw off. The faster the rate of draw off the smaller the temperature drop.

One solution of this problem is shown whereby the demand flow is measured by a flow sensor and the output signal is used to alter the set value of the temperature controller, increasing it for small flows and vice versa.

9.2. CONTROL ACTIONS

(a) *Discontinuous Action Controllers*

The simplest type of controller has two step or ON/OFF action and so alternates two pre-determined output signals at a chosen value of the controlled condition. A simple example is a contact-operated thermostat. This leads to continuous cycling of which the amplitude and period depend on the characteristics of the plant and on the potential corrections applied by the correcting unit in response to the controller signals. These potential corrections must be adequate to correct for the largest disturbances likely to occur in operation. Therefore, it follows that unless the disturbances are expected to be always of the same magnitude the potential corrections applied to correct for the smaller disturbances will be too large and hence excessive overshoot and frequency of cycling will occur.

This disadvantage can be overcome to some extent by using two-step action with "overlap" or by introducing more than two steps so that, at pre-determined values of the controlled condition different control signals are produced. In practice, however, the inherent discontinuity of action is a disadvantage and it is usually true that discontinuous controller action can only be used satisfactorily when the demand side capacity is large compared with the supply side capacity and when the disturbances follow a suitable pattern.

For other applications continuous action controllers are necessary.

(b) *Continuous Action Controllers*

The occurrence of a disturbance in a closed-loop system eventually causes the measured value to deviate from the set value and it is the function of the automatic controller to measure the difference (*deviation*) and to convert it into an appropriate output signal which is transmitted to the actuator. The term "controller action" describes the relationship between the deviation and the change of output signal from the controller.

There are three basic types of controller action and they are defined in terms of:

V = change in controller output signal,
θ = deviation,
K_1 = a constant, or "proportional action factor",
K_2 = a constant, or "integral action factor",
K_3 = a constant, or "derivative action factor".

(c) *Proportional Controller Action*

Proportional action is the term given to a controller action when the output signal is proportional to the deviation of the measured value from the desired value.

Alternatively, the rate of change of output signal is proportional to the rate of change of deviation.

$$V = -K_1\theta.$$

For a small range of variation of conditions about the mean working value, the component units of a control system, including the plant, should behave in a linear manner. Therefore, in steady conditions each position of the actuator corresponds to a discrete value of the controlled condition, and the potential correction applied is given by

$$KV = -KK_1\theta$$

where K is a constant for the plant (plus the actuator).

Thus for each value of θ the controller will produce a definite

potential correction as long as the operating conditions remain constant. If a temporary disturbance occurs, which causes a deviation, the controller will apply a potential correction ($-KK_1\theta$) to correct for the change. The effect of the disturbance will thus be reduced and subsequently, if operating conditions have returned to their initial values, the controlled condition will once more be at the desired value. The return of controlled condition to desired value will be through a series of damped oscillations, provided that K_1 is not adjusted to too large a value.

If, however, the disturbance takes the form of a permanent change in conditions, the controlled condition, using this type of controller, will not return exactly to the desired value, but will attain a new equilibrium value known as the "*control point*". The difference between the desired value and the control point is known as "*offset*".

To understand why offset occurs it must be remembered that the potential correction applied is proportional to the deviation. If there is no deviation, then no potential correction can be applied. However, if a permanent change has taken place in operating conditions a sustained potential correction is called for to correct for it. Consequently there must be a sustained deviation to produce the sustained potential correction. Equilibrium will therefore be established with the control point offset from the desired value.

For example, suppose the process consists in heating a continuous flow of water. If the water rate be increased by 10 per cent then 10 per cent more heat is required. The oil valve must open further to supply this heat and this can only be done if the water temperature falls below desired value by an amount sufficient to give the required valve opening.

The temperature will continue to fall below the desired value (θ_1° C), until at a given deviation (θ° C) the valve is open wide enough to maintain the water temperature at $(\theta_1 - \theta)$° C. This deviation θ°C is the offset and it will persist, using this type of controller, as long as conditions remain unchanged.

The value of the offset can be found by considering the value of the deviation which would have occurred with no controller, say θ_p. The potential correction required to correct completely for this is thus $-\theta_p$. If the controller reduces the actual deviation to the value of the offset, say, θ_x then the potential correction it has applied is $-(\theta_p - \theta_x)$.

But potential correction $= -KK_1\,\theta_x$

$$\therefore\ KK_1\,\theta_x = \theta_p - \theta_x$$

$$\therefore\quad \theta_x = \frac{\theta_p}{(KK_1 + 1)}.$$

It will be obvious that as KK_1 increases the offset decreases for a given value of θ_p. The value of K is determined by the plant so that the value of offset is inversely proportional to K_1, the proportional action factor.

In many cases of control, especially where operating conditions are not too onerous the existence of offset may not be objectionable. For others, however, and especially in the control of boiler drum level, steam temperature and steam pressure control, the condition is not acceptable, and it is essential that the deviation shall not be permanent but will reduce to zero after a time. This return to zero can be achieved by incorporating integral action.

(d) *Integral Controller Action*

Integral action is defined as an action where the output signal changes at a rate proportional to the value of the deviation and is expressed:

$$\frac{dV}{dt} = -K_2\,\theta$$

or

$$V = -K_2 \int \theta\, dt.$$

Therefore as long as any deviation exists the integral action signal will continue to increase, and finally, therefore, the deviation due to a given load change must be reduced to zero.

Reset action, i.e. removal of offset, has one inherent drawback. It creates instability during system start up when the value of the controlled variable is well below the instrument set point. This causes the integral action to work very hard in order to eliminate the discrepancy and this can result in wide cycling and prolonged instability. Integral action, therefore, should only be brought into effect when the system has reached steady conditions.

Similarly too great an increase in integral action (which is adjustable) can cause hunting of the variable about the set point.

The great majority of marine control problems can be solved by a judicious use of proportional and integral actions. There is, however, a third control action which is used where the process has a considerable time lag, e.g. temperature control of plants with large thermal storage. Such control action is known as derivative action.

(e) *Derivative Controller Action*

With this action the output signal is proportional to the rate of change, i.e. acceleration, of the deviation and may be expressed as:

$$V = -K_3 \frac{d\theta}{dt}.$$

It will be obvious that derivative action, dependent only on rate of change of deviation, will not produce any corrective action (whatever the value of the deviation) if the deviation is not changing. It is little used in marine systems.

Proportional and integral controller actions are used both singly and in combination with each other. Other combinations are $P + D$ and $P + I + D$. Derivative action cannot be used by itself.

Figure 9.8 illustrates the output response of the three types of controller action to (i) a step input deviation and (ii) a steadily rising input deviation.

(f) *Proportional Band or Band Width*

The "proportional band" of a proportional controller is the range of input signals which will cause the output signal of the controller to vary over its whole working range. It is frequently given as a percentage of the full-scale range of the measuring element. Usually the full working range of controller output signal is arranged to be equal to the full operating range of the actuator, but this is not always done.

For example, the diaphragm operated control valve is a type of actuator used in many control systems and it is usually designed to change from shut to full open (or vice versa) when the pressure

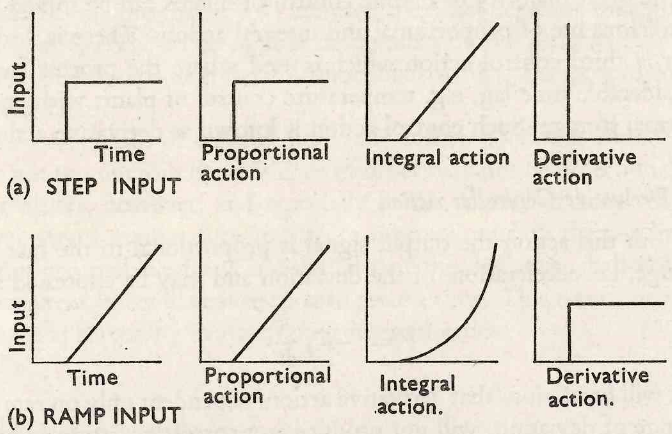


FIG. 9.8. Controller actions.

applied to the diaphragm changes from 3 psi to 15 psi. The range of the output signal from the controller used with such actuators is generally 3 to 15 psi also. The proportional band of the controller will then be the range of deviation which causes the output signal to change from 3 to 15 psi. (It must be appreciated that higher pressures may be used, particularly with cylinder-type actuators, to obtain greater working forces and faster response.)

Suppose a change of temperature of 40°C causes the output pressure of a temperature recorder/controller to change by 12 psi then the width of the proportional band is 40°C.

Alternatively if the full-scale range of the recorder were 200–400°C then the proportional band width can be defined as 40/200 or 20 per cent (of full-scale range).

The dimensions of the proportional band width are the same as those of the controller input signal, and the band width is given numerically by:

$$\text{Proportional band width} = \frac{{}_mV}{K_1} \quad \text{(units, say °C)}$$

where V_m = full working range of controller output signal (units, say, psi),

K_1 = proportional action factor of the controller (units, say, psi/°C).

In other words proportional band width is inversely proportional to proportional action factor (K_1).

If the full operating range of controller output signal is equal to that of the actuator, as is often the case, and if the proportional band is symmetrical about the desired value, then the greatest offset which can occur is equal to half the proportional band width. This is true only if the actuator can apply a potential correction, θ_m, when it is full open such that $\theta_m \geqslant \theta_p$ where θ_p = potential deviation due to change in plant conditions.

If $\theta_m < \theta_p$ control is lost and the final equilibrium deviation will be $(\theta_p - \theta_m)$.

The aim in designing a control system is to ensure that, after a disturbance, the following shall be reduced to a minimum:

(i) deviation,
(ii) the time of return to control point,
(iii) offset.

Considerable experience and knowledge is required to decide which type of controller to adopt, or when setting the proportional band width and action times of the controller. The decisions are influenced by plant characteristics, required performance and economics. In one application it may be essential to reduce peak deviations to a minimum, in another considerable deviations can be tolerated for short periods but perhaps even small deviations cannot be permitted for long periods. The subject is too complex to be dealt with briefly.

It may be sufficient to say that the proportional, integral and derivative controller actions bear a close similarity to the reactions of a human operator manually controlling a process as illustrated in the following examples:

(i) A sudden increase in temperature will cause the operator to close the oil valve by an amount proportional to the increase (proportional action).

(ii) A sustained, but constant deviation will cause the operator to adjust the valve, little by little, in the right direction until the deviation is eliminated (integral action).

(iii) A rapid increase in temperature, whether it is below the set value or not, will cause the operator to close the oil valve more than he would for a slow rise in temperature (derivative action).

Single term (or proportional action) will achieve reasonable accuracy in systems where load changes are not large. Integral action is used where large and frequent load changes are expected. Integral action can compensate for offset but cannot rectify the initial overshoot experienced following large load changes. Conversely, derivative action can minimize overshoot but does not affect offset.

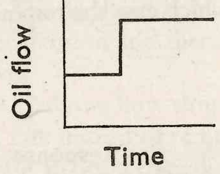
For the temperature control loop shown in Fig. 9.3 a three-term PID controller would be desirable. For the less important pressure control loops shown in Fig. 9.4 proportional controller action alone might well be sufficient.

9.3. PLANT CHARACTERISTICS

The characteristic of a plant can be described in terms of the input/output relationship. It can be measured by injecting a predetermined input disturbance at some point into the plant and analysing the output response. A common input disturbance signal

is in the form of a step. The output response can generally be recognized as one of three types—to illustrate these, reference will again be made to the boiler application previously described.

(a) A step increase in the oil flow will cause an exponential rise in the water temperature as measured at a point on the outlet of the boiler. This is shown in Fig. 9.9.



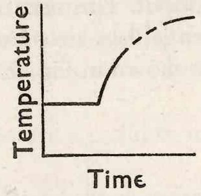


FIG. 9.9. Exponential lag.

This is due to the capacity effect of the boiler, which absorbs heat until equilibrium conditions are reached. This section of the plant is said to have an *exponential lag*.

(b) If live steam is injected into the pipe at a point close to the outlet of the boiler, and the temperature is measured at another point much further along the pipe, the response will appear like that shown in Fig. 9.10.

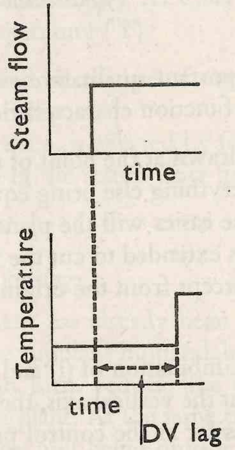


FIG. 9.10. Distance velocity lag.

The part of the system comprising the long length of pipe will be described as having a *distance/velocity lag* defined as the time interval between an alteration in the value of the signal and the manifestation of the alteration at a later part of the system, and arising solely from the finite speed of propagation of the signal.

(c) There is a third type of response from a plant subjected to a step input. This occurs when a combination of successive exponential lags are present, in which case the response is in the form shown in Fig. 9.11.

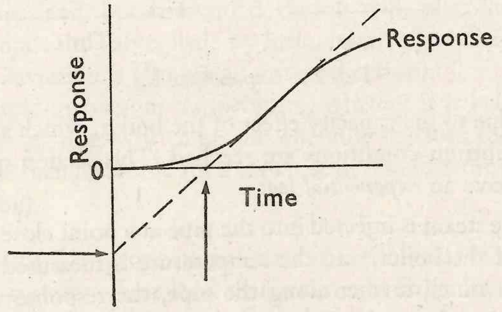


FIG. 9.11. Typical plant lag.

There are two important qualitative conclusions which can be drawn from the step function characteristic.

(i) If a tangent is drawn at the point of inflection of the response curve, then everything else being equal, the smaller the angle of the slope, the easier will the plant be to control.

(ii) If the tangent is extended to cut the horizontal axis, then, the shorter the intercept from the origin, the easier is the control problem.

It follows from a combination of (i) and (ii) that if the tangent is extended further to cut the vertical axis, then the smaller the vertical intercept the better, as far as the control problem is concerned.

9.4.

(a) *Computers. Simple Computation*

Simple computation includes a few arithmetical operations such as adding, subtracting, multiplying and dividing. Devices have been produced for adding a number of signals from sensing elements or combining, by addition or subtraction, a number of signals to control units. The application of ratio control, described earlier, is an example of division and there are other cases where multiplication is employed.

Figure 9.12 shows how simple computation could be used to control B.t.u. rate instead of temperature.

In this case, in order to make the application look more reasonable it is assumed that only the B.t.u. of the recirculated water is controlled and that there is another uncontrolled hot water consumer. Without this additional uncontrolled consumer, constant B.t.u. would be obtained simply by keeping the oil flow constant.

The expression for computing B.t.u. rate is:

$$\text{B.t.u.} = Q(T_1 - T_2)$$

where Q = water flow (lb/hour),
T_1 = flow temperature (°F),
T_2 = return temperature (°F).

Computation is achieved by the use of one subtraction unit and one multiplier.

Automatic control of B.t.u. is effected by feeding the signal from the controller to a valve in the return flow line.

(b) *Computers—Data Processing*

Data logging equipment has already been described. One of its primary applications is to supplant human labour in logging instrument readings which, in turn, reduces the number of recording instruments required in a plant. At the same time it sends out alarm signals in the event of any unit being outside limits.

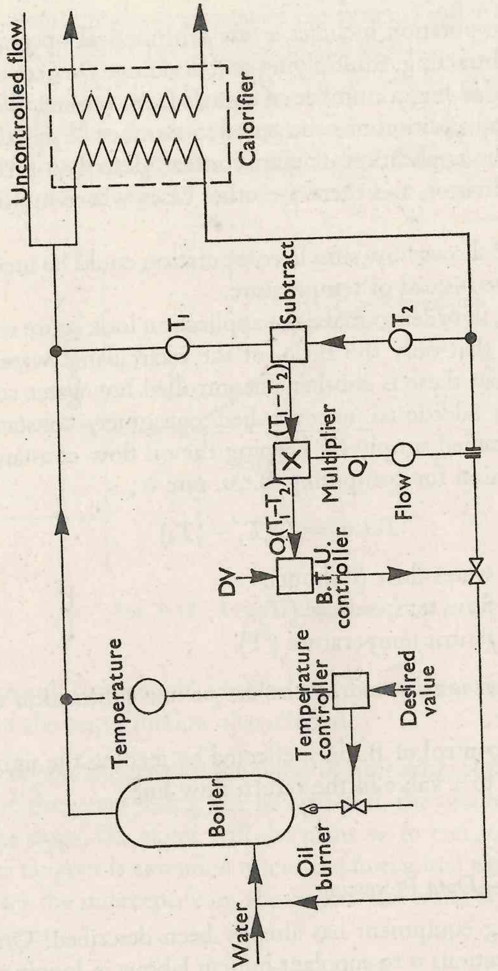


FIG. 9.12. Simple computing.

In the control field data processing equipment is used to obtain more information about the plant process—a vital step towards the ultimate target of controlling the whole plant by means of a computer.

If justification could be found for fitting data processing equipment to the boiler installation already described, it would take readings of all the seven variables together with other measurements such as, air flow through the boiler, flue gas temperature, oxygen and carbon monoxide analysis of the flue gases, and present all this information at regular intervals or on demand.

In addition it could be designed to carry out calculations on secondary data, a typical example being boiler efficiency. At this stage the secondary data would not be used directly for control but would permit those in charge to make certain decisions to improve the efficiency of control.

As the purpose of such a computer is merely to present information, and not to control the decisions still being made by the human operator, such an application is known as "off-line" computation.

(c) *Computers—Wired Type and Stored Programme Type*

The wired type is specifically designed and wired to perform a predetermined and fixed sequence of operations.

The stored programme type would receive the same number of inputs and provide the same number of outputs as the wired type. However, the major advantage of the stored programme type of computer over the wired type is that the control sequence can be changed comparatively simply by modifying the programme.

Modifying the control sequence in a wired-type computer involves rewiring the equipment and probably could not be economically justified in most cases. Some versatility in wired control can be achieved through modular design utilizing alternative plug in boards. Programme variations may then be made by the addition or removal of the plug in boards to modify certain portions of the system. However, the degree of modification to the programme is still limited.

(d) *Computers—Control*

A computer can perform three types of functions in a system; the recording functions, the supervising functions and the control functions. Figure 9.13 shows how these would apply to a system.

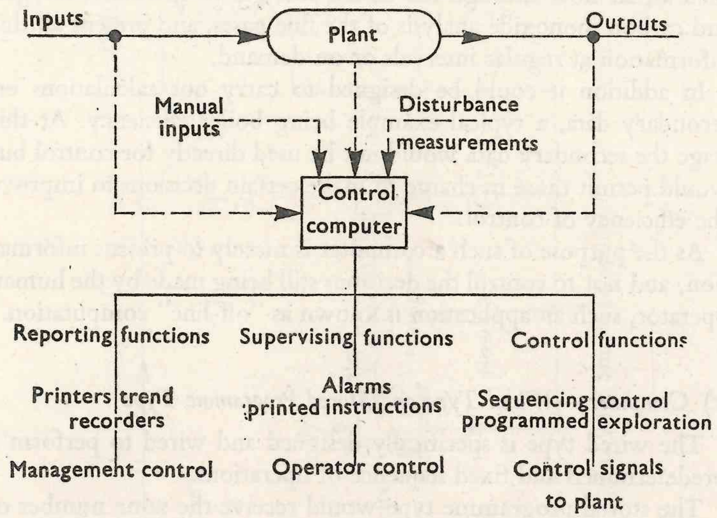


FIG. 9.13. The control computer.

In the recording and supervising modes the computer presents tabulated information obtained from the sensors and also acts as an important link between the plant and the operator.

The control functions represent the more advanced operations although the sequencing control functions are relatively simple and involve application of the techniques already described. Sequencing control involves logical decisions in carrying out emergency actions, in starting up and shutting down items of machinery or in changing the mode of operation.

In most business or scientific computers the speed of computation, and input/output operations, is dictated not by the nature of the information to be processed but rather by economic considerations. Fast input/output units are provided to utilize the internal high-speed capability of the computer but a slow computer could equally well be used, since information is processed on a historical rather than a real time basis. In other words, one can wait until the problem being processed is completed before requesting the computer to solve a new problem.

Control computers, although similar to general purpose business or scientific computers differ from them in many respects. For example, since control computers handle real time problems, their computational speed is relevant and one can draw a line between problems that can and those that cannot be handled by any one-control computer. Since a control computer must communicate with a real time process as well as with a human operator, the speed, nature and number of input/output devices required are also different from those of a business or scientific computer.

The area of input/output operations—the means of computer communication with the outside world—is the area where the greatest differences exist between business or scientific computers and control computers. In addition to the conventional input/output devices such as paper tape or card readers, paper tape or card punchers, line printers, etc., there is need for the following input/output operations.

Analogue Inputs. The plant to be controlled by the control computer may have anything from a few to several hundred or several thousand analogue continuous variables. The signals representing these variables may be electrical, pneumatic or mechanical. It is necessary that these signals should be sensed by the control computer and converted to digital form for storing in the computer memory. This conversion is carried out in the analogue to digital (A/D) converter the input to which is usually a voltage. Consequently any variable not in that form requires an appropriate transducer, and also since there are many variables some type of switching in the input of the A/D converter from one input signal to another is required.

Analogue Outputs. The control computer exercises control over the plant by computing the values of certain variables for optimum response. These computed variables are in digital form and they must be converted into analogue signals. This is done in a digital to analogue converter. The analogue output signals are used either as reference signals for set point controllers or as dynamic variables in the actual control system.

Digital Inputs. Another type of information that the control computer must be able to obtain from the plant is digital information. This information can be in two forms:

(i) Pulse signals representing continuous variables, e.g. the output of some types of telemetering devices, the output of turbine type flowmeters, digital clock, etc.
(ii) Contacts representing the on/off state of various devices in the system.

Digital Outputs. The digital output requirements of a control computer fall into the same two categories as the digital inputs from the plant.

(i) Pulse output signals. It is possible to use variables computed in digital form as inputs to digital telemetering transmitters.
(ii) Contact outputs for actuating devices such as circuit breakers, valves, displays, etc.

Although general purpose business computers have some means of manual inputs by, and displays for, the operator, the need for such operator/computer communication is much greater in "on-line"

applications. Some of the functions which it would be desirable that the operator should be able to perform through the operator's computer console are:

Initiate start up or shut down procedure.
Initiate or interrupt various input scanning sequences.
Add or eliminate points from a scanning sequence.
Demand logging of any number of input variables.
Request the display of any quantity stored in the computer memory.

Interrupt and Priority Control. In case more than one operation has to be performed at any time, the computer should be able to recognize the relative importance of each and assign a priority order in which operations are to be performed. When an operation is interrupted the computer should be able to "remember" where it left off so that later it can continue from the same point. Some examples of interrupt and priority control features in on-line computer applications are:

In case of a plant emergency detected by the closing or opening of a contact or a combination of contacts, the computer should interrupt its present operation and proceed with an emergency shutdown programme.

In case of computer failure detected by built in self-checking circuits the current computer operation should be put to manual control.

If it is required that an operation (e.g. logging) be performed at a particular time, then the clock should be able to interrupt whatever operation is going on (provided it is not of higher priority) and initiate the logging operation.

CHAPTER 10

The Basic Principles of Automatic Controllers

10.1. PNEUMATIC CONTROLLERS

Fundamentally the actions of all pneumatic controllers are generated by the same methods although there is a wide variety of generating units based on the same general principles. The development of these units has been aimed at increasing their reliability, consistency of operation, ease of setting and maintenance. Any of the well-known makes of controller on the market will meet most of the present process requirements as far as their general performance is concerned. It is, however, a matter of individual choice to decide which possesses most reliability (which is the first essential for any controller), consistency and is the most economic to install and maintain.

In this section the basic principles employed in pneumatic controllers will be described. Detailed descriptions of proprietary designs are best obtained from the manufacturer's literature.

(a) *Proportional Action*

The basic principle used to generate proportional action is shown in Fig. 10.1.

The supply air, at constant pressure, flows through a constriction to a nozzle from which it escapes to atmosphere depending on the

position of the flapper. (Note: some manufacturers refer to the flapper as a "baffle" or a "vane".)

The constriction is designed to give a greater pressure drop than the nozzle. Suppose the constriction gives four times the greater pressure drop. Then, if the flapper is too far away from the nozzle to affect the flow and the constant supply pressure is 20 psi the pressure (p) between constriction and nozzle will be 4 psi.

If the flapper were then moved up to the nozzle so that it completely closes the nozzle, then the pressure (p) will rise to 20 psi. If

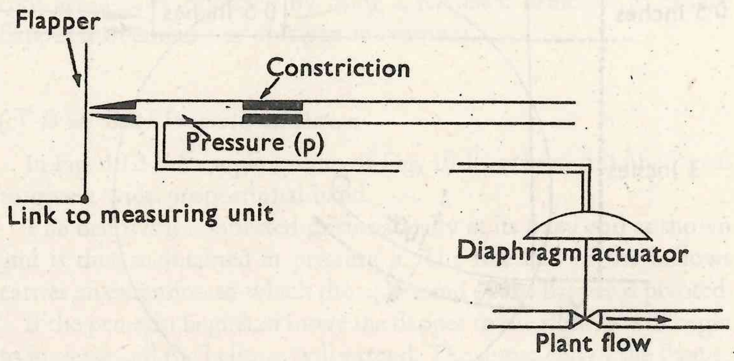


FIG. 10.1. Generation of proportional action (pneumatic).

the flapper is now moved slowly away from the nozzle the pressure (p) will drop again to 4 psi. Only a very small movement of the flapper is necessary to achieve this. The necessary movement can vary between a few thousandths to a few tenths of a thousandth of an inch. In one widely-used controller a movement of 0·00025 in. is used.

Over such small ranges of movement the change in pressure (p) is proportional to the change in distance of the flapper from the nozzle. Hence such a system produces proportional action and it is obvious that it will be very sensitive to movement of the flapper relative to the nozzle.

(b) *Narrow Band Proportional Action*

Consider such a system applied as shown in Fig. 10.2.

Here the flapper is shown linked to the pen arm (YY') of a circular chart temperature recorder/controller with a range 0–200°C corresponding to a pen movement of 3 in.

Suppose that a movement of the flapper relative to the nozzle of 0·002 in. changes the pressure (p) over its full working range. With

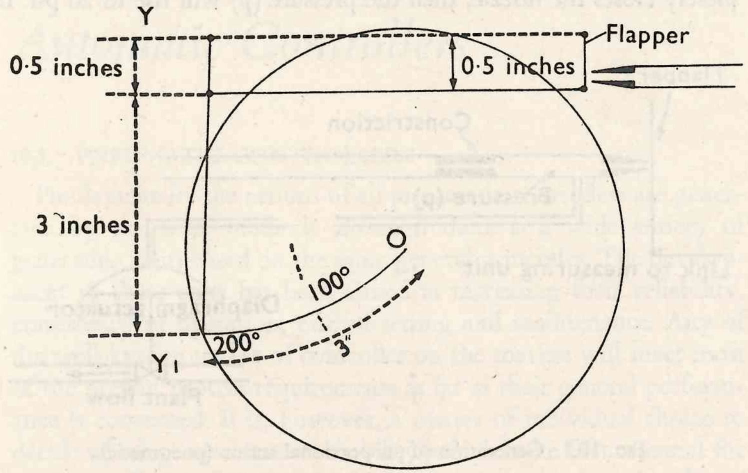


FIG. 10.2. Flapper/nozzle sensitivity.

the dimensions shown a flapper movement of 0·002 in. will be given by a pen movement of

$$0{\cdot}002 \times \frac{3{\cdot}5}{0{\cdot}5} = 0{\cdot}014 \text{ in.}$$

This corresponds to 0·93°C and represents 0·47 per cent of full scale.

Thus the proportional band width of such a controller is less than 0·5 per cent which, for practical purposes, is equivalent to "two-step" (or on/off) controller action. Such an action will result in the

continuous cycling characteristic of two step control and obviously some means of reducing the sensitivity of the system must be adopted. In other words a wider proportional band must be provided.

In order to obtain a band width of even 10 per cent, the ratio of flapper movement to pen movement must be made 20 times smaller. Linkages and mechanisms giving reductions of such an order can be constructed without introducing any sensible backlash or friction, but reductions of a higher order demand an uneconomic amount of precision in manufacture.

Consequently proportional band widths wider than about 10 per cent are usually obtained by using a feedback principle to reduce flapper movement per unit pen movement.

(c) *Wide Band Proportional Action*

In Fig. 10.3 the simple system of Fig. 10.1 has been modified so as to give a wide proportional band.

The bellows is connected pneumatically at its fixed end as shown and is thus maintained at pressure p. The free end of the bellows carries an extension to which the upper end of the flapper is pivoted.

If the pen arm begins to move the flapper to the right, p will begin to increase and the bellows will extend. The upper end of the flapper will thus be moved to the left. Hence the movement, due to the pen arm, of the part of the flapper opposite the nozzle will be reduced. Thus a decrease in sensitivity is provided. The extent of the decrease in sensitivity depends on:

(i) the ratio of a:b,

(ii) the extension of the bellows per unit change in p.

Variation of either or both of these factors will cause a change in proportional band width but, in practice, variation in the ratio a:b is usually used as a means of altering band width.

This pneumatic feedback system is very simple yet it enables the positioning of the flapper with a precision of the order of 1×10^{-6} in. With such a system proportional bands up to 600 per cent are available but it is rare to use a band width greater than 300 per cent.

(d) *Setting the Desired Value*

There is only one separation distance between flapper and nozzle which will give any required controller output pressure (p). The systems shown in Figs. 10.1 and 10.3 will thus give a change in output pressure proportional to the displacement of the pen from the position at which p is a minimum, say, 3 psi. Suppose this corresponds to a measured temperature of 50°C and maximum output,

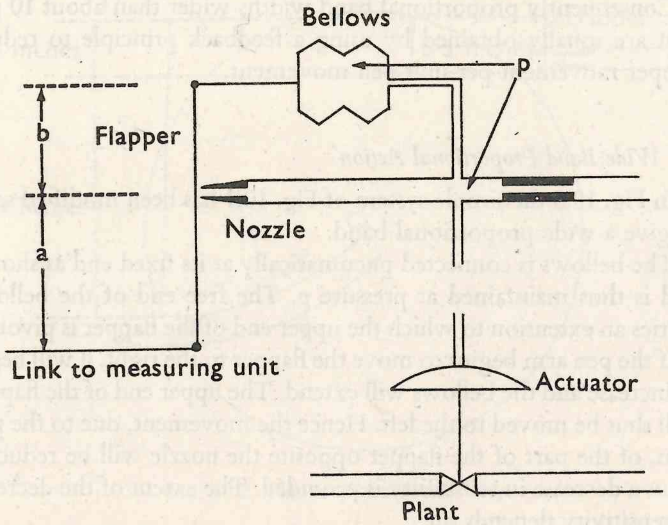


FIG. 10.3. Wide band proportional action (pneumatic).

say, 15 psi is given at 60°C. The system should be set up so that the output pressure is the mean of maximum and minimum values, i.e. 9 psi at the desired value; in this case 55°C.

It is set up so that the position of the flapper relative to the nozzle depends on the measured value of the deviation, i.e. the difference between the measured value, as shown by the pen and a given desired value. A desired value pointer is usually provided to indicate the setting on the chart or scale, and when the pen and the desired

value pointer are aligned at the same reading the controller output is adjusted to be 9 psi; i.e. the flapper occupies a mid-position in its operating range of movement relative to the nozzle.

It must be arranged that the desired value setting remains constant irrespective of adjustments to proportional band width. Conversely the proportional band width must not be affected by changes in desired value settings.

(e) *The Pneumatic Relay*

The performance of the system shown in Fig. 10.3 is limited by the slow rate at which the pressure p will change when the flapper moves to a new position if the actuator (and the line from the controller to the actuator) has any considerable capacity. The rate of increase of p is limited by the rate at which the supply of air flows through the constriction C. The rate of decrease is limited by the rate of air leakage to atmosphere through the nozzle.

The diameter of the nozzle must be small so that the force exerted on the flapper by the air jet is negligible compared with the working force available to deflect the measuring unit pointer (or pen arm). This is very important when the measuring unit is one which must be deflected with only a small working force, e.g. a milliammeter. When it is servo operated, e.g. a self-balancing potentiometer, the working force available to position the flapper is of a higher order and larger nozzle diameters are possible.

From this point of view and also to minimize wastage of air it is normal practice to make the nozzle diameter as small as practicable. The limitation of lower limit on nozzle and constriction diameter is set by the necessity of avoiding blocking up by small particles of dust, oil or water which are not removed by the filters normally used. Very elaborate precautions are necessary to obtain completely dry air, free of all oil and dust. Generally, therefore, the nozzle diameter is of the order of 0·02 to 0·04 in. and the diameter of the constriction may be one-half or one-third of these values. Clearly it is very important that the filter units are properly maintained.

Because of these small diameters of nozzle and constriction a relay valve, such as that shown in Fig. 10.4 is used to provide more rapid response.

In Fig. 10.4 the valve, v, is positioned by the bellows or capsule, b, according to the value of the pressure, p. The relay is designed so that when p is at its maximum value, the valve cuts off the air supply to the actuator, which is then vented to atmosphere through the leak provided. When p has its minimum value, the leak is sealed and full air pressure is applied to the actuator. At intermediate values of p

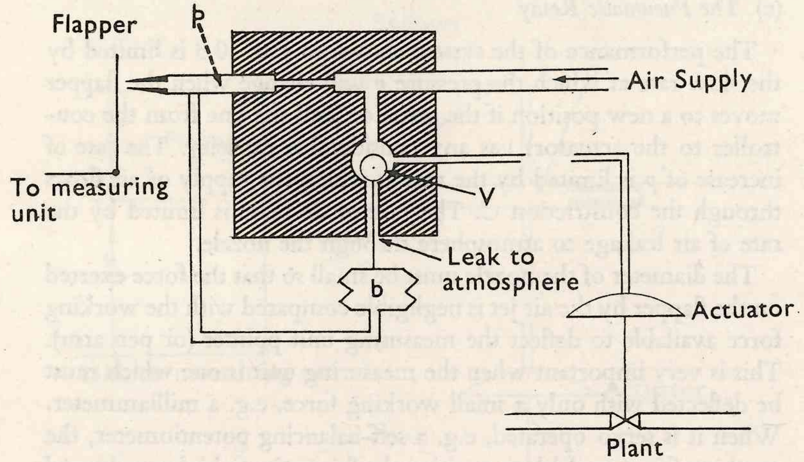


FIG. 10.4. Reverse-acting relay (pneumatic).

the controller output is inversely proportional to p. Such a relay is therefore said to be "reverse-acting".

Using the same principle a direct-acting relay in which output pressure is directly proportional to p can be made. This is illustrated in Fig. 10.5.

The range of pressure p required to operate v and hence give full range of controller output pressure P, depends on the extension of the bellows or capsule b per unit pressure applied. Hence P can be arranged to vary from, say, 3 to 15 psi for a variation in p of, say, 4 psi. Such a pressure amplification by the relay valve eliminates the

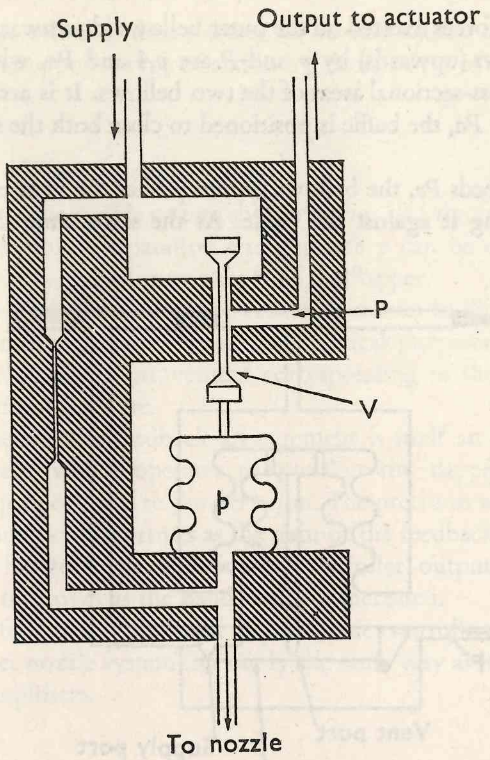


FIG. 10.5. Direct-acting relay (pneumatic).

necessity of arranging for the flapper to seal the nozzle in order to build up the full output pressure specified for the controller.

A different type of relay is shown in Fig. 10.6.

This is designed to avoid the bleed to atmosphere which occurs continuously in the designs of Figs. 10.4 and 10.5. Consequently this design is known as a "non-bleed" type whereas the two previous designs are known as "continuous-bleed" type.

The operation of this relay depends on the opening and closing of the supply and vent ports by the baffle according to the values of p

and P. The forces exerted on the outer bellows (downwards) and the inner bellows (upwards) by p and P are pA and Pa, where A and a are the cross-sectional areas of the two bellows. It is arranged that when $pA = Pa$, the baffle is positioned to close both the supply and vent ports.

If pA exceeds Pa, the bellows system pushes the vent port downwards closing it against the baffle. At the same time the baffle is

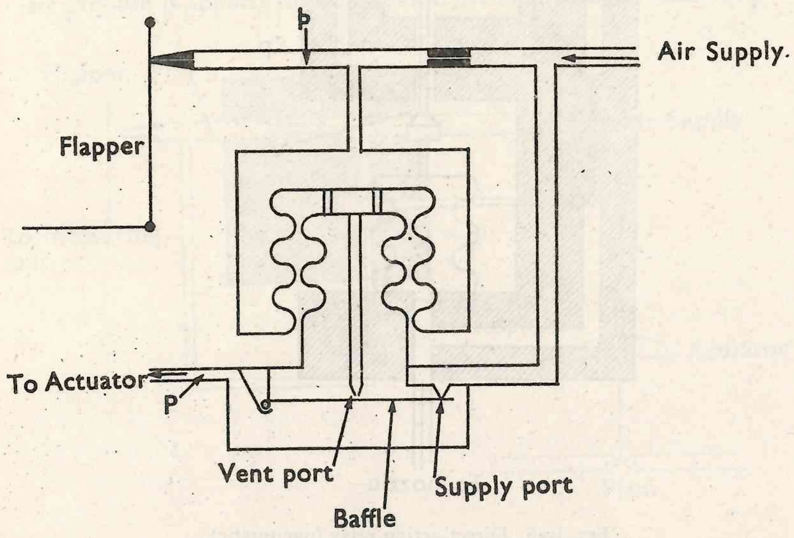


FIG. 10.6. Non-bleed relay valve (pneumatic).

depressed so that the supply port is opened. Therefore P builds up again until pA equals Pa at which time the baffle has again moved upwards with the bellows system to close the supply port. If p then remains constant, P will remain equal to pA/a and no air bleed occurs.

If p decreases, so that Pa exceeds pA, the vent port is moved upwards, away from the baffle, thus allowing air to bleed from the transmission line until Pa once more is equal to pA. Thus the relay is direct-acting.

Most relays whether direct-acting or reverse-acting can be arranged for the opposite type of action by change over of a linkage.

(f) *Linearity of Flapper-nozzle System*

Referring to the simple system shown in Fig. 10.1 the relation between flapper/nozzle separation and pressure p can be considered linear but only for small movements of the flapper.

When feedback is added to the system as shown in Fig. 10.3 the output becomes sufficiently linear, for practical purposes, over the whole range of flapper movement corresponding to the range of controller output pressure.

This is because the feedback arrangement is itself an automatic control system which operates to position the flapper so that pressure p is proportional to the deviation. The precision with which this can be achieved improves as the gain of the feedback system is increased. Therefore the linearity of controller output/deviation relationship improves as the band width is increased.

Negative feedback is thus used in pneumatic controllers to linearize the flapper nozzle system in exactly the same way as it is used in electronic amplifiers.

(g) *Integral Action*

If a control system is set up using proportional action alone, each deviation, θ, from the desired value produces a proportional change in output pressure given by $V = K_1 \theta$. Therefore as explained in Chapter 9 the effect of a sustained load change is to produce offset.

Offset can be eliminated by adding an integral action unit. This gives an increasing contribution to the output pressure until no deviation persists. Such a unit is designed so that its contribution, V_1, increases at a rate proportional to the deviation. Thus

$$\frac{dV_1}{dt} = -K_2\,\theta \quad \text{or} \quad V_1 = -K_2 \int_0^t \theta\, dt.$$

The value of V_1 at any time, t, is thus dependent on the integral of the deviation, measured from a time when the system has been running in equilibrium at the desired value and when $V_1 = 0$.

Consequently such action is called integral action.

Figure 10.7 shows the commonly used method of generating integral action.

This consists of a bellows of capacity, C, and a restrictor of resistance, R. If a step change, p_1, is made in the input pressure, when the system has been in equilibrium, the pressure in the bellows, p_2, will commence to change at a rate proportional to p_1.

If it is arranged that p_1 is proportional to the deviation θ then the rate of change of p_2 will be proportional to θ initially.

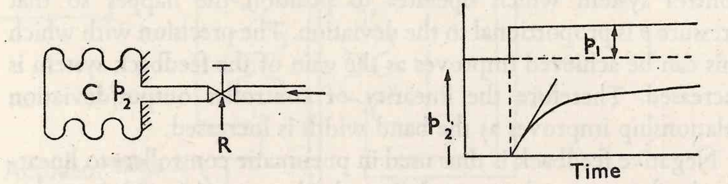


FIG. 10.7. Generation of integral action (pneumatic).

If the output signal from a controlling unit, V_1, is proportional to p_2 then dV_1/dt will be proportional to θ and this is integral action.

Usually the bellows will extend proportionally to the change in p_2 and an air pressure proportional to p_2 can be obtained by using a flapper/nozzle system. The bellows extension per psi change in p_2 is usually controlled by a spring fitted in the bellows.

It must be noted that dp_2/dt is proportional to p_1 only initially. Subsequently dp_2/dt decreases progressively as the pressure drop across the restrictor decreases, due to the build up of p_2. Hence it is only at first that true integral action is generated. Subsequently the departure from int gral action increases continuously.

In practice it is usual to provide both proportional plus integral action. This principle is shown in Fig. 10.8.

When a step change in deviation occurs the pressure in the integral bellows will begin to increase at a rate proportional to the deviation. The pressure in the integral bellows opposes the pressure in the proportional bellows. The end B of lever AB will move to the right as p_2 builds up and the effect of this is to decrease flapper/nozzle separation. This adds a continuously increasing component to the output pressure p of the controller.

As explained above, initially the rate of increase of p_2 is

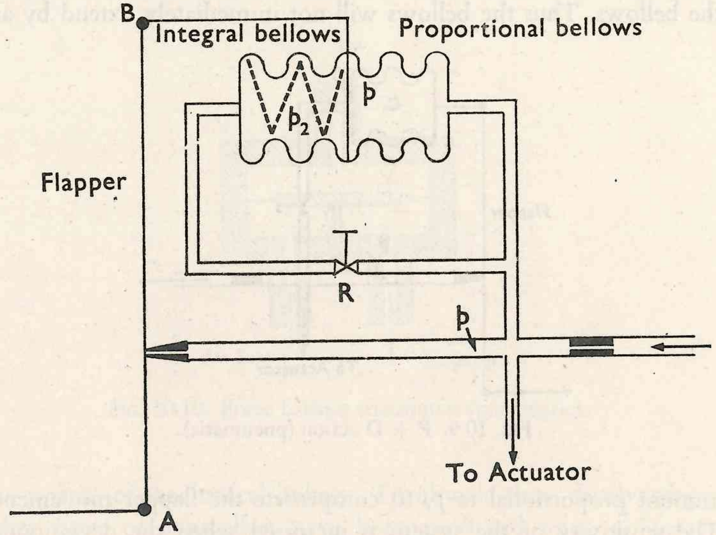


FIG. 10.8. P + I action (pneumatic).

proportional to the deviation and, therefore, initially the addition to the output pressure will also be proportional to $d\theta/dt$. Hence, proportional plus integral action is generated.

(h) *Derivative Action*

Derivative action is added to a proportional action controller in order to produce a phase advance in the controller output signal, i.e.

its function is to produce a control correction sooner than would be possible with proportional action alone. It is often regarded as providing an anticipating action. As explained in Chapter 9 it has no direct effect upon offset.

A system of generating proportional plus derivative action is shown in Fig. 10.9.

This shows that the output pressure p of the controller is applied to the bellows through a resistance R. If there is a change in deviation θ, the full change in p will not be transmitted immediately to the bellows. Thus the bellows will not immediately extend by an

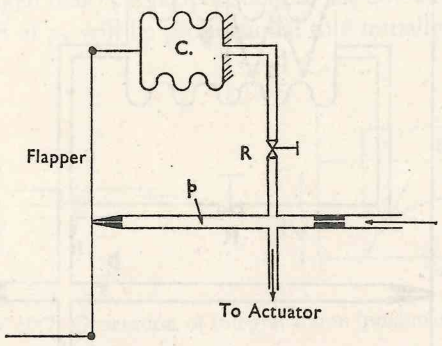


FIG. 10.9. P + D action (pneumatic).

amount proportional to p, to compensate the flapper movement. The sensitivity of the system is increased when the deviation is changing. This increase in sensitivity results in greater controller output when the deviation is changing and this increase in output depends on "rate of change".

It will be obvious that the effect depends not only on the value of resistance R but also on the capacity C of the bellows. Thus the contribution of derivative action to controller output can be increased by increasing R or decreasing C. In practice it is usual to adjust the derivative contribution by adjusting R. As R is increased the time constant CR is increased and the derivative action contribution to controller output is increased.

(j) *Force Balance Controllers*

All the pneumatic controllers described so far are sometimes described as "position balance" controllers, i.e. a movement is derived from the feedback mechanism which balances that derived from the deviation input. They all involve mechanical linkages for measurement of measured value or desired value of the variable. These tend to display the usual disadvantages of friction, backlash, inertia, temperature dependence and non-linearity, which apply to

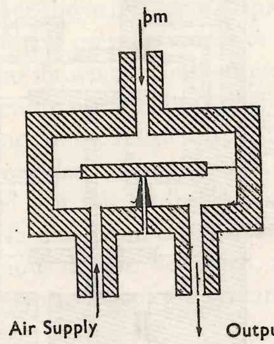


FIG. 10.10. Force balance transmitter (pneumatic).

any system of mechanical linkage. To obviate these disadvantages other types of controllers have been produced using a "force-balance" principle.

With such controllers both the measured value and desired value of the variables are converted into pressures (instead of into mechanical movement). These are then combined in either a bellows or a diaphragm system with the control pressures produced by the methods already described. With such a system no separate mechanisms are required and the mechanical movements in such a system are reduced to a small value (0·001 in. or less).

Figure 10.10 which is a schematic diagram of a 1:1 pneumatic transmitter illustrates the basic principles of a force balance system.

The pressure p_m to be transmitted is applied to a diaphragm which carries the equivalent of the flapper. If the diaphragm exerted no restoring force equilibrium would be set up by the flapper/nozzle system so that the transmitted pressure p would be equal to the applied pressure p_m.

Thus it can be seen that the flapper/nozzle system is used merely as an out-of-balance detector and the transmitted pressure is not

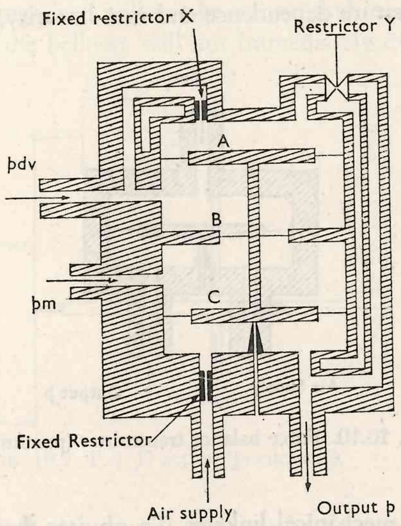


FIG. 10.11. Force balance proportional controller (pneumatic).

dependent upon its characteristics. In practice it is not possible to make the diaphragm restoring force negligible and it is, therefore, necessary to reduce the movement of the diaphragm to a very small value (1×10^{-4} in.).

Figure 10.11 shows the principle of operation of a proportional controller employing force balance.

The measured value of the variable is applied as a pressure p_m and

compared with a pressure p_{dv} proportional to the desired value of the variable.

These two pressures are applied to the enclosures formed by the three diaphragms A, B and C which are rigidly connected together. The difference between pm and p_{dv} is balanced by the output pressure p, controlled by a flapper/nozzle arrangement as previously described. Hence p is proportional to $(p_{dv} - p_m)$ and therefore proportional to the deviation θ.

Proportional band width can be adjusted by means of the restrictor Y which controls the feedback pressure from the output of the controller to a chamber above the upper diaphragm A. This chamber is connected to a constant pressure source through a fixed restrictor X. In the arrangement shown the constant pressure source is that for the desired value pressure p_{dv}. The pressure in the chamber is determined by the resistance of the restrictors X and Y and by the output pressure p of the controller.

It should be noted that this feedback of output pressure p is an example of positive feedback and this increases the sensitivity of the system. In the previous designs described, negative feedback was used to decrease the sensitivity of the system.

(k) *A Commercial Three-term Controller*

Figure 10.12 shows the principle of the Bristol "Free Vane" Controller for combining the signals corresponding to the three actions. This has been selected as an example since it employs the free vane system in place of the more usual flapper/nozzle system but it should be obvious to the reader that the basic principles are the same as those already described.

Proportional action. The nozzle pressure depends on the position of the leading edge of the free vane relative to the axis of the nozzles D, so that nozzle pressure changes proportionately to the movement of the vane just as in the flapper/nozzle system.

The desired value is set by rotating the double-nozzle system on the air-tight swivel joint.

Suppose the controller is in a balanced condition with a 9 psi

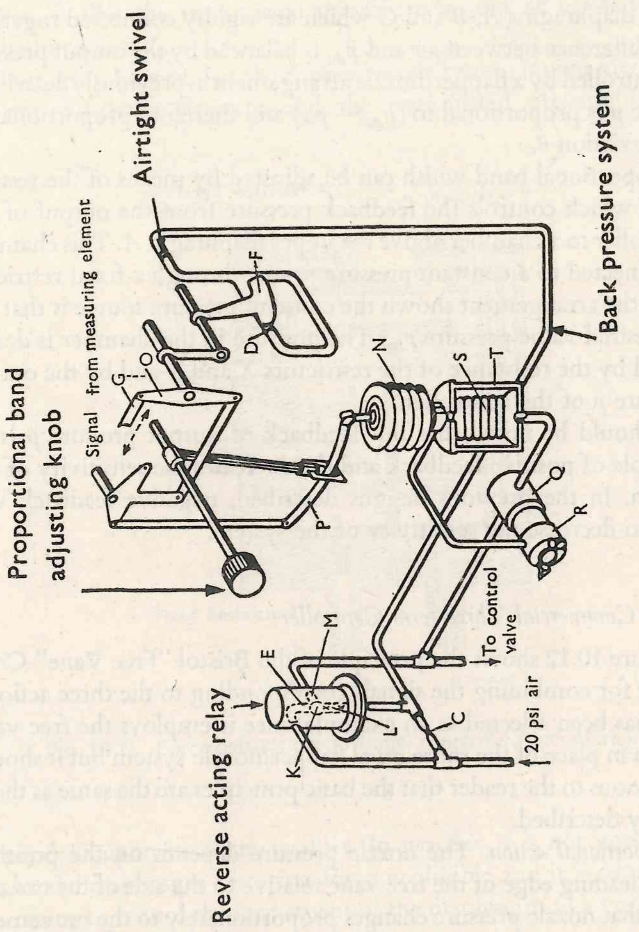


Fig. 10.12. A commercial pneumatic controller (three term).

signal being fed to the final control element. Suppose the value of the controlled medium increases.

This causes link *G* and lever O to move to the right to bring the free vane *F* into the jets *D*. This results in an increase in the back pressure system and, thus, a decrease in output pressure from the reverse acting relay. The drop in output pressure is communicated to the final control element and, at the same time, the feedback bellows *N* contracts and, through the rocker arm *P*, moves the lever O, thus moving the free vane out of the jets. In doing so it retracts part of the original vane movement.

This retraction of the free vane causes a further change in output pressure, initiating another cycle. This continues until equilibrium is reached.

Alteration of proportional band width is achieved by changing the multiplication of the feedback motion between the feedback bellows *N* and the differential lever O by altering the position of the fulcrum on the rocker arm.

Derivative action is supplied by the derivative restriction Q and the derivative tank *T*. If Q is fully open the output pressure from the relay valve port *M* is applied through the open valve Q to the tank and feedback bellows *N*. The controller then acts just in the same manner as a proportional controller.

If Q is partially open and the value of the controller variable increases, the vane moves into the jets, the jet back pressure increases and the relay valve output pressure decreases. As Q is partially closed it offers a restriction to the flow of air from the feedback bellows *N* and the derivative tank *T*, so that the drop in pressure of the output air from the relay valve is not immediately communicated to the feedback bellows. This at first results in no feedback action, the vane is not retracted and, the controller would tend to act as a two-step controller. However, air bleeds slowly through the restrictor Q to equalize pressure on either side of it. This causes the feedback bellows to move downwards and slowly retracts the vane from between the jets.

Eventually the pressure equalizes on either side of Q and the same feedback vane retraction will exist as with the simple proportional control and, until another change in the controlled variable occurs, the derivative system will have no further effect.

Integral action is provided by the integral valve *R* and the integral bellows *S*. If the integral valve *R* is fully open and the feedback system is thus open to atmosphere, then variations in the relay valve output pressure cannot be communicated to the feedback bellows. This is because although the integral bellows will compress and expand with variations in relay valve output pressure, any air displaced by it is dissipated to atmosphere through *R*, which is open; i.e. the feedback bellows remains stationary.

When the integral valve *R* is partially open a resistance to air flow is created into and out of the integral system. Consider an increase in value of the controlled variable moving the free vane into the jets. This causes a drop in outlet pressure from the relay valve, thus allowing the integral bellows to expand downwards. This at first causes a slight vacuum in the integral system causing a downward motion of the feedback bellows thus retracting some of the original vane movement. This slight vacuum is slowly reduced by air bleeding in through the integral valve. This allows the feedback bellows to expand slowly and allow the free vane to move into the jets again.

This vane movement further decreases relay valve output pressure which maintains the vacuum in the integral system. Such a condition continues as long as there is a deviation of the controlled variable above the set point and the output pressure from the relay valve will gradually drop to zero.

The integral action can come to rest only when there is atmospheric pressure in the integral system and, until this condition is reached, air will bleed into or out of the integral system causing a continuous movement of the free vane. When the integral system comes to rest both the integral bellows and the lower end of the differential lever, *O*, return to set and constant positions known as "neutral". At this position the free vane is positioned, in relation to the jets, at a point where the value of the controlled variable coincides with the set point and thus eliminates errors of off-set caused by the proportional and derivative actions.

10.2. ELECTRONIC CONTROLLERS

The obvious advantage of an electronic control system lies in its flexibility and the high speed of signal transmission. Some electronic controllers are designed to compare the measured value and desired value electrically and, in some cases, to generate proportional action electrically, retaining pneumatic systems to generate integral and derivative action. However, all three actions can be generated electrically. Indeed it is probable that as the application of automatic control becomes more precise the need for controllers with other actions will arise and there is no doubt that electronic systems offer the most convenient and flexible means of producing other, and more complex, controller actions when these become necessary.

The electronic controller must transmit a signal which is proportional to the error or deviation between the measured value and the desired value. This output signal will be zero when the measured and desired values are equal and will change in sign according to whether the measured value is greater or less than the desired value.

In a typical system the measured value (the amplified output of a potentiometer) and the desired value (the amplified output of a manually-set potentiometer) are each d.c. currents within the range, say, 0–10 mA. These two signals are compared and the difference between them, i.e. deviation θ, is obtained by passing the two currents in opposition through a common resistor. The voltage across this resistor is, therefore, proportional to deviation.

This voltage is then amplified by an amplifier which must, of course, accept inputs of either polarity and transmit output signals of corresponding polarity.

It is vital that this amplifier should have the smallest possible zero drift because any drift at this point would be seen by the control system as a deviation. This would produce a deviation error in the controlled value, which would not be corrected elsewhere in the control loop. In order to prevent zero drift a chopper-type amplifier (see Chapter 5) is used, where the d.c. input is converted to a.c. by use of a chopper, operating through the use of transistors as "on/off" switches. This type is used, also, since it is important that the chopper itself shall not produce any signal when the true input is zero. By means such as this an electronic controller can be produced in which the zero drift under all normal operating conditions is less than 0·1 per cent of the maximum input signal.

The a.c. signal thus produced is then amplified and rectified to give a d.c. output. With the use of these techniques any drift in the main amplifier does not constitute a zero drift because only the a.c. signals are amplified.

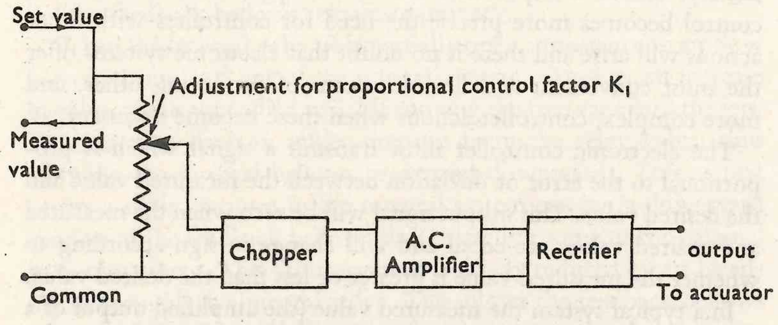


FIG. 10.13. Proportional action (electronic).

(a) *Proportional Action*

Figure 10.13 shows a typical proportional control circuit where the measured and desired values are fed into a resistor.

This resistor takes the form of a gain or sensitivity control potentiometer so that adjustment of the proportional control factor K_1 may be achieved very easily by varying the ratio of the two parts of the potentiometer. By varying this ratio the value of K_1 can be continuously varied from 0·1 to 50 giving a range of proportional band adjustment of 1000 per cent to 2 per cent.

(b) *Integral Action*

The typical method of adding integral action to an electronic controller is shown in Fig. 10.14.

The error (deviation) signal is amplified in the same manner as in the case of proportional action but the output signal from the controller is fed back through a capacitor to the input terminals as a negative feedback action. If the input signal is suddenly changed from zero to some value, the output will change in such a manner that the input current to the amplifier, which is produced by the error signal, is just balanced by the current fed back through the capacitor *C*. This results in the amplifier output changing at a rate

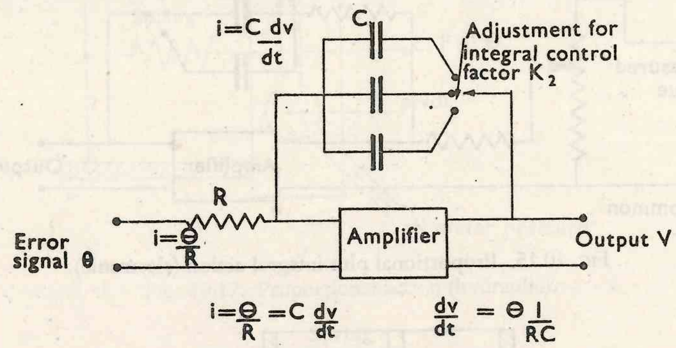


FIG. 10.14. Integral action (electronic).

proportional to deviation of the input signal, i.e. the amplifier output at any moment is proportional to the integral of the input signal. Variations in integral action are obtained by altering the value of capacitor *C*.

(c) *Proportional plus Integral Action*

Figure 10.15 shows, in simple form, the typical method of producing proportional plus integral control in the same controller.

10.3. HYDRAULIC CONTROLLERS

Hydraulically operated controllers are not widely used. This general lack of application is probably explained by the general availability of compressed air and the preference for air leaks rather than oil leaks. However, they have long been used in warships for the positioning of heavy guns and they are popular in the steel industry. This is probably because their chief feature is often regarded as being an ability to produce large working forces. Such a conception, however, ignores the extremely high speed of operation

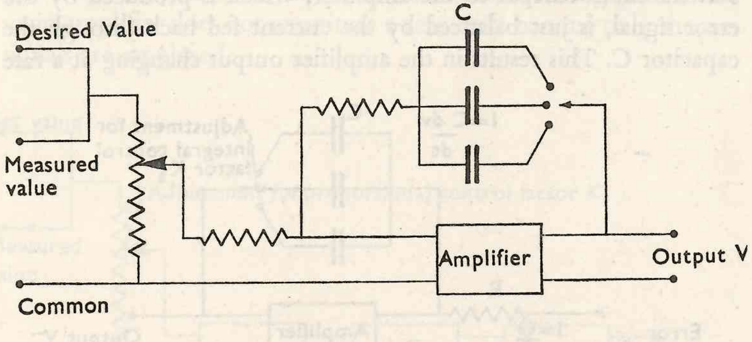


FIG. 10.15. Proportional plus integral action (electronic).

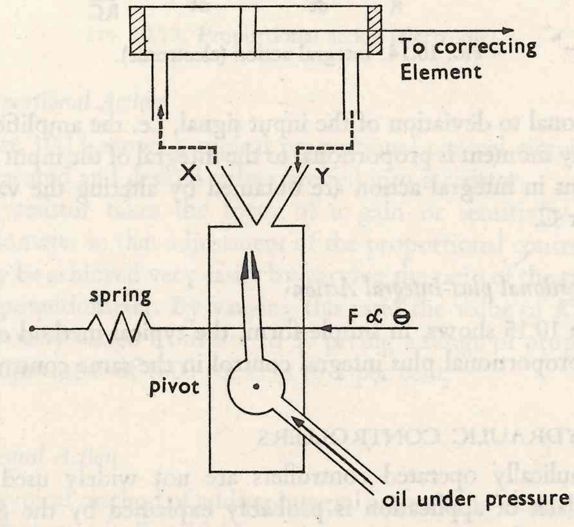


FIG. 10.16. Hydraulic controller.

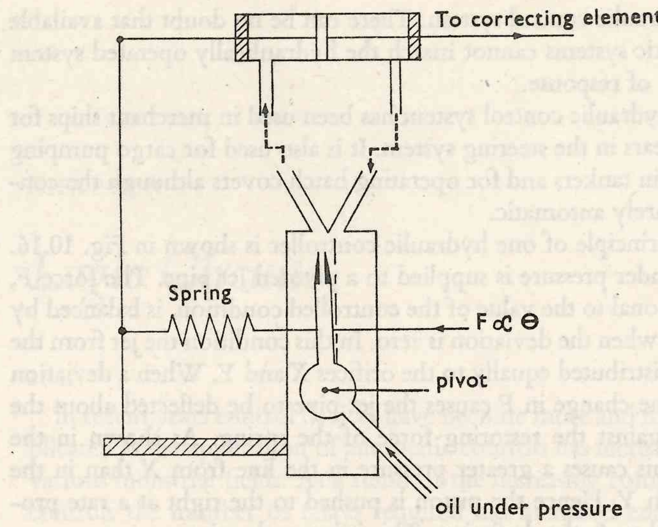


FIG. 10.17. Proportional action (hydraulic).

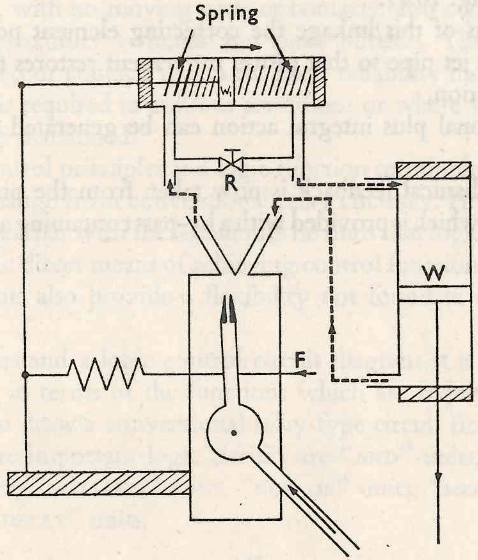


FIG. 10.18. Proportional plus integral action (hydraulic).

of a hydraulic control system. There can be no doubt that available pneumatic systems cannot match the hydraulically operated system in speed of response.

The hydraulic control system has been used in merchant ships for many years in the steering system. It is also used for cargo pumping systems in tankers and for operating hatch covers although the control is rarely automatic.

The principle of one hydraulic controller is shown in Fig. 10.16. Oil under pressure is supplied to a pivoted jet pipe. The force F, proportional to the value of the controlled condition, is balanced by a spring when the deviation is zero. In this condition the jet from the pipe is distributed equally to the orifices X and Y. When a deviation occurs the change in F causes the jet pipe to be deflected about the pivot, against the restoring force of the spring. As shown in the figure this causes a greater pressure in the line from X than in the line from Y. Hence the piston is pushed to the right at a rate proportional to θ, the deviation. This is integral action.

Proportional action is produced by fitting a mechanical linkage as shown in Fig. 10.17.

By means of this linkage the correcting element position is fed back to the jet pipe so that piston movement restores the jet to the central position.

Proportional plus integral action can be generated as shown in Fig. 10.18.

The mechanical feedback is now taken from the piston W_1 the cylinder of which is provided with a by-pass containing a restrictor R.

CHAPTER 11

Logic Circuits

11.1.

In recent years control devices have become more and more complicated as the application of automatic controls has increased in the various industrial fields. As a result of the increasing complexity of controls the number of relays required has increased enormously. This has produced a requirement for a very high operating reliability and long life. The solution to this problem has been provided by the static relay, with no moving parts or contacts: also contactless limit switches, proximity switches and push buttons. The absence of moving parts or contacts makes for great reliability more especially where use is required in arduous conditions or where very frequent operation is demanded.

Static control principles and logic function terminology represent a radical change from conventional relay circuitry. However, once the user is familar with the techniques he finds that logic symbols are a simple and direct means of achieving control functions. Static control elements also provide a flexibility not found in conventional relays.

To understand a logic control circuit diagram it is necessary to think only in terms of the functions which are required. It is not necessary to draw a conventional relay-type circuit first.

The more important logic circuits are "AND" units, "OR" units, "NOT" units, "NOT AND" units, "NOT OR" units, "MEMORY" units, and "TIME DELAY" units.

(a) AND *Unit*

With the AND function all connected inputs to a unit must be live before the output is live. In Fig. 11.1 there is an output when both inputs *A* and *B* are present.

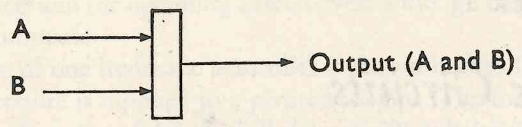


FIG. 11.1. AND unit.

(b) OR *Unit*

The OR function means that at least one input must be live in order to have an output. In Fig. 11.2 there is an output when either input *A* or *B* (or both) are present.

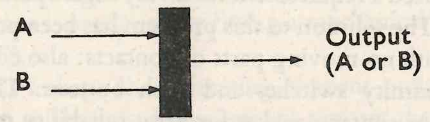


FIG. 11.2. OR unit.

(c) NOT *Unit*

The NOT function, which is often called an "inverter", means that the input to the unit must not be energized in order to have an output. In Fig. 11.3 there is an output only when input *A* is not present. This is equivalent to a "normally closed" contact in conventional relay work.

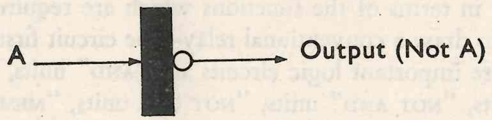


FIG. 11.3. NOT unit.

(d) NAND (*or* NOT AND) *Unit*

When a NOT unit is combined with an AND unit as shown in Fig. 11.4 (a) there is an output only when there is not an input at *A* and not an input at *B*.

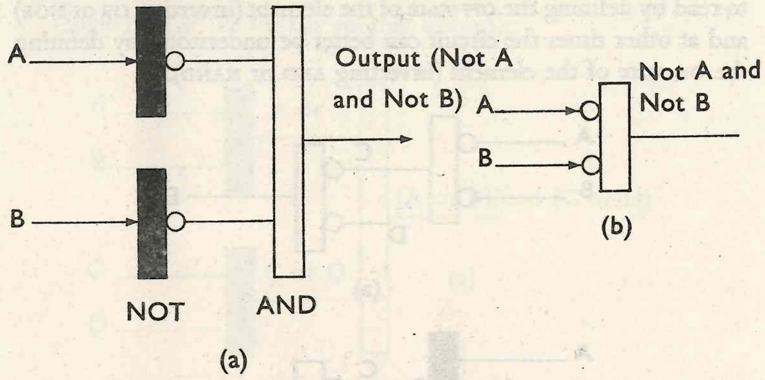


FIG. 11.4. NOT AND unit.

(e) NOR (*or* NOT OR) *Unit*

The logic of Fig. 11.4 can also be expressed in another way. Instead of saying that there is an output when there is not an input at *A* and not an input at *B*, the same function can be expressed by saying there is not an output if there is an input at *A* or an input at *B*. This is shown in Fig. 11.5 (a).

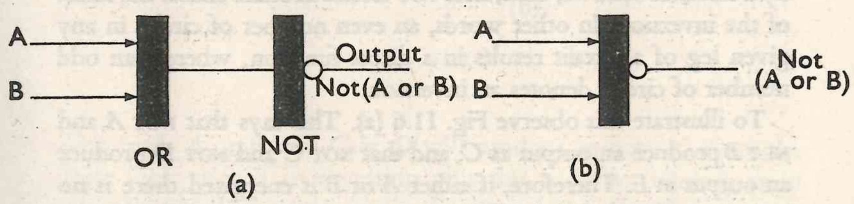


FIG. 11.5. NOR unit.

There are, then, two ways of representing the same logic function. If the inverters (NOT) are combined with the basic AND and OR units the symbols of Fig. 11.4 (b) and 11.5 (b) are produced, which are equivalent. The choice between the two forms is used by the designer for the best approach to the problem. Sometimes a circuit is easier to read by defining the OFF state of the element (inverting OR or NOR) and at other times the circuit can better be understood by defining the ON state of the element (inverting AND or NAND).

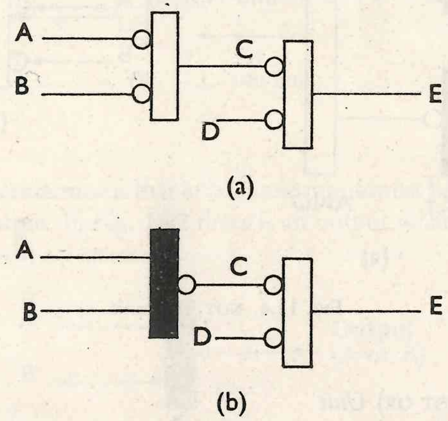


FIG. 11.6.

It must be noted that the symbols used are quite arbitrary since there is no recognized standardization for them yet. For the convenience of the reader an inversion (NOT unit) has been represented by a circle. It follows, then, that two circles in series cancel the effect of the inversion. In other words, an even number of circles in any given leg of a circuit results in a direct function, whereas an odd number of circles denotes an inversion.

To illustrate this observe Fig. 11.6 (a). This says that NOT *A* and NOT *B* produce an output at *C*, and that NOT *C* and NOT *D* produce an output at *E*. Therefore, if either *A* or *B* is energized there is no output at *C* and there will be an output at *E*, provided *D* is not energized.

However, the logic is easier to understand if NAND and NOR symbols are used as shown in Fig. 11.6 (b). By noting that two circles cancel out this can be read directly as *A* or *B* and NOT *D* produce an output at *E*.

Consider Fig. 11.7 (a) as another example.

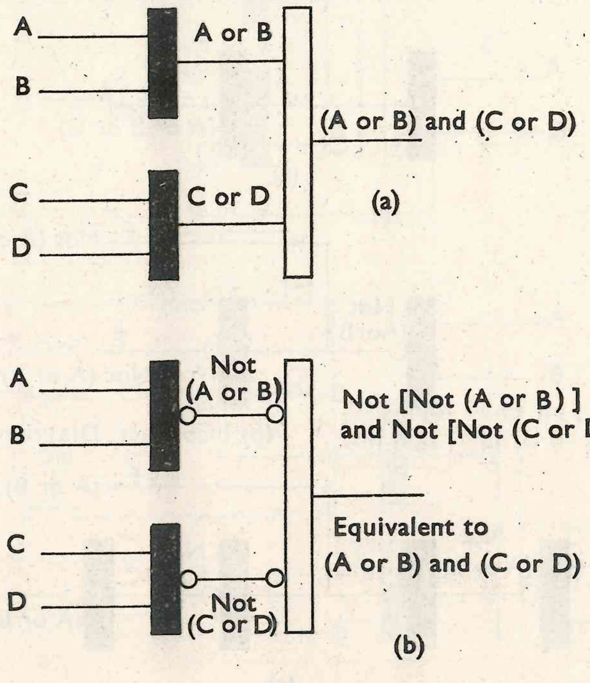


FIG. 11.7.

This reads *A* or *B*, and *C* or *D* will produce an output. Since the basic logic unit is an inverting unit this may be redrawn as in Fig. 11.7 (b) which can be read as *A* or *B*, and *C* or *D* will produce an output, which is the same function as before.

Another example illustrates what happens when OR is followed by OR as in Fig. 11.8 (a).

Here there is an interest in output *A* or *B* and also in *A* or *B* or *C*, otherwise a simple three input OR unit would have been used. When redrawing Fig. 11.8 (a) to use the NOR symbol, the first impulse is to show it as Fig. 11.8 (b) which is incorrect since the outputs do not

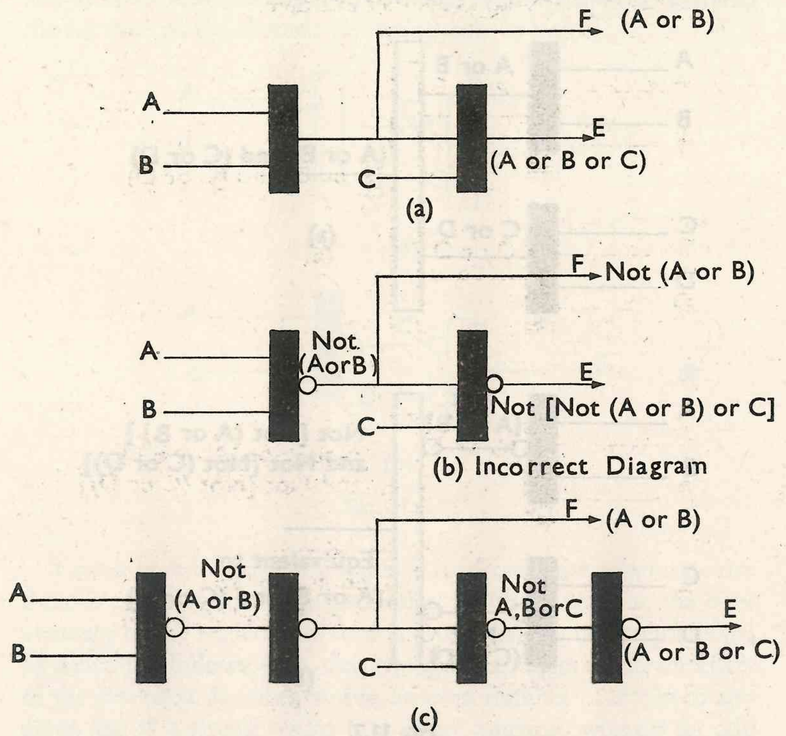


FIG. 11.8.

correspond with those of Fig. 11.8 (a). If the effect of *A* and *B* on output *E* is considered, then the two inverting circles cancel one another. However, when input *C* is considered the output is inverted from what it should be. This emphasizes the importance of checking all inputs to ensure that the correct number of inversions are carried out.

Figure 11.8 (c) shows the diagram as corrected by the addition of two inverters. This now reads *A* or *B* will produce an output at *F*, and *A* or *B* or *C* will produce an output at *E*, which is correct.

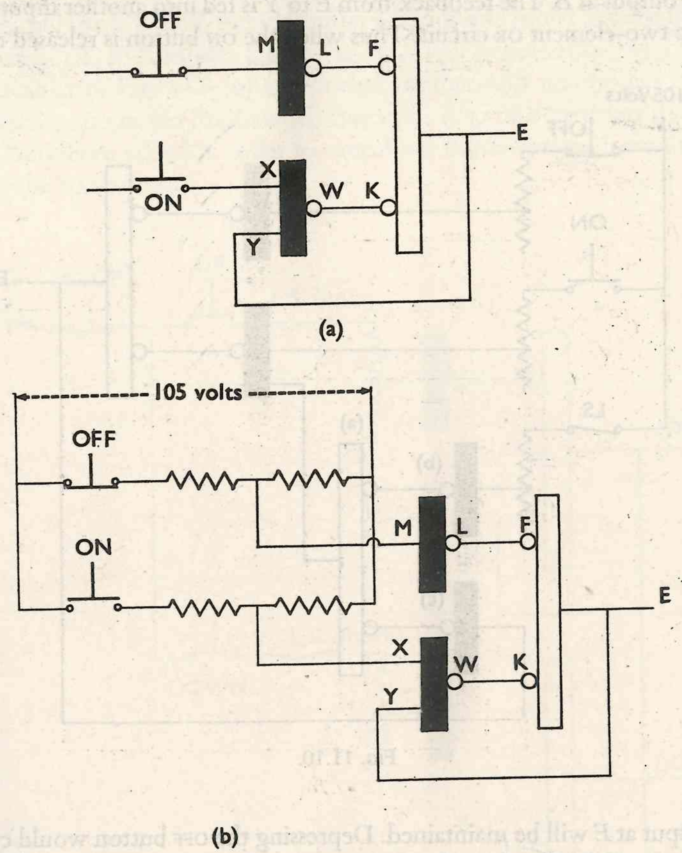


FIG. 11.9.

Consider next the simple circuit shown in Fig. 11.9 (a).

In logic terms it reads: "if the OFF signal is not interrupted and the ON signal or the *E* signal (interlock or feedback signal) is energized there will be an output at *E*."

When the ON button is pressed and the OFF button is not pressed, there will be signals at *M* and *X*. Since the circles in lines *L–F* and in *W–K* cancel, the AND requirements have been satisfied to produce an output at *E*. The feedback from *E* to *Y* is fed into another input of the two-element OR circuit. Thus when the ON button is released the

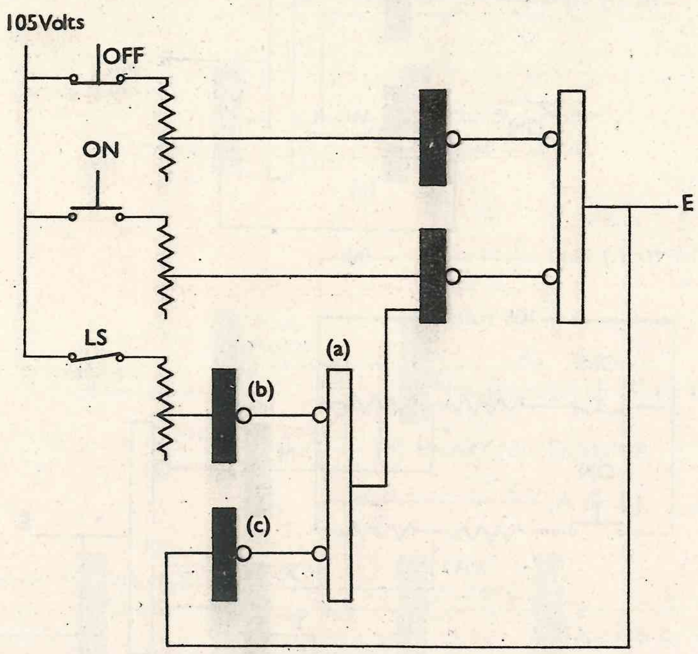


FIG. 11.10.

output at *E* will be maintained. Depressing the OFF button would cut off one of the signals in the AND circuit and remove the output at *E*.

The circuit shown in Fig. 11.9 (a) is over simplified in that it is necessary to use a high input signal at *K*, or to otherwise make sure that the AND function operates faster than the OR function. Also, the logic elements require a 6 to 18 volt signal, whereas push buttons or switches would be operated at about 105 volts (for reliable contact

operation) and hence cannot be connected direct to the logic elements. The connections would, therefore, be made through voltage dividers as shown in Fig. 11.9 (b).

A further example is shown in Fig. 11.10.

With this circuit one may start and continue to operate by holding the ON button closed, but it requires a permissive signal like the limit switch (*LS*) to complete the operation. It will be noted that Fig. 11.10 is similar to Fig. 11.9 (b) except that another AND unit (a) has been added to tie in the limit switch function. Also two inverters (b and c) have been added in order to cancel out the inversions introduced by the NAND unit.

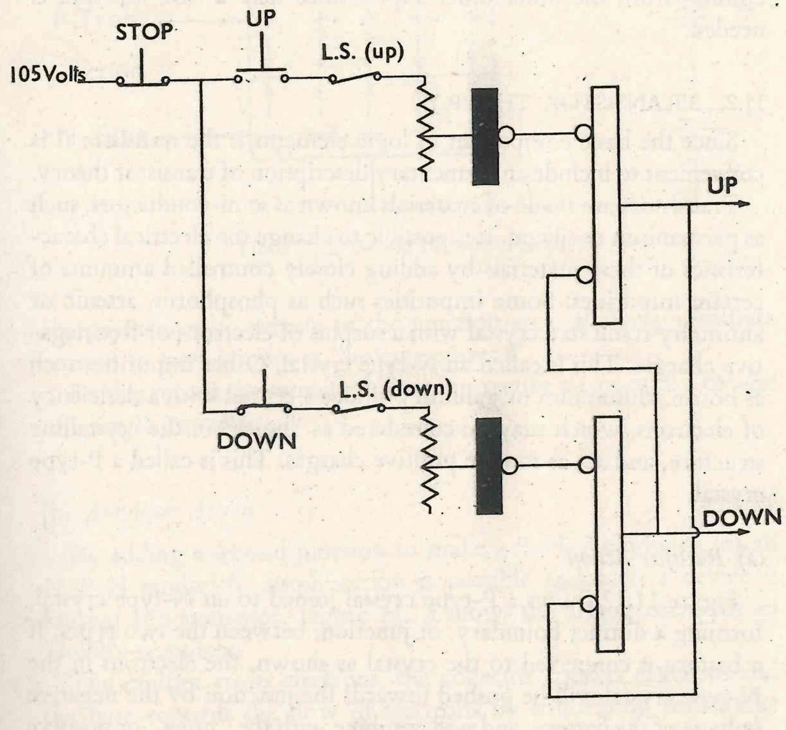


FIG. 11.11.

Another example, shown in Fig. 11.11, can be developed by starting out with the following requirements:

(a) The STOP button shall override all other signals and prevent operation.
(b) There shall be an UP output signal as long as:
 (i) the UP button is held closed, and
 (ii) the UP, end of travel limit switch, *LSU*, is closed, and
 (iii) the DOWN button is not closed.
(c) Similar requirements for DOWN operation.

In this example note that inverters are necessary in the UP and DOWN inputs to the AND units, but they are not required in the inputs coming from the other units' input, since here a NOT function is needed.

11.2. TRANSISTOR THEORY

Since the basic component of logic elements is the transistor it is convenient to include an elementary description of transistor theory.

Transistors are made of materials known as semi-conductors, such as germanium or silicon. It is possible to change the electrical characteristics of these materials by adding closely controlled amounts of certain impurities. Some impurities such as phosphorus, arsenic or antimony result in a crystal with a surplus of electrons or free negative charges. This is called an N-type crystal. Other impurities such as boron, aluminium or gallium produce a crystal with a deficiency of electrons, which may be considered as "holes" in the crystalline structure, and act as mobile positive charges. This is called a P-type crystal.

(a) *Rectifier Action*

Figure 11.12 shows a P-type crystal joined to an N-type crystal, forming a distinct boundary, or junction, between the two types. If a battery is connected to the crystal as shown, the electrons in the N-type crystal will be pushed towards the junction by the negative voltage of the battery, and will combine with the "holes" or positive

charges attracted to the junction. Electrons from the battery constantly enter the crystal at the N terminal to replenish those that have combined with "holes", and electrons leave the P terminal to replenish the "hole" supply of P, resulting in current flow.

If the polarity of the battery is reversed, the positive and negative particles are drawn away from the junction, leaving the section of the crystal near the junction practically void of charge carriers and the crystal effectively blocks the flow of current. A few random

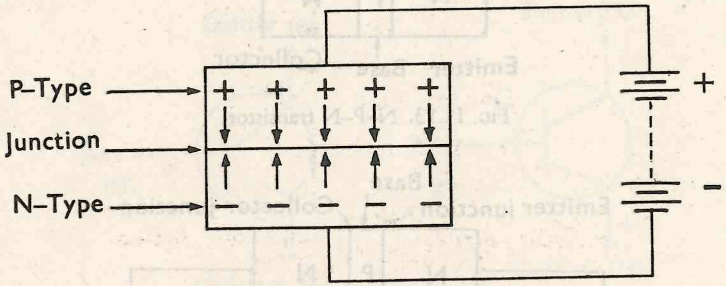


FIG. 11.12. P- and N-type crystals.

charge carriers do remain in the junction area, allowing a minute current to pass, known as "leakage current".

By this means the semiconductor can rectify a current by the use of a single junction.

(b) *Amplifier Action*

By adding a second junction to make a P–N–P sandwich, or an N–P–N sandwich, amplification is possible and such a device is known as a transistor. Figure 11.13 shows the arrangement for an N–P–N transistor.

The emitter emits electrons, the collector collects electrons and the base controls the flow of electrons by controlling the charge concentration in the base region.

Consider the circuit of Fig. 11.14.

The emitter junction will pass current easily because it has a forward bias. The collector junction, however, will not pass current from the collector to the base because this junction is back biased. However, it is found that when a small positive voltage is applied to the base relative to the emitter, many of the electrons drawn from

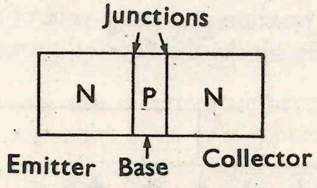


Fig. 11.13. N–P–N transistor.

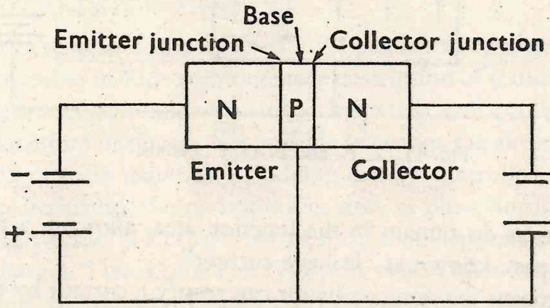


Fig. 11.14. Action of N–P–N transistor.

the emitter diffuse through the very thin base region into the collector without combining with "holes" in the base. As the base is made more positive, more electrons are drawn out of the emitter and are made available for diffusion into the collector. Thus a relatively large collector-to-emitter current is controlled by a small base-to-emitter voltage and current.

The electrons that enter the base, but do not reach the collector, combine with "holes" in the base and contribute to the base current, reducing the gain of the transistor. To reduce the base current, the base is kept as thin as possible, usually less than 0·001 in., and the

"hole" content kept to a minimum by using fairly high purity material. In other words, the base material is only slightly P-type material.

Similar reasoning can be applied to a P–N–P type of transistor by reversing the polarities of the batteries.

The symbols for the two types of transistor are shown in Fig. 11.15.

It should be noted that the emitter is designated by having an arrow, and the distinction between P–N–P and N–P–N units is made by the direction of the arrow. This convention agrees with that

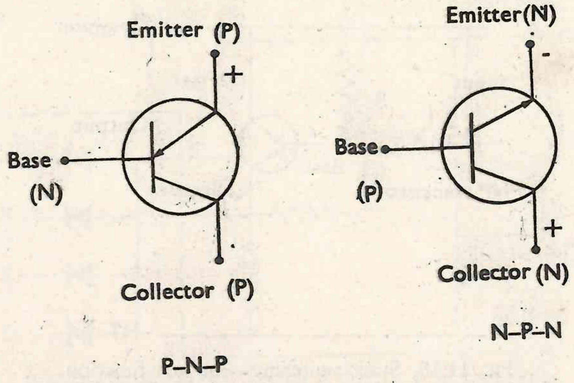


Fig. 11.15. Symbols for P–N–P and N–P–N transistors.

used for rectifiers, in that the arrow indicates the forward current or easy direction of conventional current flow.

The ratio of collector current to base current is called "current gain" or "beta". It is desirable to have the current gain as high as possible and gains of 50 to 100 are quite common in commercially available transistors. Since very little voltage (0·1 to 0·5 volt) is needed to cause appreciable emitter to base current, the input power is very low. Beta times the base current will flow in the collector circuit, and the voltage can be as high as 120 volts for some types of transistors. Therefore, a relatively large amount of power can be controlled in an external load, and the power gain of a transistor (power out/power in) can be as high as 40,000 in some applications.

(c) NOT *Element*

Consider next a static switching circuit using a P–N–P transistor as shown in Fig. 11.16.

When there is no input signal, the base is biased slightly positive with respect to the emitter as it is connected through R_1 to the positive supply. Hence no current will flow in the transistor. In other words, the transistor is cut off. Under these conditions the transistor represents a very high resistance between emitter and

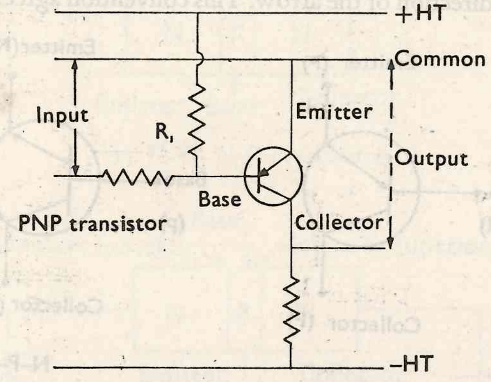


FIG. 11.16. Static switching—the NOT function.

collector, and the output voltage is essentially equal to the negative supply voltage.

When an input signal of sufficient magnitude is applied, the base becomes negative with respect to the emitter, and current flows from emitter to base, which permits collector current to flow. As the level of the input signal is increased, the base current increases, and the collector current follows at a level of about 50 times the value of the base current. This proportional amplification is used in analogue amplifiers.

For static switching applications, however, a bi-stable amplifier is required, with only two conditions, completely cut off or fully saturated. This is accomplished by proper selection of signal voltages, supply voltages and resistor values. If, say, a minus 6 volt signal is

applied at the input terminal, sufficient base current is drawn to fully saturate the transistor. Since the voltage drop (emitter to collector) across the saturated transistor is very low, the output voltage will be essentially zero.

When the input signal is removed, the base again becomes positive with respect to the emitter, and the transistor is cut off. The output voltage then goes to, essentially, the value of the negative supply voltage.

It should now be recognized that the action described is that of the NOT logic function, since there is an output only when there is no

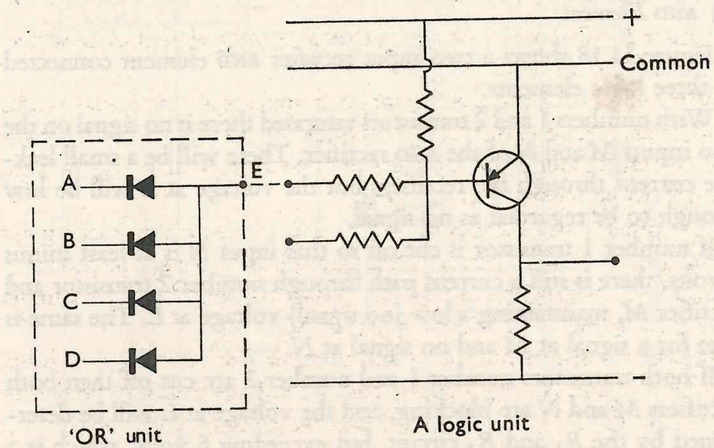


FIG. 11.17. OR function.

input signal. The circuit shown in Fig. 11.16 is the basic circuit used in universal logic elements, and is the most widely used logic element. The NOT function is performed when only one input is used. With multiple inputs combinations of these circuits can also perform NAND, NOR and permanent memory functions.

(d) OR *Element*

Figure 11.17 shows a four-input OR element connected to a transistor logic element.

With zero voltage on *A*, *B*, *C* and *D* with respect to the common busbar there will be zero voltage at output *E*.

If there is a minus 6 volt signal on *A*, *E* will be minus 6 volts. Similarly if there is a signal on *B*, *C* or *D* (or any combination), there will be an output at *E*. This is OR logic.

Inputs *A*, *B*, *C* and *D* are isolated from each other, as one rectifier is always blocking between any two signals.

The output can then be used to drive a further logic element.

(e) AND *Element*

Figure 11.18 shows a two-input rectifier AND element connected to three logic elements.

With numbers 1 and 2 transistors saturated there is no signal on the two inputs *M* and *N* of the AND rectifier. There will be a small leakage current through the rectifiers but the voltage at *L* will be low enough to be regarded as no signal.

If number 1 transistor is cut off so that input *N* is at least minus 6 volts, there is still a current path through number 2 transistor and rectifier *M*, maintaining a low (no signal) voltage at *L*. The same is true for a signal at *M* and no signal at *N*.

If both transistors number 1 and number 2 are cut off then both rectifiers *M* and *N* are blocking, and the voltage at *L* will be determined by the R_1 and R_2 circuit, but exceeding 6 volts, which is a signal.

Thus the only combination of inputs which produces an output is both inputs *M* and *N*. This is AND logic.

(f) MEMORY *Element*

Figure 11.19 shows a memory circuit in simplified form.

This is a memory logic element that returns to the state it was in, before loss of supply, after the supply has been restored.

The memory function is carried out by the iron-cored reactor, which is saturated in one direction or the other, depending on the state of the element at the time power was lost.

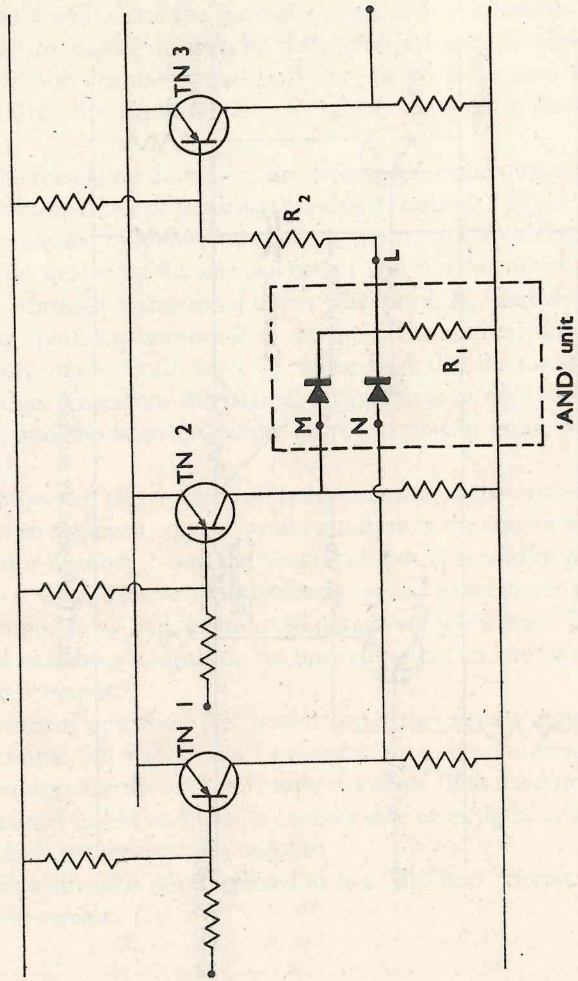


FIG. 11.18. AND function.

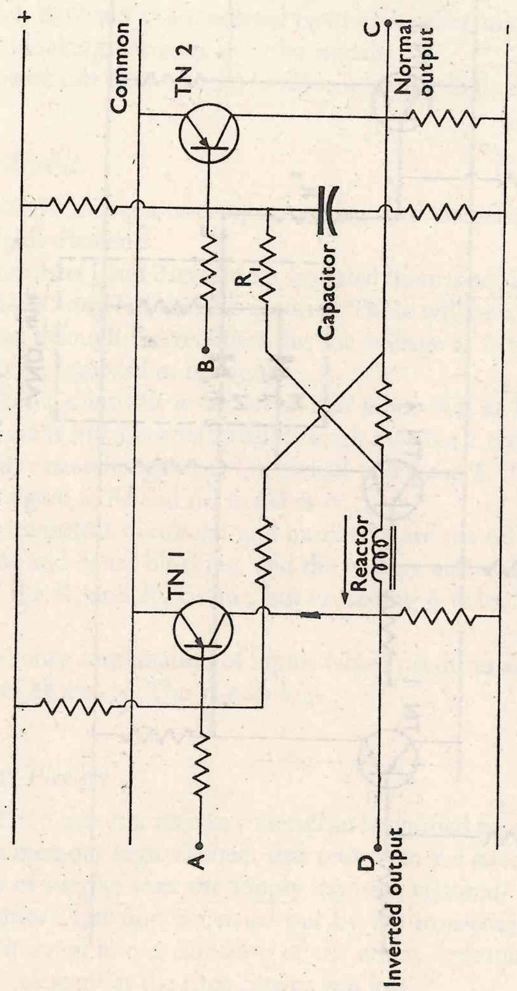


FIG. 11.19. MEMORY function.

When power is restored and with no signals on A or B the capacitor will start charging and will momentarily saturate transistor number 2 and make the normal output zero. In, say, 0·25 millisecond, the capacitor will be fully charged and, if nothing else happens, the normal output will start to go from zero to minus 6 volts, as the capacitor is no longer drawing transistor base current.

In the meantime, however, the zero volt normal output has been applied to an input of transistor number 1, causing a signal to appear at the inverted output terminal D. If the reactor is saturated in the direction shown by the arrow, current will flow from the common busbar through transistor number 2, resistor R_1, and the reactor, causing transistor number 2 to saturate. This current flows within the short time interval, say 0·25 millisecond, that the capacitor takes to charge. Therefore the normal output stays at zero volts, or no signal, and the inverted output stays at minus 6 volts, which is a signal.

If, however, the reactor had been saturated in the opposite direction, then sufficient current would not flow in the reactor to saturate transistor number 2, and the normal output C would have gone to minus 6 volts. This would have been applied to transistor number 1 as an input signal and the inverted output would be zero. Thus there would have been a signal on the normal output and no signal on the inverted output.

In normal operation (i.e. power ON) a momentary signal on the set terminal, (A) will produce a signal at the normal output terminal and no signal at the inverted output terminal. This condition will be maintained unless and until a momentary reset signal is applied at (B) which will reverse the outputs.

Such a circuit is often referred to as a "flip-flop" circuit or multivibrator circuit.

(g) TIME DELAY *Element*

Figure 11.20 shows a circuit which will produce a time delay. Assume that initially there is zero voltage at input A, i.e. A is

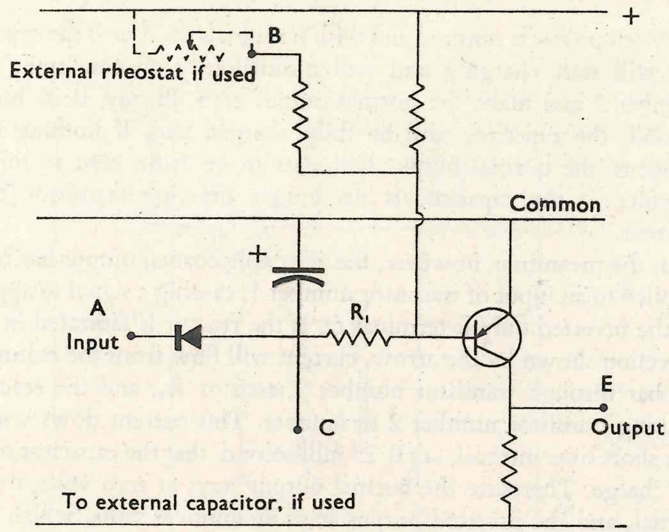


FIG. 11.20. TIME DELAY function.

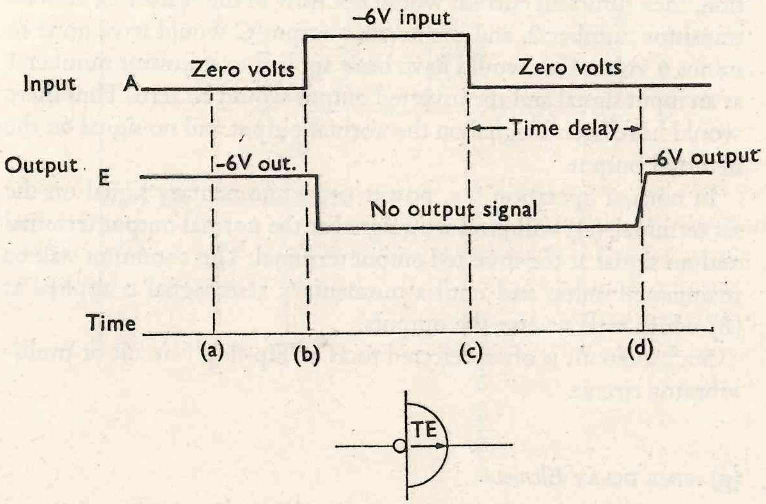
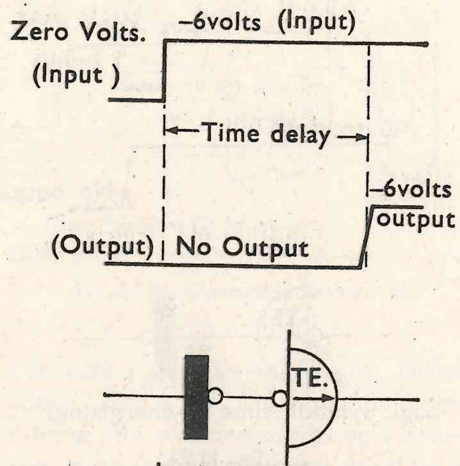


FIG. 11.21.

shorted to the common busbar. There is no voltage across the capacitor and the transistor is cut off. Hence there is an output at *E*.

Now if a negative signal (minus 6 volts) is applied to *A*, the capacitor will commence charging. In, say, 1 millisecond, the voltage from emitter to base is sufficient to saturate the transistor and output *E* goes to zero.



Logic symbol
(time energizing)

FIG. 11.22.

If the input signal is now removed from *A* (short *A* to the common busbar), the capacitor cannot discharge through the input circuit because the rectifier at *A* is blocking. It discharges through the transistor emitter circuit and resistor R_1. As the transistor will remain saturated until the capacitor voltage decays to a low figure (say less than 1 volt), it follows that there is a time delay before the transistor cuts off and a signal reappears at *E*.

By connecting point *B* to the positive busbar, the timing can be reduced. Variations in the reduction of time delay can be obtained by means of an external rheostat at *B*.

Longer time delays can be obtained by adding an external capacitor between point *C* and the common busbar.

Figure 11.21 shows a sketch of signal voltages plotted against time.

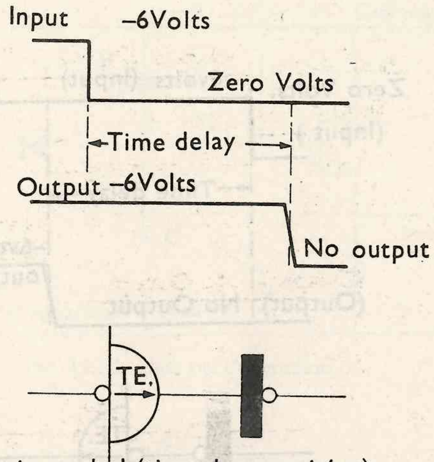


FIG. 11.23.

Time (*a*) represents the assumed initial conditions with zero volts on the input and the transistor cut off. At time (*b*) a signal is applied to input *A*, and at (*c*) this signal is removed, starting the timing cycle. The time delay is measured from (*c*) to (*d*), or from the time the input goes to zero to the time the output signal reappears.

The output is thus time energizing, but timing starts from the removal of the input signal. By adding an inverter a true time energizing device is obtained as shown in Fig. 11.22

Time de-energizing is obtained by inverting the output as shown in Fig. 11.23.

11.3. CONTACTLESS PROXIMITY LIMIT SWITCH

This switch replaces the conventional limit switch which is operated by physical contact, e.g. by a dog or a cam.

The switch has a fixed sensing element *B* and a sensed element *A* mounted on the moving part as shown in Fig. 11.24.

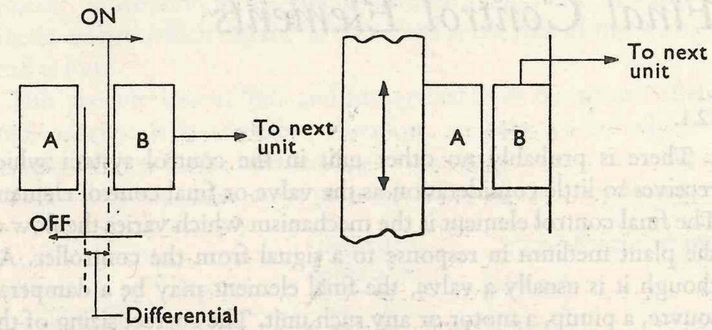


FIG. 11.24. Contactless limit switch.

Element *A* contains a small moving magnet. Element *B* contains a stationary "potted" core or coil. Upon attaining a pre-set proximity (say, 0·25 or 0·5 in.) the two elements set up a ferro-resonant effect which provides a snap action ON response signal for input to a logic element. Thus switching is effected without either contacts or mechanisms.

Contactless push buttons are similar to contactless proximity or limit switches, the details of operation being identical.

CHAPTER 12

Final Control Elements

12.1.

There is probably no other unit in the control system which receives so little consideration as the valve or final control element. The final control element is the mechanism which varies the flow of the plant medium in response to a signal from the controller. Although it is usually a valve, the final element may be a damper, a louvre, a pump, a motor or any such unit. The correct sizing of this final control element is vital. If it is a valve the choice of inner valve characteristic has to be determined by the plant characteristics and the size by the load variation to be catered for. This is critical, otherwise the full valve travel cannot be exploited and precise positioning with stability cannot be achieved. Sizing merely to line size will not achieve success, since the control valve is a variable area orifice and is not at all like the conventional valve designed for simple shut-off service. With controllers of the continuous type it is very important that smooth and even changes are made in the flow to the process and this depends to a great extent on the flow characteristic of the final control element.

It is often assumed that the flow of liquids is smooth without turbulence but this condition is rarely encountered in practice because irregularities inside the valve body and in the piping system break up the streamline flow. Turbulent flow causes an increased pressure loss because of the energy lost in turbulence. The flow, therefore, decreases proportionately as the turbulence due to higher velocity increases.

The pressure loss through the lines leading to and from the valve must be taken into account, for it must be subtracted from the total pressure differential available in order to obtain the actual pressure differential at the valve. The effect of non-uniformities in the line such as fittings, valves and bends is to cause a pressure drop since the velocity distribution in the pipe is disturbed. The result is a loss of kinetic energy which cannot be recovered and a part of the pressure head is lost.

This pressure loss in lines and fittings varies as the square of the fluid velocity. It is advisable, therefore, to have a relatively low velocity of flow in the lines leading to and from the control valve in order to keep the variation in pressure differential at a minimum.

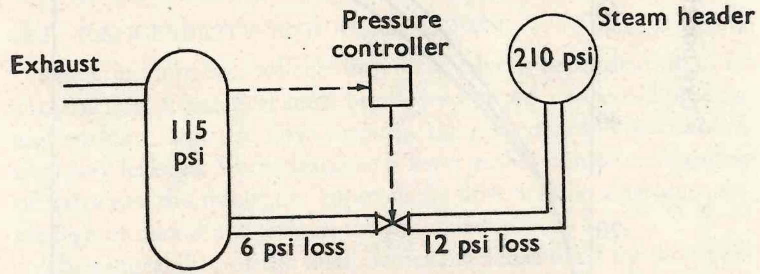


FIG. 12.1. Pressure differential at a control valve.

When a valve, or other final control element, is installed a pressure differential must exist across the element in order to obtain a flow. This pressure differential is not an arbitrarily selected value but depends upon the arrangement of the piping system.

Suppose the pressure controller in Fig. 12.1 is holding the pressure at the vessel constant at 115 psi by controlling the flow of steam through a control valve. Assume that the header pressure is 210 psi.

Also suppose that the line losses from header to valve are 12 psi and that the losses from valve to vessel are 6 psi.

The upstream pressure at the valve will be $(210 - 12) = 198$ psi.
The downstream pressure at the valve will be $(115 + 6) = 121$ psi.

The pressure differential at the valve is not any preselected value but must be $(198 - 121) = 77$ psi.

The total pressure differential available includes both the line losses and the pressure differential at the valve. If the line losses are large the pressure differential across the valve must be small, by comparison. For example, it would be possible to increase the velocity of flow to such an extent that the lines and fittings would

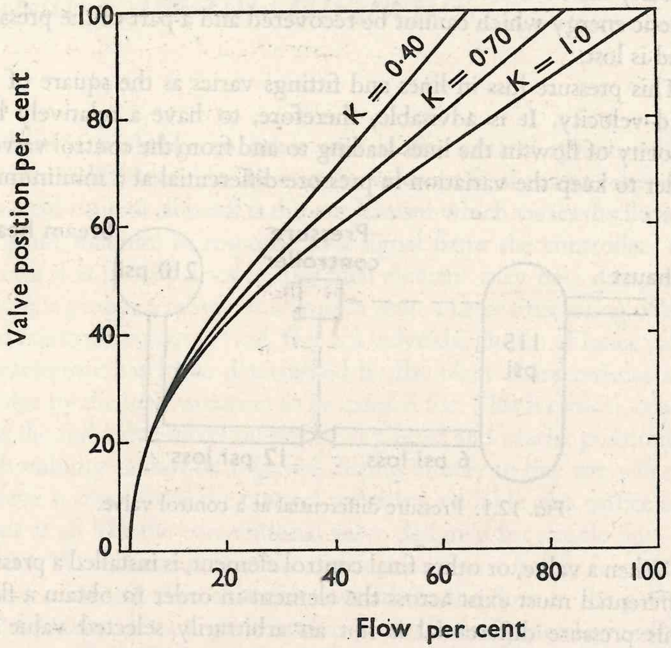


FIG. 12.2. Effect of variation in line losses.

absorb all the available pressure differential. The valve would then be completely ineffective in controlling the flow.

Such loss of control is shown in Fig. 12.2.

The curve $K = 1$ represents the flow characteristic of the valve. This flow characteristic is obtained by setting the valve at various openings and measuring the flow. It is representative of the action of the control system, since the controller regulates the valve opening, thereby adjusting the flow to the process.

The curves shown in Fig. 12.2 represent the flow characteristic when K is the ratio of the pressure differential existing at the valve to the total differential available. The maximum flow through the valve is seriously affected by increases in line velocity. Notice that if the velocity reaches a point where only 40 per cent of the differential exists at the valve ($K = 0{\cdot}40$) the valve can pass only about 63 per cent of its maximum flow. Line losses greater than 50 per cent of the available differential would practically put the controller out of operation because the normally required flow could not be obtained.

Consequently line losses must be minimized in order to provide the maximum differential at the valve. The maximum flow as well as the flow characteristic can then be maintained and the final element can provide effective control of the flow.

12.2. RANGEABILITY AND TURN DOWN

The minimum controllable flow of a valve is dependent upon its construction. Clearances must be allowed in order to avoid binding and sticking, and the flow through these clearances constitutes a necessary leakage. Since clearance is more or less constant regardless of valve size the minimum controllable flow will be a greater percentage of maximum flow in smaller valves.

The rangeability of the final element is determined by the minimum controllable flow. It is defined as the number of times the minimum flow may be increased before maximum flow is obtained, or as the ratio of maximum to minimum controllable flow.

For instance, manœuvring valves must have a high rangeability since a marine power plant is required to perform over a very wide load range, e.g. from stop to full ahead.

The turndown of the final element is also determined by the minimum controllable flow but is based on the normal maximum instead of the maximum flow. It is defined as the number of times the minimum flow may be increased before normal maximum flow is obtained, or as the ratio of normal maximum to minimum controllable flow. If the normal maximum flow through a valve is 70 per cent and the minimum controllable flow is $3{\cdot}5$ per cent, the turndown is $70/3{\cdot}5 = 20$. Turndown therefore depends to a great extent upon the valve size but equally it depends on whether a large or small flow is required, as a normal maximum, i.e. it is dependent on plant design.

The size of the final control element is important in the operation of the control system because of its effect on rangeability and flow characteristic. Before selecting valve size it is necessary to know the characteristics of the plant medium, the pressure differential across the valve and the flow required by the process under control. Any of these three factors may vary, and considerable experience is required in estimating their magnitude. It is normal to select the size of valve which will pass the normal operating flow at about 70 per cent of maximum flow.

If the valve, or other control element, is oversize it will pass the normally required flow at some lower setting, e.g. an oversize valve may pass the normal maximum flow at 35 per cent of its maximum flow. The minimum controllable flow may be 2 per cent and this represents a greater proportion of the normal flow. The turndown is then only 17 instead of 35. In addition the valve is operating around the lower portion of the flow characteristic. Oversizing is undesirable because in any final control element the low flow part of the characteristic is the most irregular.

If the valve is undersize it will pass the normally required flow at a high setting, e.g. at 90 per cent of its maximum setting. In the normal operation of the control system the valve is moved both above and below its normal flow position. The valve may then reach its open position without satisfying the demands of the controller if the valve is undersize.

The shape of the inner valve determines the valve flow characteristics and there are many varieties of inner valve such as parabolic, V-port, bevel plug and rectangular port. The characteristics of such valves can be obtained from the valve manufacturer.

Thus before a valve can be selected considerable technical data must be obtained including maximum, normal and minimum flow, acceptable pressure drop and static pressure if satisfactory control is to be achieved.

12.3. VALVE POSITIONER

The occasional necessity for valve positioners has already been mentioned in Chapter 7. They are used in conjunction with the control valve to ensure accurate and dependable positioning of the valve. Some of the applications where a positioner would be required are:

(i) single port valves handling high-pressure drops,
(ii) installations involving high pressures which require tight valve stem packing,
(iii) valves handling viscous fluids,

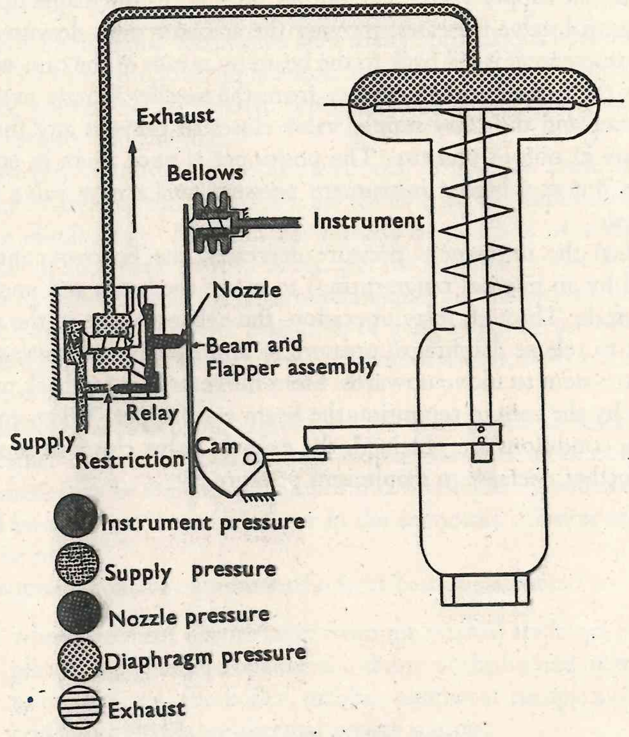


FIG. 12.3. Valve positioner.

(iv) installations where the control valve is located a great distance from the controller.

Figure 12.3 shows a typical valve positioner.

Operating medium air pressure is supplied to the relay supply valve and fixed restriction. The diameter of the fixed restriction is less than the diameter of the nozzle so that air can bleed out faster than it is being supplied when the flapper is not restricting the nozzle.

When the instrument pressure increases the bellows expands to move the beam, causing the flapper to restrict the nozzle. The nozzle pressure increases and moves the relay diaphragm assembly to open the supply valve. The output pressure to the diaphragm of the control valve increases, moving the actuator stem downwards. Stem movement is fed back to the beam by means of the cam which causes the flapper to move away from the nozzle. Nozzle pressure decreases and the relay supply valve closes to prevent any further increase in output pressure. The positioner is once again in equilibrium but at a higher instrument pressure and a new valve plug position.

When the instrument pressure decreases, the bellows contracts (aided by an internal range spring) to move the beam and uncover the nozzle. Through relay operation, the exhaust valve in the relay opens to release diaphragm pressure to atmosphere, permitting the actuator stem to move upwards. Stem movement is fed back to the beam by the cam to reposition the beam and flapper. When equilibrium conditions are obtained, the exhaust valve closes to prevent any further decrease in diaphragm pressure.

CHAPTER 13

Applications of Control Engineering

CONTROL engineering techniques find many applications in modern ships. The more obvious applications are to the machinery for propulsion, generation of electric power, refrigeration, pumping of liquid cargoes and the like.

However, it must not be assumed that it is only these large machinery areas of the ship which are suitable for the application of control engineering. Mention has already been made of a classic example of automatic control at sea, viz. the automatic pilot. Other areas which are suitable for control techniques are the power operation of hatch covers, power operation of gangways, the use of self-tensioning mooring winches, domestic water systems, air-conditioning plant and many others.

Whichever area is being considered the basic appraisal described in Chapter 3 must be carried out. Automatic or remote control equipment may be expensive to install and the frequency with which it will be used can be a vital factor in the economic decision which must be made.

Automatic control equipment finds its best applications:

(i) where transient disturbances from the normal are large, complicated and perhaps beyond the ability of the human operator to control, e.g. the boiler, turbine, alternator complex of the critically rated large thermal power station;

(ii) where the process is steady running and/or repetitive.

A process pattern where disturbances from the normal occur only infrequently and where they are not complicated is not suited economically to full automatic control.

For instance, automatic control of the main propulsion machinery is often fitted. The large modern cargo vessel or tanker has long steady running periods from pilot to pilot with manœuvring carried out at the terminal ports. The tug, the ferry and the coaster all have process patterns with very frequent manœuvring. Obviously there is a better case for fully automatic control of the propulsion machinery (i.e. control of both steady running and manœuvring conditions) in these latter vessels than in the former. Nevertheless, for the deep sea vessel automatic control of the propulsion machinery may be very appropriate for the steady running portions of the voyage since this represents the major portion of the vessel's life. This means that manual control would be used for manœuvring.

Once the decision has been made to apply controls to an area of the machinery it must be borne in mind that the provision of remote or automatic control often involves dispensing with ideas and arrangements previously adopted for local manual control of the machinery. Whereas local manual control has been effected in the past by means of rods and mechanical linkages, this method is only rarely the most desirable or economical way of carrying out a remote-control installation. For bridge controls it is only in the smallest vessels that rod controls can be considered as satisfactory.

For all remote-control systems it is desirable and for bridge control it is essential that the fundamental problems are pin-pointed and the best system produced to solve these problems.

For remote control the equipment must give reliable and quick response to the operator's commands for long service periods. There is, however, no need to go beyond the limits of accuracy already existing as good practice if, by doing so, the reliability of the equipment is reduced in any way. For all remote-control systems reliability in service is more important than any other single factor.

It must be remembered that it is very difficult to arrange "fail safe" conditions under all circumstances. Circumstances can be envisaged in which any failure of bridge controls may imperil the ship and its crew or passengers and it should be clearly understood that equipment which is satisfactory by industrial standards ashore will not necessarily be suitable for marine application.

By making an analysis of the fundamental operating requirements, including interlocks, that are required on the main machinery rationalization of design can produce simple and reliable circuits for remote engine controls. As is frequent marine practice, these should be interlocked with the telegraph equipment so that the engine cannot run in a direction contrary to the orders from the bridge. One way of achieving this is by placing an interlock valve behind the reply lever handle—the warning system will then continue ringing until the telegraph has been answered correctly.

As regards the control sequence, or programme, if one considers the direct reversing diesel engine the basic control functions are:

(i) Speed control, with special requirements for fuel rack setting when starting.
(ii) Ahead and astern selection of rotation.
(iii) Starting air control.

Previously it has been the practice, with certain types of engines, to combine these functions under one hand wheel. However, speed control has a graduable function whereas ahead and astern selection and starting controls are just simple positioning controls. If, therefore, a graduating type of operation be applied directly to the central hand-wheel control it will be found that it cannot take advantage of the full range of sensitivity, as the first movements from STOP will be taken up with selection of ahead or astern and then starting, before speed control can be applied.

By splitting the functions into the three fundamental groups the graduating unit can be applied over the complete speed control range whilst the ahead–astern selection and starting can be dealt with by simple cylinder control, suitably interlocked to meet the engine designers' requirements.

For remote engine-room or control-room control it can be assumed that the watchkeeping engineer is capable of controlling all the separate circuits satisfactorily, and whilst single lever control may be desirable it is not essential, provided he is given adequate indication of what is happening to the engine. The interlock system may, therefore, follow along the lines of that already considered necessary by the engine designer.

For bridge control, however, it is obvious that extra safeguards must be built into the system over and above those required for remote engine-room or control-room control. The object must be to take as much responsibility as possible from the bridge watch-keeper so that he may be free to devote his attention to ship handling and be able to move the bridge controller without having to consider its reaction on the main machinery. Thus for control of steam turbines the rate of acceleration and conditions of reversal should be governed by the allowable thrusts on the blading and gearing bearing in mind that the ship will carry way on it for a considerable time when reversing and reverse torque through the propellers must be overcome without damage to the machinery.

The criterion for bridge-control design should be the ability to handle the ship as quickly as possible consistent with the safety of the main machinery and with allowances made for emergency manœuvres. Thus, with a ship manœuvring at slow speed it should be possible to put the machinery from ahead to astern or vice versa with minimum delay, whereas at high speed it is necessary to stop the engine as soon as possible and, where practicable, stop the propeller shaft turning, then engage reverse direction of rotation as soon as the main machinery will stand it. Further, this criterion will vary with the type and make of engine used.

In the following chapters some of the main marine applications of control engineering techniques will be described. However, the nature of the control system will depend to a large extent upon the individual design of the main machinery so that only general principles can be described. For descriptions of particular systems which will always be linked with particular designs of machinery the literature produced by the manufacturer should be consulted.

CHAPTER 14

Steam-raising Plant

THERE are many tasks associated with centralized control of steam-raising plant, but the basic requirements are to maintain constant steam pressure at the desired value and obtain optimum combustion conditions by regulation of air flow. Two allied problems are also involved: maintenance of constant water level in the boiler and constant steam temperature.

14.1. AUTOMATIC COMBUSTION CONTROL

In any boiler system constant steam pressure must be maintained over the complete range of operating conditions, i.e. once the plant has been started up the pressure must be maintained over the full range of propulsion demands.

With the manually operated plant this may be achieved by manually cutting burners in or out in response to the steam demand.

By the use of logic elements automatic burner control can be achieved and burners can be lit up and shut down just as with the manually operated plant.

In most ships, however, one of the most important factors in the success of the centralized control concept is the availability of a system which will permit the boilers to satisfy load changes, without lighting off or shutting down burners. This means that a wide range burner must be employed. The conventional mechanical atomizing type burner is known to be successful for turn down ratios of about 2:1. However, burners of other types are available, e.g. those using steam atomizing claim a turn down ratio of about 10:1. Others claim even higher figures. There are burners available of combined type which will provide either mechanical atomization or steam atomization depending on the setting selected by the operator.

Even with steam atomizing a problem may exist at low flows due to erratic air flow to the burners. An automatic combustion system will decrease the air supply in proportion to oil flow; however, because the areas in the register through which the air flows are constant, the velocity of the incoming air can be quite low and may fluctuate. It may become unstable and result in either an air deficiency or enough of an air puff to extinguish the burner flame.

One solution to the problem of getting efficient burner performance over a wide load range is to use mechanical atomizing during steady-state steaming of long duration. Prior to the end of the steady-state steaming period, when it is known that low speed and manœuvring conditions will be required a change is made to steam atomizing.

As a final safety precaution a steam dump valve may be used. With such a system, the minimum oil pressure to the burners is limited to a safe value to prevent loss of flame when operating at the low minimum rating during manœuvring. The dump valve from the exhaust steam system to the main condenser is opened and controlled automatically, by the control system, in response to a signal proportional to steam flow from the boiler. Thus a minimum flow limit is established on the boiler. This prevents excessive pressure build up in the boilers when the burners are operating at their minimum output and prevents steam from escaping via the boiler safety valves.

Such a system gives high efficiency during all steady-state long-time operation with the advantage of being able to manœuvre fully without pulling burners. The burners would be changed but this will be done at a time when the operators' duties are at a minimum. The steam dump valve would be closed when the engines are secured.

If such a system is used, or if a system involving very wide range burners is used in an unmanned boiler room, it is essential that loss of flame be detected. Flame failure devices of the infra-red type are

not suitable since they can be actuated by hot brickwork. Consequently the ultra-violet type must be used. The basic principle involved comprises a tube filled with a gas sensitive to ultra-violet rays and which is directed so that it sees the flame. The ultra-violet rays cause the gas to ionize and conduct current between plates inside the tube. Loss of flame causes the tube to operate a relay which triggers an alarm and shuts off the oil fuel supply.

The ultra-violet rays do not penetrate a carbon dioxide layer so the device must be focused on the base of the flame before the carbon dioxide blanket starts. For this reason the device is suitable for detecting only the burner at which it is aimed, which means that one flame failure device per burner must be fitted. This type is sensitive to ultra-violet rays only, so that it will not react to hot brickwork, etc.

The circuitry usually contains a time delay so that fluttering or a momentary loss of flow will not actuate the relay.

Figure 14.1 shows a simplified layout of one control system applied to a marine boiler installation.

Since the boilers feed into a common steam main, one master pressure sensor only is required for the installation and this will be connected at a point where the measured pressure is representative of the average.

In order to improve the response of the installation and to provide high-speed control with stability, the primary signal to the control loop derives from steam flow rather than pressure. The signal from the steam-flow sensor $FS1$ is calibrated in relay $R1$ to give a signal which is proportional to the fuel requirements and, therefore, varies the fuel input at the same rate as the load variations. The steam-pressure signal acts as a trimming effect to the firing rate and prevents any sustained deviation in pressure. The output signal from relay $R1$ is the summation of both the flow and pressure signals, the pressure signal being either positive or negative depending upon whether drum pressure is below or above the desired value setting.

The output signal from $R1$ represents fuel input requirements and is applied to both relays $R2$ and $R3$ as a variable "desired value" for both the oil-flow and the air-flow loops respectively. In the case of the oil-flow loop the maximum permitted signal is limited by relay

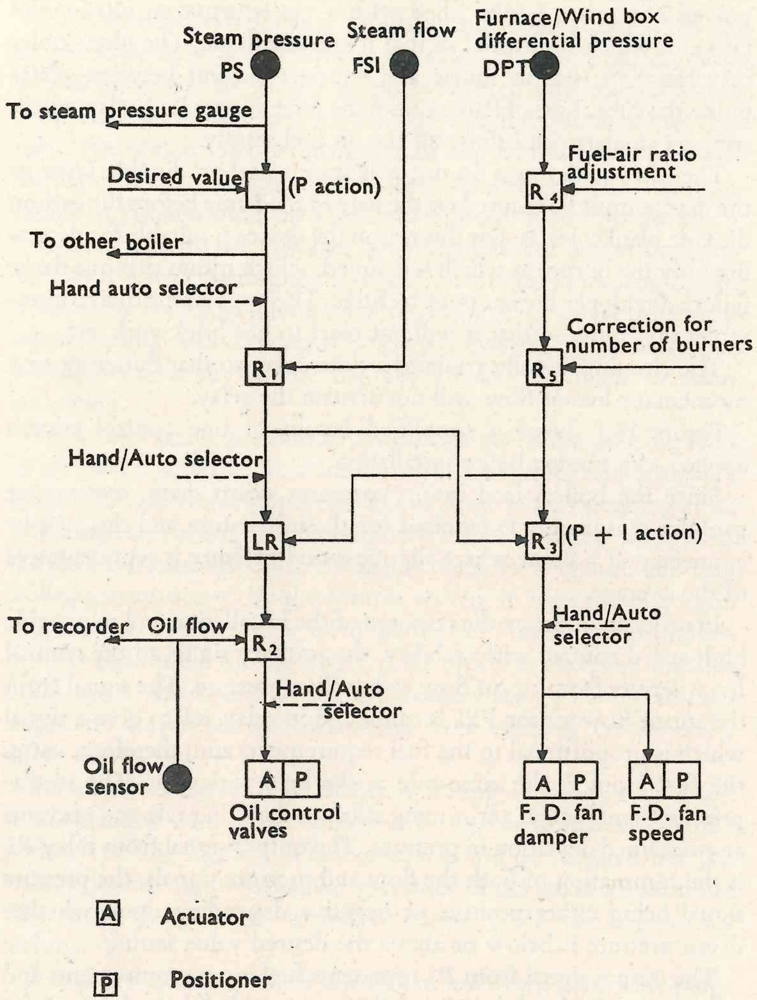


FIG. 14.1. Automatic combustion control.

LR and must always be lower than the air-flow signal at times of load change. The purpose of this limitation is to prevent a deficiency of combustion air and the danger of smoke during periods of load change and at any other time when there is a limitation on air flow. A signal from the actual oil flow, determined by the oil-flow sensor, is fed back to relay *R2* to bring the system into balance. The output signal from the oil-flow sensor is also led to an oil-flow recorder in the control room.

Since the layout of the air-supply ducting does not permit the installation of any pressure-differential-producing device, the total air flow is determined by measuring the differential pressure across the burner register, i.e. between the wind box and the furnace by the sensor DPT. This signal is linearized and calibrated in relay *R4* to make it compatible with the system. Relay *R5* is fitted to permit the system to be corrected for the number of burners in service, since the closing down of a burner necessitates the closing of its associated air register which affects the effective area of the air-flow orifice and, therefore, the air flow as recognized by the system. The total air-flow measurement as set up by relay *R5* is compared with the total load required (stated as "air demand") in relay *R3* and the air supply to the furnace is then controlled by the variation in opening of the forced draught damper at low loads and variation in fan motor speed at high loads.

14.2. AUTOMATIC CONTROL OF BOILER DRUM LEVEL

There are several systems for automatically controlling the feed water into the boiler to permit the boiler to satisfy flow demands, at essentially constant water level.

Figure 14.2 shows a two-element system.

The flow of feed water is controlled in terms of steam flow from the boiler and water level in the drum. When steam flow changes the immediate effect is a signal to the feed water-control valve with little effect from the level element. The relation between the flow sensor and movement of the water valve is programmed so that the level control only acts to make minor corrections which may be necessary. The signal from the level control passes through a delay device, so that its action is not instantaneous but is timed to modulate the overall action on a settled stable basis. This prevents false drum level signals, due to transients, from affecting the instantaneous actions of the flow control.

14.3. STEAM TEMPERATURE CONTROL

Steam temperature control is an implicit part of automatic boiler control. Careful design of the boiler is necessary to minimize slag-

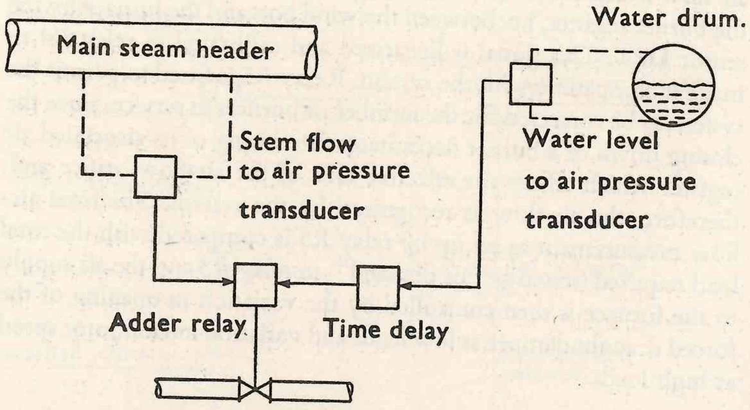


FIG. 14.2. Automatic control of boiler drum level.

ging and corrosion of superheaters, such as might occur with the use of certain bunker fuels, so that the designed boiler steam temperature can be maintained.

Superheat temperature may depend on combustion factors which are already controlled and it is important to ensure that interaction between control loops does not occur. In single-furnace boilers increase in superheat temperature arises generally from an increase in the ratio of hot gases, flowing over the superheater tubes, to the steam flowing through them. Anything which increases the former

or decreases the latter will increase the steam temperature. The most important causes are:

(i) Too much air.
(ii) Rapid firing to recover steam pressure after a sudden demand for steam.
(iii) Increase in feed rate to bring up water level.
(iv) Low feed temperature.

Automatic combustion control should result in less fluctuation in the first three cases and therefore the task of maintaining steady steam temperature should be easier when such a control system is fitted.

In many marine boilers there is a definite relationship between load and steam temperature. The major disturbances to steam temperature derive from variations in boiler load. It is therefore common practice to introduce into the temperature loop a signal which is a function of boiler load and to use this load signal to make immediate changes to modify the rate of heat transfer to the steam without waiting for temperature to change. The role of the temperature controller is, therefore, greatly reduced and consists largely of trimming the loop for changes in conditions other than those of load. This is shown diagrammatically in Fig. 14.3 which shows one practical system.

Here the steam temperature is controlled by varying the proportions of the gaseous products of combustion flowing through the superheater and the saturated steam sections. In order to permit the operation of the control system over as wide a range as possible, the system is provided with a manual relay by means of which the temperature setting can be selected within certain pre-determined limits, as laid down by the boiler designer. Relay *R* provides proportional plus integral action so that the steam temperature will return to its desired value after a change.

Steam flow is used to provide the required anticipatory signal.

It must be appreciated that this is only one form of control and others are both available and in use depending upon the design of the boiler.

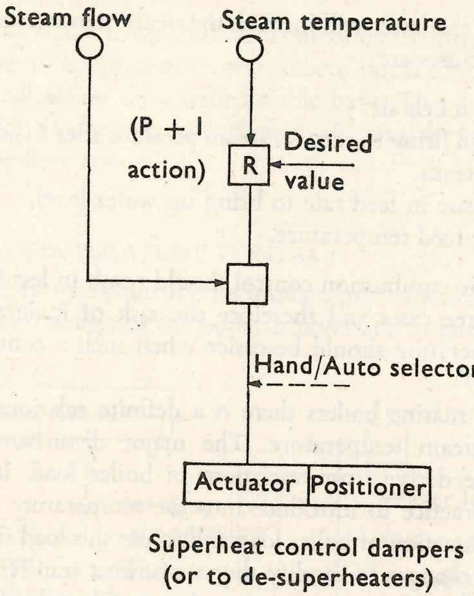


FIG. 14.3. Automatic steam temperature control.

14.4. SAFETY CIRCUITS

The detection of flame failure has already been described. In addition a boiler purge system is necessary with an unmanned boiler room to ensure that the control will not attempt to ignite a burner until the furnace has been purged of flammable gas by operating the F.D. fan at a pre-determined minimum load for a specified period of time. A suggested programme for such safety circuits is given in Fig. 14.4.

14.5. SUBSIDIARY CONTROL LOOPS

In addition to the controls already described there are many other control loops necessary for the safe and efficient operation of the steam-raising plant. The more important of them are:

(a) *Fuel oil filling system.* Controls are required to close the filling valves when tanks are filled; overflow arrangements and alarms are required.

Failure	Action: Audible alarm	Visual alarm	Shut off fuel to burner affected	Shut off fuel to boiler affected	Shut off fuel to all burners	Reduce rate of firing	Air purge	By-pass feed water	Remarks
Forced draught failure	×	×		×			60 sec minimum before reignition		
Induced draught failure	×	×		×			60 sec minimum before reignition		
Flame failure	×	×	×						
Failure of flame failure device	×	×							
Air/air atomizer failure	×	×	×						
Steam atomizer failure	×	×	×						
High water level	×	×		×				×	
First low water level	×	×							
Second low water level		× (on a separate circuit)		×					
High steam pressure		×				×			
Boiler water contamination above pre-determined level	×	×			×			×	
Fire in uptake, casing funnel, etc.	×			×					Funnel base temperature actuated. Stop fans close registers
Pressure fluctuation in firing and gas spaces under normal steaming conditions	×	×				×			

FIG. 14.4. Scheme of signals and actions for automatic control of steam-raising plant.

(b) *Fuel oil storage system.* Level indication in each tank and low-level alarm. A means of maintaining the oil at a suitable temperature which is shut off if the level of oil falls below a pre-determined level. A means of removing water from the oil.

(c) *Fuel oil transfer system.* Arrangements to transfer oil to the daily service tanks which must include the facility for drawing oil from any storage tank. The provision of filters plus a means of changing over filters.

(d) *Soot blowers.* These can be arranged for automatic sequential operation.

(e) *Smoke density* indication.

(f) *Carbon dioxide* recording.

(g) *Water contamination.* A means of indicating salinity at selected points in the system such as the main extraction pump discharge, the atmospheric drain tank and the feed filter tank plus the evaporator outlet.

(h) *Level controls* for such items as the atmospheric drain tank and the de-aerator feed water-storage tank.

(j) *Remote starting* together with pump running lights and motor-stopped alarms for such items as:

main extraction pumps,
drain tank extraction pumps,
combustion control air compressors,
soot blowing and general service air compressors,
ion-exchange plant feed pumps,
forced-draught fans,
induced-draught fans,
fuel oil pressure pumps,
fuel oil transfer pumps,
auxiliary circulating pumps,
main circulating pumps,
fire and bilge pumps,
general service pumps.

(k) *Emergency hand controls.* For emergency use manual controls together with adequate instrumentation are necessary so that the plant can be operated locally.

14.6. INSTRUMENTATION

The following gives a list of the main instrumentation and alarms which would be required with such a system:

	Alarm
Pressures	
Steam drum	
Desuperheater outlet	
Main and auxiliary feed pump discharge	
Fan discharge	Low
Burner windbox	
Furnace pressure	
Air heater gas inlet	
Funnel gas	
Fuel oil pressure at burners	
Burner atomizer air/steam pressure	
Fuel oil pressure to combustion-control system	Low
Combustion-control air pressure	Low
Temperatures	
Superheater outlet temperature	High
Desuperheater outlet temperature	High, Low
Combustion air to fans	
Combustion air from heaters	
Exhaust gas to heaters	
Exhaust gas from heaters	High, Low
Fuel oil heater outlet	High, Low
Fuel oil in settling tanks	High
Levels	
Boiler water level	High, Low
Fuel oil settling tanks	High, Low
Other quantities	
Fuel oil viscosity to burners	High
Feed water heaters, steam air inlet and outlet temperatures ...	Low
Flame failure	Alarm
Fuel oil flow to burners	
Boiler blowdown valves	Alarm when opened
Fan motors	Overload
Fuel oil pump	Overload

CHAPTER 15

Steam Turbines

THE basic requirement of the control system for steam turbines is that the shaft speed shall be controlled in accordance with the command from the bridge.

For control in the engine room where the operator would normally be a watchkeeping engineer the normal interlocks required by the turbine designer would be the only requirements. In the past the engine-control station has been arranged in some convenient location the connection between the control point and the machinery being made by mechanical linkages. The optimum method of control has been decided by the watchkeeping engineer who acts as a servo link in the chain of command between the bridge and the machinery. With remote controls and particularly with bridge control the same actions must be carried out but further safety interlocks are required since control may now be effected by persons other than a watchkeeping engineer, and the servo link previously provided by the engineer may now be absent.

These interlocks would include the following:

(a) When the bridge control is in charge the control room or engine room controls should be ineffective and vice versa.

(b) The throttle control system should be arranged so that it cannot be energized until the turning gear is disengaged.

(c) No matter how rapidly the bridge control lever is moved in the ahead or astern direction the valves, ahead or astern, should open or close at a rate no more rapidly than the set maximum rate as laid down by the turbine designer or as adjusted during trials. Both

ahead and astern valves should be equipped with overspeed limiting devices.

(d) If the controller is moved rapidly from ahead to astern the astern valve will open while the ahead valve closes. This will speed up the change in shaft speed and also reduce the range of load swings on the boiler. A similar action will take place when moving the controller rapidly from astern to ahead.

(e) In order to further reduce the possibility of excessive boiler level swings and pressure swings the control system should continuously sense boiler levels and turbine inlet pressure. Thus the turbine valves should not open more rapidly than boiler level and steam pressure will permit.

(f) When preparing for manœuvring it has been normal practice for the bridge to request the watchkeeping engineer to set the plant up for manœuvring. With bridge controls many designers now provide a "Mode" switch for this purpose. Actuation of the mode switch from "Normal" to "Manœuvring" carries out the following actions:

(i) Opens the astern guardian valve.
(ii) Starts the main circulating water pumps on high speed.
(iii) Opens L.P. turbine drains when speed has fallen below a certain figure.
(iv) Shuts off bled steam and provides alternative heating of either feed water or combustion air.

If, for any reason, the bridge operator should find it essential to go rapidly from ahead to astern and should fail to operate the mode switch then the movement of the controller into the astern position should automatically set the mode switch into the manœuvring position. It is normal with this arrangement for the mode switch to be reset to the normal operating position only by the control room or engine room watchkeeper, i.e. the switch is not reset by movement of the control lever.

(g) If the turbines can be operated from the bridge it is possible for them to be temporarily stopped and this may be unnoticed by the men in the control room or engine room. When turbines are stopped temporarily is it essential to maintain a film of oil on the shaft bearing surfaces and to rotate the turbines from time to time. Thus, during such periods, they must be placed on turning gear. Failure to do this within a short time after stopping may cause bowed shafts with the possibility of future vibration and damage. Consequently remote-control systems for turbines normally include arrangements for them to be rotated slowly by intermittent admissions of steam during periods of temporary stoppage.

(h) The steam valves should be closed for a number of emergency conditions such as high- or low-water level, loss of electric power, low condenser vacuum, failure of extraction pumps, loss of lubricating-oil pressure, excessively high temperatures in gearing or turbine bearings, excessive axial movement of rotor or excessive vibration in the turbine.

15.2. SUBSIDIARY CONTROL LOOPS

In addition, subsidiary control loops may be required on such services as:

(a) evaporating and distilling plant.
(b) gland steam,
(c) turbo generators.

15.3. A TYPICAL TURBINE CONTROL SYSTEM

Figure 15.1 shows the essential features of one control system. The bridge telegraph is of conventional type but incorporates a scale graduated in propeller speed.

(a) *Operating Alternatives*

Two main operating alternatives can be selected by means of a selector switch on the engine room console. These are:

Bridge control—by means of the bridge telegraph.
Engine room control

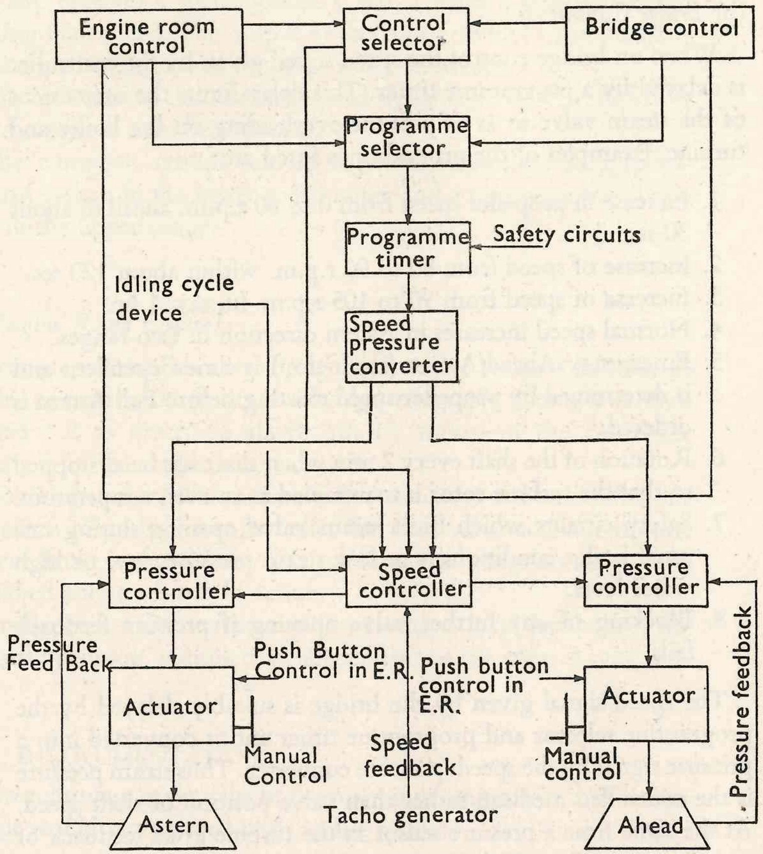


FIG. 15.1. Schematic diagram of control system for main propulsion turbines.

(i) by means of a speed dial and controller and using the same circuits as in bridge control;
(ii) by means of push buttons and switches operating the motors of the actuators directly;
(iii) by means of handwheels placed on the actuators.

(b) *Bridge Control*

When on bridge control the speed signal given by the controller is delayed by a programme timer. This delay limits the movement of the steam valve so as to prevent overloading on the boiler and turbine. Examples of the programmes fitted are:

1. Increase in propeller speed from 0 to 80 r.p.m. ahead in about 30 sec.
2. Increase of speed from 80 to 90 r.p.m. within about 120 sec.
3. Increase of speed from 90 to 105 r.p.m. in, say, 1 hr.
4. Normal speed increases in Astern direction in two ranges.
5. Emergency Ahead/Astern (crash stop) is time-dependent and is determined by propeller speed existing before Full Astern is ordered.
6. Rotation of the shaft every 2 min when shaft has been stopped so that the turbine rotor is maintained at an even temperature.
7. Safety circuits which limit steam valve opening during unusual boiler conditions, e.g. low steam pressure, low or high water level.
8. Blocking of any further valve opening if pressure feedback fails.

The speed signal given by the bridge is suitably delayed by the programme selector and programme timer and is converted into a pressure signal by the speed–pressure converter. Thus steam pressure is the controlled medium rather than valve position or shaft speed. At the same time a pressure sensor in the turbine gives feedback of steam pressure thus giving accurate positioning of the actuator and steam-valve opening. The control of the valve motor is provided by thyristors contained in the pressure controller.

However, the relationship between steam pressure in the turbine and shaft speed is not always linear since several factors can influence this, the principal one being the draught of the ship. Furthermore, in the lower-speed ranges the steam pressure is low and may be imprecise. Consequently a vernier control is provided by the speed controller. This receives the speed signal from the bridge controller

and also receives a speed feedback signal from a tacho-generator reading shaft speed. The output of this unit provides a speed adjustment signal which is fed to the pressure controller.

The influence of this ancillary speed signal is greatest in the lower speed range, as may be expected.

The complete system can be regarded, therefore, as a speed-control system in the lower speed range and a pressure-control system in the upper range.

(c) *Engine Room Control*

Communication between bridge and engine room is established in the normal manner via the bridge telegraph. Control is then carried out as described above but by means of the controller mounted on the engine room console.

Direct operation of the steam valve actuators can be carried out by means of push-button switches for each valve actuator motor. When being operated in this manner the programming units described above are out of action.

Finally each valve actuator is fitted with a local handwheel for manual operation, completely independent of the control system.

15.4. INSTRUMENTATION

The following gives a list of the main instrumentation and alarms which would be required with such a system:

	Alarm
Main turbines and condenser	
Pressure at manœuvring valves	
Pressures and temperatures at ahead and astern nozzles	
Condenser vacuum	High pressure
Condensate level	High
Air ejector steam pressure	
Condensate salinity	High
Condenser cooling water inlet and outlet temperatures	
Condensate temperature	
Gland seal pressures	

Feed water system	*Alarm*
Extraction pump discharge pressure	Low
Feedpump suction and discharge pressure	
Heater feed water inlet and outlet temperature	

Lubricating oil system	
Lubricating oil pump discharge pressure	Low
Drain tanks contents gauge	Low
Gravity tank contents gauge	Low
Pressure before filters	
Pressure after filters	Low
Temperature of oil from turbine and gearing bearings ...	High

CHAPTER 16

Diesel Engines

16.1. DIRECT REVERSING DIESEL ENGINES

This type of diesel engine is the most popular for marine propulsion and in the normal, or hand, control the watchkeeping engineer has four controls:

(i) Direction control.
(ii) Fuel-control lever.
(iii) Starting air lever.
(iv) Speed governor.

The action of the direction control is to position the camshaft so that the engine will turn in the required direction.

The fuel control lever is then set to the starting position.

The starting air lever is now operated. This lever controls a valve which admits compressed air to the cylinders via a distributor. The purpose of the distributor is to supply air to one cylinder at a time so that the engine may turn. When the engine fires the starting air lever is released and the fuel control lever advanced to the required running speed, viz. slow, half, full speed. The quantity of fuel admitted to the engine is limited by a load lever the movement of which is restricted by the fuel-control lever.

The speed governor is arranged to limit the speed of the engine and is set to the maximum speed at which it is desired to operate the engine.

Various interlocks are provided so that mal-operation is avoided. For example, it is not possible to operate the starting air lever until the direction changing linkage has adjusted itself.

The optimum method of control is decided by the watchkeeping engineer who acts as the servo link in the chain of command between bridge and machinery. Consequently when bridge controls are fitted further interlocks are required just as with bridge-control systems for turbines.

These additional safety interlocks would include such items as:

(a) Where bridge control is fitted automatic starting should be incorporated so that the bridge watchkeeper has maximum freedom to concentrate on ship handling.

(b) When on bridge control the engine-room or control-room levers should be ineffective.

(c) Upon initiating a manœuvre by means of the bridge controller the system must first check that the fuel admission is set to zero before starting air is admitted to either the ahead or astern ports of the air distributor. The direction of rotation must then be checked and only after firing speed "n" is reached should the fuel control linkage be allowed to move to the starting position. After a suitable time delay the starting air can be shut off.

Simultaneously the engine speed must be checked to determine that firing has occurred. If so, then the fuel-control lever can be gradually moved to correspond with the controller (or telegraph) setting. The time taken to carry out this movement will depend on the engine designer's instructions since it depends both on the size of the engine and the thermal time constants of the particular design of engine.

(d) In the event of failure to fire the control linkage should return to zero and the starting operations repeated. Only a limited number of starting attempts would be acceptable, dependent upon the design of the engine plus control system, since an unlimited number of starting attempts may exhaust the available air supply resulting in a "dead" ship.

(e) Once the engine is running normal speed increases, reductions or stoppage of the engine are achieved by the system moving the fuel-control lever to correspond with the telegraph setting.

(f) If there are any barred speed ranges, these must be passed

through automatically, even if the controller setting has been located within the barred speed range.

(g) To effect emergency manœuvres from full ahead to full astern, or vice versa, the fuel-control lever must be moved to zero and, when the engine revolutions have dropped to "n", air is admitted to the ports of the distributor to effect rotation in the opposite direction. When the engine has stopped and reversed the normal starting procedure must be executed.

16.2. UNIDIRECTIONAL DIESEL ENGINES WITH REVERSE GEAR

With this form of propulsion unit the main problems which arise during slowing down and reversal are:

To ensure that the reverse torque from the propeller does not exceed the output torque of the engine when attempting to reverse. Failure to ensure this can result in stalling.

To ensure that relative speeds of engagement of input and output shafts of clutches are at a minimum and that the engine speeds up only after the clutches are fully home. A relevant factor here is that there is normally a delay between the clutch-control lever being home and the clutches themselves being fully home.

16.3. A TYPICAL CONTROL SYSTEM

Figure 16.1 shows the basic principles of a control system for a direct reversing diesel engine.

Such a system allows the bridge watchkeeper to assume direct control of the engine and enables him to stop, start, reverse and change revolutions as desired. The action is automatic and the control system controls starting air and fuel admission to give satisfactory manœuvring.

Alternatively the engine can be controlled from the engine room or control room using the same circuits.

Finally a manual control is provided, this being independent of the automatic system.

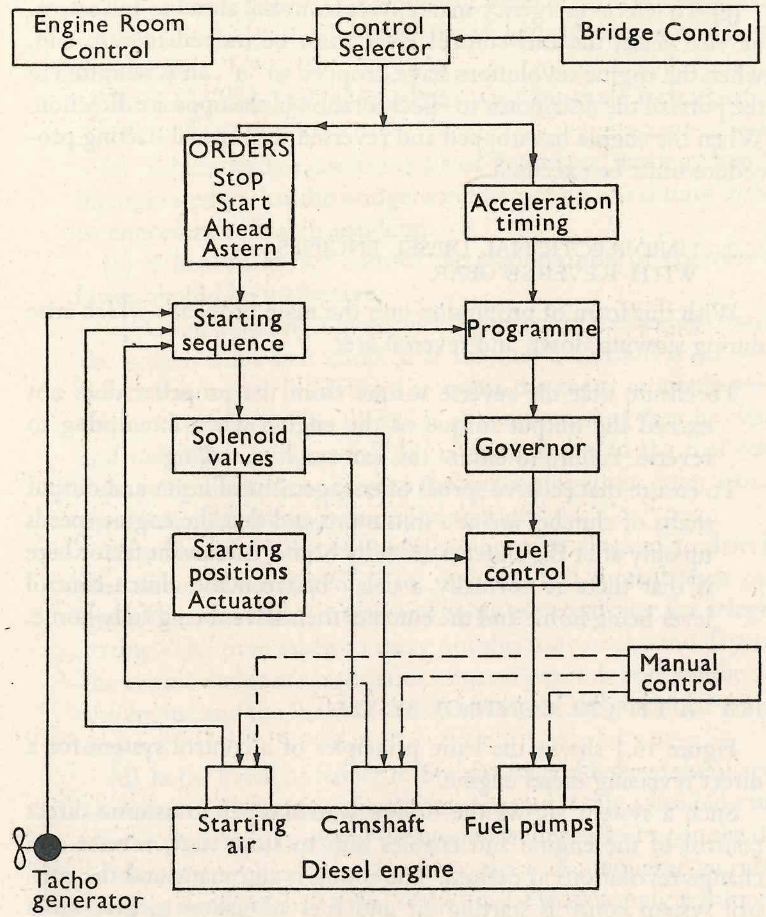


FIG. 16.1. Schematic diagram of control system for direct reversing diesel engine.

Manœuvring Sequences

When the bridge watchkeeper has initiated a particular manœuvre the control system performs the following sequences of actions:

(a) *Stop to Full Ahead*

1. Check that the fuel admission is set to zero.
2. Move the starting air lever to "ahead"; this positions the camshaft so that starting air is admitted to the "ahead" ports of the distributor.
3. Check that engine rotation is "ahead".
4. Wait for engine to reach firing speed "*n*" (adjustable).
5. Raise fuel-admission linkage quickly to starting position.
6. After time delay t_1 (adjustable) cut off starting air.
7. After time delay t_2 (adjustable) check whether engine speed is above or below "*n*".
8. If speed is above "*n*", showing that engine has fired, lower or raise the fuel-admission linkage to correspond with the controller setting. The time of travel from "gate" to "full" is adjustable.

(b) *Starting Astern*

9. As for starting "ahead" except that air is admitted to the "astern" ports of the distributor, by positioning the camshaft, and a check made for astern rotation.

(c) *Changing Engine Revolutions*

10. Move the fuel-admission linkage to correspond with the new telegraph setting as in (8).

(d) *Full Ahead (or Full Astern) to Stop*

11. Move the fuel-admission linkage quickly to zero. Travel from "full" to "zero" is adjustable but 0·5 sec is a common figure.

(e) *Full Ahead to Full Astern*

12. Move the fuel-admission lever quickly to zero as in (11).
13. Wait until the revolutions ahead have fallen to "*n*".
14. Move the starting air lever to "astern". This admits starting air to the astern ports of the air distributor.
15. Wait until the engine has stopped and reversed.
16. Perform normal starting actions (3) to (8).

(f) *Failure to Fire*

If, at item (8) of the sequence, the speed is below "*n*" showing that the engine has failed to fire, move the fuel-admission linkage quickly to zero and repeat actions (1) to (7).

17. If attempts to start are still unsuccessful after a time t_3 from the instant the engine has been turned in the appropriate direction, stop further action and sound alarms on bridge and in engine room. Time t_3 is adjustable from, say, 1 to 60 sec. Alternatively time t_3 can be replaced by a counting device which counts the number of starting attempts. Up to, say, four starts may be permitted before the system shuts down.

(g) *Barred Speed Ranges*

After item (8) or (10) when telegraph is moved to increase speed a time delay is allowed to cover fuel-lever movement, governor adjustment and settling of engine speed.

18. If engine speed is in a critical (or barred) speed setting, after this delay, the fuel-lever setting is increased slowly until engine speed is "out of critical speed range" signal.
19. Conversely if telegraph setting is lowered the fuel lever is lowered further than initial setting to bring the engine speed below critical. To prevent "hunting" the raise/lower fuel circuit is locked off by the critical speed circuit until a further telegraph movement takes place.

16.4. INSTRUMENTATION

The following gives a list of the main instrumentation and alarms which would be required with such a system:

	Alarm
Main engine cooling system	
Pump suction and delivery pressures	Low
Cooler inlet and outlet temperatures	High
Cooling tanks' contents gauge	Low
Reserve cooling tank contents gauge	Low
Outlet temperature of each component cooled	High
Main engine lubricating oil	
Pump suction and delivery pressures	Low
Cooler inlet and outlet temperatures	High
Drain-tank contents gauge...	Low
Pressures before and after filters	High
Pressure at turbo blowers	Low
Temperature at each main bearing	High
Pressure at main bearing inlet manifold	Low
Cylinder lubrication	
Temperature of propeller shaft bearings	High
Pressure of oil to reduction gears	Low
Temperature of reduction gear bearings	High
Temperature of thrust bearing	High
Main engine exhaust gas	
Temperature of gas leaving cylinder	High
Temperature of gas before and after blowers	High
Main engine pressure charging system	
Temperature of air at inlet	
Air pressure after filters	Low
Temperature of air after blowers	High
Temperature of air after coolers	High
Turbo blower speed	Low
Temperature inside scavenge belt	High
Main engine fuel oil	
Oil pressure before and after high pressure fuel pumps ...	High, Low
Oil temperature before and after heaters	High, Low
Main engine starting air	
Pressure in each air bottle	Low
Pressure in starting air manifold on engine	

CHAPTER 17

Generation and Distribution of Electric Power

COMPLETELY reliable operation of the electric generating plant is vital for the reliable operation of any automatic-control equipment and for the reliable operation of the main engineering plant in any modern ship where most auxiliary machinery is now electrically driven.

To date it has been normal practice for generating sets to be started up, synchronized and connected to the ships system under manual control. Various protective devices are fitted to ensure continuity of supply in the event of overload or failure of a prime mover. These include devices for automatic load shedding or preferential tripping systems, overload tripping devices and short-circuit tripping devices.

Equipment is now being fitted for the automatic synchronizing of a.c. generators. Essentially such equipment carries out the following functions:

(i) Adjusts the voltage of the "incoming" generator until it equals the busbar voltage.

(ii) Compares the frequency of the "incoming" generator with that of the busbars and provides an appropriate signal to the engine governor to increase or decrease speed until the frequencies are within a predetermined range. It ensures that the incoming generator frequency is slightly higher than that of the busbars.

(iii) Checks that voltage and phase angles are within predetermined limits.

(iv) Provides a signal to close the generator circuit breaker of the incoming machine.

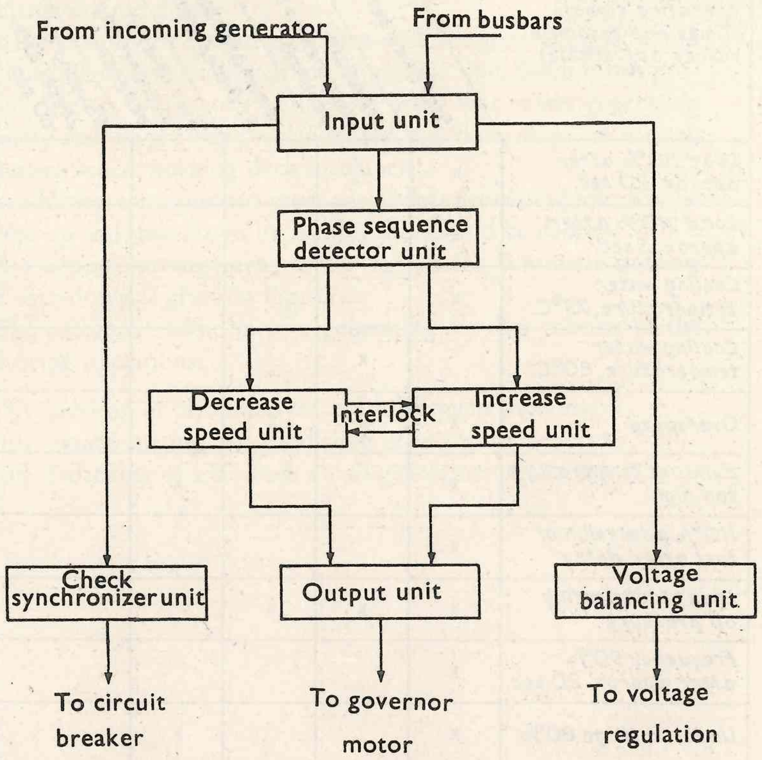


FIG. 17.1. Schematic diagram of typical automatic synchronizing unit.

This sequence is illustrated in Fig. 17.1.

Other equipment is available to control start-up and shut-down of the complete system, so as to ensure security of supply under all loading conditions. With such a system the generating sets are numbered in a definite sequence and automatic start-up always takes

Disturbances and operating signals (Times and response values adjustable)	Start new set	Immediate disconnexion of generator	Immediate stopping of Diesel engine	Time delay in disconnecting generator	Stopping of Diesel after 60 sec.
Load 80% after approx. 20 sec.	x				
Load 110% after approx. 5 sec.	x				
Cooling water temperature, 75°C.	x				
Cooling water temperature, 80°C.		x			x
Overspeed	x	x	x		
Exhaust temperature too high	x			x	
110% admission of fuel after delay	x				
Loss of lubricating oil pressure	x	x	x		
Frequency 90% after approx. 20 sec.	x				
Under voltage 80%	x			x	
Reverse power after approx. 2 sec.	x	x			
Short circuit trip of generator C.B.	x	x			

FIG. 17.2. Scheme of signals and actions for automatic control of generating sets.

place in the same order, e.g. 1–2–3. With a dead ship set no. 1 is started up and connected to the system and this generator acts as the pilot machine. If the system load exceeds the capacity of the set in operation a further generator is automatically connected after a predetermined time delay.

If the load on the system is too low and one generating set could be shut down this is indicated but automatic shut-down is not provided. This is deliberately omitted in order that excess generating capacity shall be provided when desired, e.g. if the ship is in narrow waters or when working deck machinery.

In addition arrangements are made so that generating sets may be started up and shut down by hand if this should be desired.

A schematic arrangement of the signals for automatic start-up and shut-down is given in Fig. 17.2.

The automatic addition of a generating set thus consists of the following operations:

(i) Start up of prime mover and subsequent supervision.
(ii) Synchronizing and paralleling of incoming generator.
(iii) Balancing of kW loads on all generators connected.

CHAPTER 18

Pumping of Liquid Cargoes

UNTIL recent years most tankers were relatively slow, carrying bulk cargoes on long hauls, or they were parcel carriers having a multi-port discharge at slow rate. Today tankers are fast, efficient transporters of oil in bulk and each ship now replaces several of her predecessors. Large tankers now load at rates in excess of 9000–10,000 tons/hour and discharge at rates up to 6000 tons/hour at modern terminals. At these rates a large number of cargo valves must be opened and closed rapidly to avoid overflows. With the relatively few men available this becomes a tremendous task and introduces an element of danger since an oil overflow can result in harbour pollution or fire.

To appreciate this it is useful to study what is required when a tanker reaches a loading terminal. De-ballasting is the first task and continual care and frequent ullaging is required so as not to draw air and trip out the pumps. Also stripping has to be carefully watched to avoid the waste of a pump working on a dry tank. Even though this may mean only frequent visits to the tank top and a look inside, considerable time and effort is required frequently in great heat or in a deluge of rain or sand.

The critical moment comes when ready to load for it is essential to see that all cargo valves have been correctly set and, as the oil starts to flow, that it goes only to the correct places. Throughout loading ullaging has to be carried out, the loading volume continuously calculated and the tonnage rate arrived at. Dipsticks must be available for topping off and all the time the trim and list of the ship must be carefully watched. Whether or not the ship starts to list will

depend on how often the tanks have been ullaged and on how up-to-date the information is so as to trace the cause and take remedial action.

When "stick level" is reached the officer in charge has to be highly skilful and observant in deciding the order in which the tanks are topped up if unnecessary movement about the deck is to be avoided. In the meantime continuous calculation is necessary checking temperatures and specific gravities so as to finish with the ship properly trimmed and to know how much to put into the last tank to bring the ship to her permitted draught.

When discharging at the high rates previously mentioned and using large centrifugal pumps, a group of tanks is usually discharged simultaneously. When shifting from one group to another the cargo valves must be operated rapidly to avoid having the cargo pumps lose suction, overspeed and trip out.

Large tankers have 18 in. valves in the cargo tanks and soon 20 in. valves will be in use; 20 in. valves are used in the pump room and discharge manifold. With a free-flow system, 30 in. valves can be used. It is obvious that with such conditions mechanical assistance must be provided for opening and closing the cargo valves if high loading and discharge rates are to be maintained and also some form of level indication is required to eliminate the necessity for sounding tanks manually. In addition some form of loading calculator is necessary for the assistance of the officer in charge and to obviate the setting up of unduly high stresses in the hull.

The logical step which has been applied in an increasing number of modern tankers is to locate the cargo valve controls and remote level indicators in a central control room. With such an arrangement the officer in charge has control over the entire cargo system and complete knowledge of the tank contents.

Where such systems are applied the bridge-aft construction is almost essential so that a clear view of the deck is available from the control room.

The operating system must be chosen with care. Electric and pneumatic actuators are generally not acceptable for use in cargo compartments; the former because of the hazard of incendive sparking and the latter because of the possibility of generation of static electricity, the power requirements and the long distances involved. For indication, however, e.g. limit switches and float switches, electrical elements are often used, but in these applications the units must be certified "intrinsically safe". This means that the energy in the circuit is so low that even if faults occur in the electrical elements danger of explosion cannot occur; as a general guide

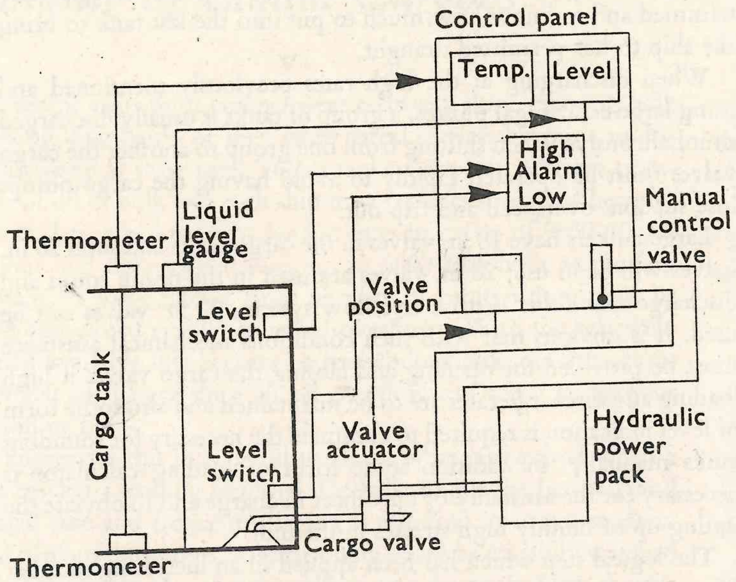


FIG. 18.1. Schematic diagram of cargo pumping controls.

this means an energy level of less than 0·2 millijoule. Pneumatic circuits are often used as pilot systems for operation of an hydraulic system.

A typical control system, drawn for one tank, is shown in Fig. 18.1.

Two main hydraulic lines run from the power source along the weather decks and from these lines are taken smaller-size intermittent lines to each control valve and associated equipment.

In addition to a centralized control console it is normal to provide local control of each valve actuator sited on deck relative to the cargo valves and in the pump room adjacent to the cargo valves. Speed control is achieved by the setting of a restrictor valve sited in the line between the controller and the actuator.

A locking valve is usually incorporated in the system to ensure that actuators remain locked-closed for the long periods on voyage when the motor pump unit is not running and when the accumulator would be discharged.

In many cases an emergency hand-pump equipment is provided. In case of a power failure or weather deck damage then by means of normally plugged tee pieces this equipment can be connected to the intermediate lines of the valve to be operated. The unit comprises a hand pump, supply tank, control valve and a pressure gauge.

In addition to the remote controls described above some automatic controls are occasionally fitted in modern tankers.

Automatic Cargo Pump-discharge Valve Control

When centrifugal cargo pumps are installed it is possible to vary the throughput of the pump by fitting a discharge-line control valve. This valve is opened or closed in response to the conditions on the suction side of the pump. By this means it can be arranged that the pump runs at rated power throughout the discharging period.

Control for Shortest Turn-round Time

It is sometimes arranged for the adjustment of cargo line valves so that when a tank is emptied to a predetermined level the pump concerned is gradually transferred from that tank to the next tank in the programme. Thus drawing in of air in the cargo lines at low oil levels is minimized and the pumps always operate at full capacity. This tends to reduce the turn-round time of the ship to a minimum.

In this system, on attaining a certain minimum oil level, in a tank which is being pumped out, the pump initially draws air. This causes the pump speed to increase. This increase of speed (or, if the pump turbine is governor controlled, the change in the steam supply) is used as a signal to close the valve of the practically empty tank and to open the valve of the next full tank in the programme. Thus the pump always runs at full capacity and remains connected so that it pumps oil as long as possible from the practically empty tank.

CHAPTER 19

Refrigeration

THE majority of modern refrigerated ships use Refrigerant 12 or Refrigerant 22 as the refrigerants, frequently referred to by their trade names of Freon 12 and Freon 22.

19.1. BRINE SYSTEMS

A large number of such ships operate with a secondary refrigerant circulation using calcium chloride brine in a closed system. Figure 19.1 shows the basic principles of such a marine refrigerating system.

The refrigerant is compressed to a pressure corresponding to the saturation temperature at which it liquefies in the condenser, this temperature being governed by the temperature of the sea water circulating through the condenser at the time. Pressure gauges are fitted at the compressor suction and discharge. The gauges are calibrated in terms of equivalent saturation temperature this being more convenient than pressure for assessing the duty of the plant.

The refrigerant is then expanded through a control valve, or expansion valve, being cooled in the process and passed into the evaporator. Here the latent heat of evaporation of the liquid refrigerant is extracted from the brine thus lowering the brine temperature to the required figure. The cold brine is then circulated through batteries of coils or grids in the chambers. The air is cooled by circulation over these coils or grids by means of fans and circulated throughout the cargo compartments by means of delivery and suction ducting.

Owing to the requirements of the trades on which such ships operate these ships are extensively sub-divided. Thirty chambers is quite usual but a few ships have as many as seventy different compartments. Most, or all, of the compartments are arranged to carry a variety of different cargoes from frozen goods at temperatures down to, or below, −20°C (−4°F) up to chilled

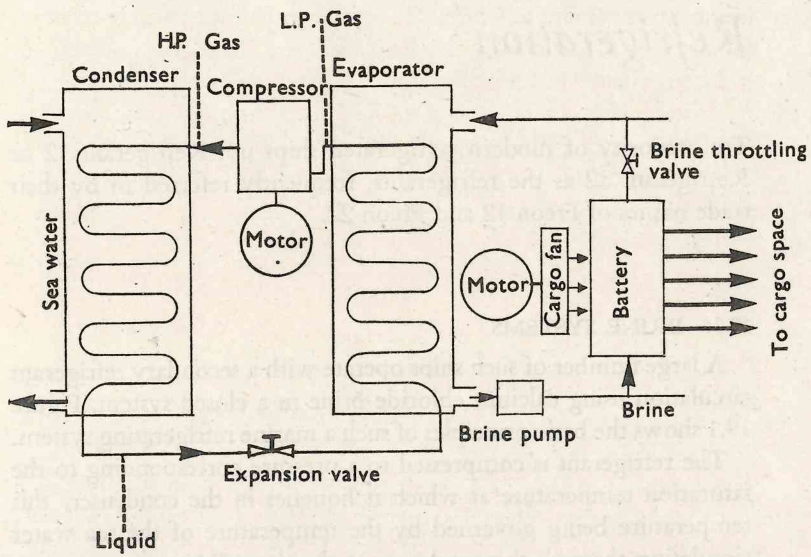


FIG. 19.1. Schematic diagram of refrigerating system using brine.

cargoes such as fruit, cheese or butter carried at 0°C–4·5°C (32°F–40°F) and, in an increasing number, of bananas carried at 12°C (54°F).

The numerous varieties of cargo not only calls for sub-division of the ship but also entails considerable divergency in the sizes of individual cargo spaces to suit the different quantities of the various cargoes available at different times of the year.

A multi-temperature brine system is used to serve these spaces the number of brine temperatures varying from one trade to another

depending on the number of different cargoes to be carried at any one time.

There are two methods of achieving different brine temperatures. The first is simply by running the compressors at different evaporation temperatures in which case each brine cooler is coupled to a separate set of brine mains connected through a multiple valve and header arrangement to whichever compartments are to carry cargoes requiring that particular brine temperature on any given voyage.

Alternatively, intermediate brine temperatures can be obtained by mixing, that is by injecting a colder brine into a higher temperature brine system. This method of injection can be used in two ways.

If the amount of cargo to be carried on the high-temperature system is small the circulating brine is by-passed around the evaporators and the temperature of the system controlled by injecting cold brine from one of the low-temperature systems. The surplus brine in the high-temperature system is then led back to the low-temperature system for re-cooling.

On the other hand, if a compressor working on a high-temperature system is short of the necessary capacity for the demands upon it, cold brine may be injected into the system to provide the additional output required, assuming of course that the compressor working on the low-temperature system has the capacity to spare.

The tasks involved in running a marine refrigerating plant may be summarized as follows:

(i) Control of expansion valve. This controls the mean difference of temperature between the brine outlet temperature and the refrigerant temperature in the evaporator.

In Refrigerant 12 and Refrigerant 22 systems this is normally a float-operated automatic valve although hand control is provided.

(ii) Control of space temperature by controlling the brine flow. In automatic multi-space systems this is achieved by means of a temperature sensor located in the air duct providing the control signal for the brine throttling valve.

(iii) Adjustment of compressor output to suit variations in load. If, for instance, the brine temperature should fall below the desired value then the compressor output is reduced. This can be achieved automatically by conventional means.

(iv) Control of condenser cooling water to maintain the required condensing temperature. At sea it is normal to use the full flow of sea water but, under deep-draught conditions, the water flow is often reduced by means of a by-pass system in order to reduce the erosion hazard in the condenser tubes.

(v) Speed variation of fans. At the end of the initial cooling-down period the fan speed is normally reduced for the steady running conditions. In some cases, reversal of fans may be necessary, say, once per day for banana and citrus fruits so that the temperature in the chamber is equalized.

(vi) De-frosting of the cooler batteries. If the mean temperature difference between the space temperature and the brine temperature increases then de-frosting may be necessary. This can be arranged automatically on a time cycle.

(vii) Logging of variables.

Variables to be Monitored

The temperatures of the cargo chambers are the parameters of the greatest interest. However, due to the thermal inertia of the cargo chamber, its contents and the insulation the cargo hold, temperature will not rise appreciably for some considerable time after a plant failure. It is important, therefore, to be able to read fan-discharge air temperatures in order to get early warning of plant failure.

The sea-water temperature, the brine temperature both entering and leaving the evaporator and the refrigerant temperatures at the compressor suction and discharge and after the condenser are also important for efficient operation.

The pressure of the gas entering the condenser, indicated as saturated temperature on the compressor discharge gauge, should normally be about 5·5°C to 8·2°C (10°F to 15°F) above the existing sea-water temperature. Consequently the refrigerant pressure at the condenser is required. This can be used to deduce the effectiveness of the condensing operation at the prevailing sea-water temperature.

For Refrigerant 12 the pressure should be about 84 psi for a sea-water temperature of 21°C (70°F).

Similarly, the liquid after the expansion valve should be at a pressure such that its boiling point is about 4°C (7°F) below the desired brine temperature, in order that the latent heat of evaporation is utilized to the full in the evaporator. For Refrigerant 12 the pressure should be about 5 psi for a brine temperature of —15°C (0°F).

The carbon dioxide content of the air in the cargo chambers is sometimes required since some fruit cargoes generate CO_2 and can affect the cargo; e.g. in the case of apples excess of CO_2 results in "brown heart" and too low a concentration may result in surface blemishes.

A Typical Monitoring List

The following is based on a ship with 250,000 ft^3 of refrigerated space divided into fourteen chambers. The plant assumed is:

3 Compressors (including 1 stand by).
2 Brine pumps.
1 Sea-water pump (plus 1 stand by).
28 Cargo fans (i.e. 2 per chamber).
14 Chambers.

List of points

28 Fan discharge temperatures.
42 Cargo space temperatures.
3 Refrigerant gas temperature at compressor discharge.
3 Refrigerant pressure at compressor discharge (i.e. saturation temperature).
3 Refrigerant pressure at compressor suction (i.e. saturation temperature).
3 Refrigerant liquid temperature after condenser.
3 Brine temperature entering evaporator.

List of points (cont.)

3 Brine temperature leaving evaporator.
1 Sea-water temperature.
1 Sea-water pressure.
Running lights for all motors.
(Some owners measure motor current.)

CHAPTER 20

Commissioning and Maintenance

20.1. COMMISSIONING

Commissioning of a ship fitted with any degree of control equipment requires a formal sequence of testing and adjustment culminating in the appraisal of performance during the sea trials. The sequence will best start with static inspection and testing, then dynamic testing and adjustment of loops and finally a series of overall system dynamic tests. As yet there is little published information on this subject and the basis of acceptance must, of necessity, be subjective. It is probable that performance assessment will be dealt with best by a "trials team".

The previous chapters have attempted to show how the ships' machinery may be split into "areas" for the application of controls. Similarly, where a control system has been applied to an "area" of the ship the system can be split into a number of simple loops. For testing prior to commissioning the machinery should again be split into areas and each system should be broken down into the single loops contained in the system. This process gives a basis for static and dynamic testing of loops. Dynamic testing of systems can then be carried out during the trials when the machinery is put through its full operating range. The preparation of such a programme involves a great deal of care and time both in the compilation of the tests and in carrying out the tests.

To carry out such a programme the following points are vital if success is to be achieved:-

(i) Sufficient time must be allocated for the tests.
(ii) During the tests no persons should be working in the machinery spaces other than those involved in the tests.
(iii) The necessary services should be available.
(iv) The various systems should have been set up previously so that minimum adjustments will be necessary.

An example of what is intended by a programme is given below, based on a ship with diesel propulsion, but it must be emphasized that each ship must have its own discrete programme prepared. In addition, where an overall control system has been fitted to one area of the machinery, e.g. automatic boiler control, the control engineer responsible for the system design should collaborate in the preparation of the test programme.

An Example of a Test Programme

Ref. no.	*System*
1	Fuel valve cooling.
2	Main engine lubricating oil.
3	Camshaft lubricating oil.
4	Turbo blower lubricating oil.
5	Jacket cooling water.
6	Main engine exhaust and pressure charging.
7	Main engine oil fuel.
8	Heavy oil fuel tanks.
9	Gas oil tanks.
10	Compressed air.
11	Shafting.
12	Steam.
13	Purifiers.
14	Bilge.
15	Distilled water.
16	Domestic.
17	Washdeck.
18	Turbo alternator.
19	Diesel alternators.

Ref. no.	*System (cont.)*
20	Crankcase mist detector.
21	Bearing wear down gauge.
22	Fire alarms.
23	Bridge emergency stop.
24	Piston rod gland drains.

System no. 1	*Fuel valve cooling*
1.1	Start no. 1 pump and demonstrate tank-level switch.
1.2	Adjust pressure on pressure switch to demonstrate alarm point.
1.3	Stop pump to demonstrate low-pressure alarm.
1.4	Remove pump alarm by opening isolator on starter.
1.5	Start no. 2 pump.
1.6	Repeat 1.3 for no. 2 pump.
1.7	Repeat 1.4 for no. 2 pump.
1.8	Demonstrate remote manual control of valves from control station; steam valve—psi; sea-water valve—psi.
1.9	Demonstrate sea-water control valve maintaining oil temperature at—°F by opening steam valve to heater by hand jack.
1.10	Demonstrate steam control valve maintaining oil temperature at—°F by opening sea-water valve to cooler by hand jack.
1.11	Demonstrate set-point regulator varying temperature.
1.12	Demonstrate high-temperature switch set at—°F, rising.
1.13	Demonstrate low-temperature switch set at—°F, falling.
1.14	Prove remote thermometer by comparing with local thermometer.

System no. 1	*Fuel valve cooling (cont.)*
1.15	Demonstrate fail-fix of sea-water valve by positioning valve at mid stroke by remote manual station and remove control air supply to valve.
1.16	Repeat 1.15 for steam valve.
System no. 2	*Main engine lubricating oil*
2.1	Start pump no. 1; pump no. 2 on auto; demonstrate tank-level switch.
2.2	Regulate pressure by adjusting by-pass valve from remote station.
2.3	Prove temperature sensors against local thermometers during warm up of main engine.
2.4	Adjust pressure on pressure switches to demonstrate pump change over, alarm and engine shut down in descending order of lubricating oil pressure.
2.5	Stop pump no. 1 to demonstrate auto start of pump no. 2 and failure alarm of pump no. 1. Remove alarm by opening isolator.
2.6	Pump no. 2 on manual; pump no. 1 on auto. Stop no. 2 to demonstrate auto start of pump no. 1 and failure alarm of pump no. 2. Remove alarm by opening isolator.
2.7	Both pumps on manual; pump no. 1 running. Stop pump to demonstrate pressure alarm and engine shut down.
2.8	Both pumps stopped; remove all plugs from pressure charger lubricating oil flow switches. Test individually by re-inserting plug.
2.9	Pump no. 1 running. Demonstrate auto start of lubricating oil filter.
2.10	Repeat 2.9 for high-differential pressure alarm.
2.11	Repeat 1.8.
2.12	Repeat 1.9.

System no. 2	*Main engine lubricating oil (cont.)*
2.13	Repeat 1.10.
2.14	Repeat 1.11.
2.15	Repeat 1.12.
2.16	Repeat 1.14 at same time as 2.3.
2.17	Repeat 1.15.
2.18	Repeat 1.16.
System no. 3	*Camshaft lubricating oil*
3.1	Repeat 1.1.
3.2	Repeat 1.2.
3.3	Repeat 1.3 and pump failure.
3.4	Repeat 1.4.
3.5	Repeat 1.5.
3.6	Repeat 1.6 and pump failure.
3.7	Repeat 1.7.
3.8	Repeat 2.10 on trial.
System no. 4	*Turbo blower lubricating oil*
4.1	Repeat 1.1.
4.2	Witness pressure gauge in control room.
4.3	Demonstrate remote manual control of sea-water valve from station—psi.
4.4	Demonstrate sea-water control valve maintaining oil temperature at—°F on trials.
4.5	Demonstrate temperature sensors in lubricating oil-bearing returns on trials.
4.6	Repeat 1.14 on trials.
4.7	Repeat 2.10 on trials.
4.8	Repeat 1.15.

System no. 5	*Jacket cooling water*
5.1	Start pump no. 1.
5.2	Repeat 1.2.
5.3	Repeat 1.3.
5.4	Repeat 1.8.
5.5	Repeat 1.9 at—°F.
5.6	Repeat 1.10 at—°F.
5.7	Demonstrate adjustment of set point from master controller.
5.8	Prove temperature sensors against local thermometers during warm up of main engine.
5.9	Repeat 1.12 at—°F for each switch.
5.10	Repeat 1.13 at—°F.
5.11	Repeat 1.14 at same time as 5.8.
5.12	Repeat 1.15.
5.13	Repeat 1.16.
5.14	Prove level switch in head tank by draining.

System no. 6	*Main engine exhaust and pressure charging*
6.1	Test scavenge space fire wires with candle.
6.2	Demonstrate remote manual control of valves from station.
6.3	Demonstrate sea-water control valves maintaining air temperature at—°F on trials.
6.4	Demonstrate set point regulator varying temperature on trial.
6.5	Prove temperature sensors on air outlet from each cooler by comparing with local thermometer.
6.6	Prove long-distance thermometer by comparing with local thermometer during 6.4.
6.7	Prove long-distance pyrometers on exhaust system by comparing with local thermometers during 6.4.

System no. 7	*Main engine oil fuel*
7.1	Start standby oil fuel surcharge pump and circulate oil fuel on engine.
7.2	Demonstrate remote manual control of steam valve.
7.3	Prove high-temperature alarm during 7.2 with set point at—°F rising.
7.4	Demonstrate steam control valve maintaining—°F.
7.5	Prove long-distance thermometer during 7.3 and 7.4 by comparing with local thermometers.
7.6	Repeat 2.10.
7.7	Reduce oil fuel temperature by manual control of steam valve to prove low-temperature alarm at—°F falling.
7.8	Repeat 1.2.
7.9	Repeat 1.3.
7.10	Repeat 1.16.

System no. 8	*Heavy oil fuel system*
8.1	With dirty heavy-oil fuel settling tank empty, fill until pump stops automatically.
8.2	Override transfer pump stop. Continue filling until tank overflows and prove overflow tank high-level alarms individually.
8.3	Raise temperature in tank to—°F rising and prove temperature alarm.
8.4	Purify into clean service tank and prove low-level alarm.
8.5	Prove high-temperature alarm set point—°F rising during 7.3.
8.6	With boiler oil fuel tank empty, fill until low-level alarm cancels and subsequently high level operates.
8.7	Heat up tank and prove high-temperature alarm.

System no. 9 *Gas oil system*

9.1 With port gas oil tank empty pump up and prove low-level alarm. Continue filling and prove high-level alarm.
9.2 Repeat 9.1 for starboard.

System no. 10 *Compressed air system*

10.1 Compress main air reservoirs to—psi.
10.2 Repeat 1.2 for each reservoir.
10.3 Compare pressure gauge in control console with local pressure gauge.
10.4 Reduce reservoir pressures individually to prove alarm.
10.5 Run no. 1 compressor and raise air discharge temperature to—°F rising to prove temperature alarm.
10.6 Repeat 10.5 for no. 2 compressor.
10.7 Repeat 1.2 for control air reservoirs.
10.8 Repeat 1.2 for control air main.
10.9 Reduce pressure to control air system and control air reservoir to prove alarm.
10.10 During 10.9 test low-pressure trip of general service air compressor.
10.11 Pump up control air reservoir and prove temperature alarm point.
10.12 Prove general service air compressor high-pressure cut out.
10.13 Compare pressure gauge in control console with local pressure gauge.
10.14 Adjust water level in compressor head tank to prove alarm.
10.15 Demonstrate remote operation of main starting air-stop valve and prove limit switch of valve in open position.
10.16 Demonstrate operation of main engine overspeed shut down cylinder by adjustment of relay.

System no. 11 *Shafting*

11.1 Prove bearing and sterntube temperature sensors at alarm set point.

System no. 12 *Steam system*

12.1 Manufacturer to prove operation and safety features of oil-burning installation.
12.2 Demonstrate remote manual control of main engine exhaust by-pass valve.
12.3 Prove operation of oil-burner installation cut in limit switches on positioner during 12.2.
12.4 Demonstrate maintaining boiler steam pressure by automatic control of main engine exhaust by-pass valve by positioner on trial.
12.5 Repeat 1.2 (steam-pressure switch).
12.6 Reduce boiler pressure to demonstrate low-pressure alarm.
12.7 Repeat 1.2 (feed pump).
12.8 Repeat 1.3.
12.9 Prove low-level alarm.
12.10 Prove extra low-level shut down.
12.11 Feed pump discharge to boiler proving remote water-level indicator by comparing with gauge glass.
12.12 Prove high-level alarm during 12.11.
12.13 Prove remote pressure gauge comparing with local pressure gauge during all functions.
12.14 Prove flame failure alarm by shutting off oil fuel to burner.
12.15 Prove remote oil-burner selector switch in control console.
12.16 Prove boiler-room fire alarm and emergency oil fuel shut off by melting fusible link.
12.17 Prove salinometer on feed discharge to boiler.
12.18 Prove salinometer and dump valve on distiller returns.

System no. 13 *Purifier system*

13.1 Motor fail each purifier. Remote stop each purifier.

13.2 Demonstrate steam valves maintaining oil temperature at—°F on oil fuel and lubricating oil purifiers.

13.3 Prove switches by alternately shutting isolating valves and opening by-pass valves as follows:
Oil fuel purifier temperature low.
" " " " high.
Lubricating oil purifier temperature low.
" " " " high.

13.4 Prove level alarm in sludge tank while purifying.

System no. 14 *Bilge system*

14.1 Prove level alarms in engine room port and starboard and tunnel bilge wells by filling with water.

14.2 Start up bilge pump and pump each bilge well separately demonstrating control-valve selector.

14.3 During 14.2 confirm vacuum-gauge function.

14.4 Reduce vacuum to control priming-system pressure switch and demonstrate alarm point—in. mercury

14.5 Reduce central priming system vacuum prove alarm and auto start pressure switch.

14.6 Prove level alarm in slop tank.

14.7 Prove level alarm in hold bilges.

System no. 15 *Distilled water system*

15.1 Prove level alarms in following tanks:
Boiler feed make up tank—low.
Engine room reserve tank—low.
Boiler distilled water storage tank—high and low.

System no. 16 *Domestic systems*

16.1 Prove the following:
Domestic fresh-water tank—low.
Domestic sea-water tank—low.
Sewage tank—high.

System no. 17 *Washdeck system*

17.1 Prove pressure gauge in control console with local pressure gauge.

System no. 18 *Turbo alternator*

18.1 Prove thrust bearing high-temperature alarm.

18.2 Prove alternator high air-temperature alarm.

18.3 Adjust vacuum pressure on pressure switch to demonstrate alarm point—in. mercury, falling.

18.4 Prove low vacuum alarm by reducing sea-water circulation to condenser.

18.5 Prove high-level alarm in condenser.

18.6 Repeat 1.2 for oil-pressure switch.

18.7 Prove lubricating oil alarm by running down alternator.

System no. 19 *Diesel alternators*

19.1 Demonstrate the following:
Alternator high air-temperature alarm.
Engine lubricating oil low-pressure alarm.
Emergency shut down.
Demonstrate sea-water valve opening.

System no. 20 *Crankcase mist detector*

20.1 Demonstrate alarm and reset.

System no. 21 *Engine bearing wear down*

21.1 Demonstrate wear-down alarm in each crank space.

System no. 22 *Engine room fire*

22.1 Prove alarm at main engine by melting fusible link.

22.2 Repeat 22.1 at diesel alternators.

System no. 23 *Bridge emergency stop*

23.1 Demonstrate switch on bridge shutting down main engine.

23.2 Demonstrate switch in control console over-riding 23.1.

System no. 24 *Piston-rod gland drain*

24.1 Prove level alarm in tank.

20.2. MAINTENANCE

Once a control system is functioning correctly there is no test or tests which can be applied to guarantee successful operation for a stated period of time. A control system is an assembly of components and random failures can occur; these can never be forecast. Consequently, satisfactory operation in service is the only yard-stick to apply. This involves frequent checking of the system in accordance with the instruction book supplied by the manufacturer.

Thus for satisfactory continuous operation any controller or control system should receive periodic inspection and careful intelligent maintenance. Dirt, corrosion and wear are the enemies of control. This means that clean conditions must be effected in the ship before the equipment is installed. It is pointless to install control equipment and then to discover, at a later date, that spray painting or installation

of asbestos or other thermal insulation is taking place in the same compartment. The covering of the control equipment with tarpaulins or with plastic sheeting is no cure for such a hazard. Compartments destined to contain control equipment should be completed in every sense including painting before the control equipment is installed.

The main machinery as well as the control system requires adequate maintenance. Every experienced engineer knows that many cases of unsatisfactory performance can be traced to faults in the main machinery. Many operators are prone to overlook this point and spend much time investigating the control system rather than the machinery itself. A partially blocked line or a clogged burner causes difficulty just as readily as a dirty air supply or a faulty electrical circuit.

Pneumatic and hydraulic components require very little maintenance. The various pivots and links in the system should be free of friction and in good working order. Oil must never be applied to a pneumatic system except by specific instruction from the manufacturer.

Connections in pneumatic and hydraulic systems should be tight. Loose connections not only waste air or oil but also produce unsatisfactory operation. Connections should always be checked for leakage.

In order to eliminate the presence of foreign material inside the lines, particularly air-supply lines, non-ferrous materials such as copper or brass should be used. Steel pipe will ultimately rust and small solid particles will cause future difficulty.

The greatest problem with pneumatic systems is the maintenance of a supply of clean, dry air at constant pressure. Moisture, oil or foreign particles carried into the system from the air supply will cause trouble. Pneumatic controllers operating with clean, dry air require virtually no maintenance or cleaning.

Moisture is usually adequately removed with a storage tank of proper size and a compressor after cooler. Oil should be prevented from entering the system. The air compressor should never be overloaded since it will pump more oil when running at high loads.

Moisture-free compressed air does not freeze even at sub-zero temperatures. Therefore, effective removal of moisture also helps in the problem of frozen air lines. Similarly air lines should be located to avoid low-temperature areas such as the upper deck.

The importance of maintaining a very high degree of cleanliness when working on control systems cannot be over-emphasized. Because of the fine clearances involved in pneumatic and hydraulic components it is imperative that the system remain free of contamination. The same need for cleanliness applies to electrical systems since dirt generates electrical noise. To prevent dirt entering a system the following precautions should be taken before any work is commenced:

(a) Thoroughly clean the outside of fittings, pipes, components and the surrounding structure before disconnecting any pipes.
(b) When pipe lines are disconnected fit dust caps to pipes and unions, mask any exposed surfaces immediately and make certain that it is not possible for foreign matter to enter the system. If suitable capping cannot be provided by plugs or caps, place a plastic bag over the pipe or connection and secure with an elastic band.
(c) When drying pipes or components use only clean, dry compressed air, i.e. from a bottle and not from a compressor.
(d) When flushing pipes use clean carbon tetrachloride from a clean container.
(e) Maintain clean hands and tools.

With instruments no repair should be carried out locally which can be done more efficiently centrally. Only minor servicing, e.g. zero adjustment and functional checking, should be done *in situ*. When repairs or re-calibration are needed the instrument should be removed and replaced by another so that maintenance can be carried out in a workshop.

The workshop equipment should be selected on the principle of "testing not manufacture". It must be possible to check that an instrument is reading correctly but the instrument manufacturer should be regarded as the repairer or the provider of replacement parts. A generous allowance of space (more than is normal for a

fitting or machine shop) is amply justified by the consequent ease and tidiness of operation. The shop itself should be dry, warm and clean.

If an efficient record system is used then information will be collected to show the amount of servicing required. Full records should be kept on all control equipment. Often it is necessary to know the date of an installation, the results of a calibration check or whether any renewal of parts has taken place. Difficulties with instruments can often be traced to the omission of cleaning of a filter or some similar detail.

The provision of block diagrams of control systems and the identification of components is vitally important. Such block diagrams should be inside panel doors or other convenient location. Too often is a system at fault and there is no means of identifying components. Control valves can be mounted the wrong way round and without an external indication of flow it is a long tedious process to discover the fault and carry out rectification.

Glossary of Terms used in Automatic Control

THE following terms and definitions have come into use in control engineering. These definitions are offered subject to revision by B.S.I. and A.S.M.E.

Actuator. A servo motor producing a limited output motion.

Automatic Controller. A device in which a signal from the detecting element is compared with a signal representing the set value and which operates in such a way as to reduce the deviation.

Automatic Control System. A control system in which the measured value of a controlled condition is compared with a set value and a correction dependent on their difference is applied to the correcting condition in order to adjust the controlled condition, without human intervention in the closed loop formed by the comparing and correcting chains of elements and the process.

Cascade Control System. A control system in which one controller alters the set value of one or more other controllers.

Chopper. A device for periodically interrupting or reversing a d.c. signal to enable it to be amplified by an a.c. amplifier.

Comparing Element. That part of the automatic controller which generates a signal proportional to the deviation by comparing the signal from the measuring element with the signal representing the set value.

Continuous Controller Action. The action of a controller whose output signal is a continuous function of the deviation.

Control Equipment. The equipment necessary for controlling the process; it includes the detecting element, the automatic controller and the motor element of the correcting unit, which, together with the plant, make up the automatic control system.

Control Point. Selected reference value of controlled variable which it is desired to maintain.

Controlled Variable. Quantity or condition which is measured and controlled.

Controller Action. The relationship between the deviation and the change of output signal from the controller.

Controller Adjustment. Manually adjustable characteristic of an automatic controller for varying relationship between controlled variable and controller response.

Controller Lag. Retardation or delay in response of final control element to change in controlled variable at the controller.

Controller Response. Output signal or impulse from an automatic controller.

Corrective Action. Controller action initiated by deviation and resulting in variation in the manipulated variable.

Cycling. Periodic change of controlled variable from one value to another.

Dead Time. The time interval between a change in a signal to the element or system and the initiation of a perceptible response to that change.

Dead Zone. The zone within which a change of value of an input signal to an element or system may take place without causing any perceptible change in output signal.

Derivative Controller Action. The action of a controller whose output signal is proportional to the rate at which the deviation is changing.

Desired Value. The specified value of the controlled condition.

Detecting Element. The element which responds directly to the value of the controlled condition.

Deviation. The difference between the measured value of the controlled condition and the set value.

Discontinuous Controller Action. The action of a controller whose output signal is a discontinuous function of the deviation.

Distance Velocity Lag. The time interval between an alteration in the value of a signal and its manifestation, unchanged at a later part of the system and arising solely from the finite speed of propagation of the signal (see Dead Time).

Drift. Wandering of controlled variable in which its value aimlessly departs from control point.

Error. See Deviation.

Final Control Element. Portion of controlling means which directly determines the value of manipulated variable.

Hunting. See Cycling.

Integral Controller Action. The action of a controller whose output signal changes at a rate which is proportional to the deviation.

Lag. Retardation or delay of one physical condition with respect to some other condition to which it is closely related.

Load Change. Change in process conditions which requires a change in the average value of manipulated variable to maintain the controlled variable at the desired value.

Master Controller. In a cascade control system, that automatic controller which adjusts the control point of another controller.

Measuring Element. The element which responds to the signal from the detecting element and gives the measured value of the controlled condition.

Metered Control System. See Cascade Control System.

Multi-step Controller Action. The action of a controller whose output signal assumes predetermined values at two or more chosen values of the controlled condition.

Offset. Sustained deviation due to an inherent characteristic of proportional controller action.

On-off Mode. See Two-position Mode.

Open and Shut Mode. See Two-position Mode.

Oscillation. See Cycling.

Process Lag. Retardation or delay in response of controlled variable at point of measurement to a change in value of manipulated variable.

Proportional Band. That range of values of deviation corresponding to the full operating range of output signal of the controlling unit resulting from proportional action only. (Expressed in percentage of controller scale.)

Proportional Controller Action. The action of a controller whose output signal is proportional to the deviation.

Rangeability. Ratio of maximum flow to minimum controllable flow.

Sensitivity. See Proportional Band. Sensitivity is the inverse of proportional band.

Set Point. See Control Point.

Set Value. The value of the controlled condition to which the automatic control mechanism is set.

Signal. The physical quantity or change in physical quantity by which one element of a control system influences another.

Tacho Generator. A generator whose output voltage is proportional to speed.

Transfer Lag. Retardation, not delay, in response of controlled variable caused by existence of distributed capacity or two or more separated capacities in a controlled system.

Transmission Lag. Retardation or delay caused in transmitting a measurement of variable from primary element to controller.

Transportation Lag. See Dead Time.

Turndown. Ratio of normal maximum flow to minimum controllable flow, through a final control element.

Two-position Mode. Controller action in which a final control element is moved from one of two fixed positions to the other at predetermined values of the controlled variable.

Two-step Controller Action. The action of a controller whose output signal changes from one predetermined value to another when the deviation changes sign.

Two-step Controller Action with Overlap. The action of a controller whose output signal alternates between two predetermined values when the value of the controlled condition passes from one to the other of two chosen values. The difference between these values determines the overlap, which may be adjustable.

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