Programme Relays: The correct sequence operation is arranged for the various actions, for example, manoeuvring conditions satisfied before moving on to say valve opening sequences, etc.

Timing Relays: To prevent excessive speed changes, by too rapid signals, which would endanger engines and boilers. For example, half speed in say $\frac{1}{2}$ min, emergency swings from say full ahead to full astern allow astern braking steam usage, rotation slightly ahead and astern at say 3 min intervals of a sustained stop of engines, etc. (manoeuvring valves now usually can be operated sequentially).

Essential Safety Locks: Override on timing by such elements as low boiler water drum level, low turbine inlet steam pressure, etc.

Emergency Control: Direct hand control of manoeuvring valves camshaft.

Local Control: Independent power control at the actuators themselves.

Feedbacks: The steam pressure feedback gives accurate positioning for pressure of the actuator. Speed feedback is arranged so that a difference of speed between measured and desired values causes an additional trimming signal to the controller. This may be necessary as pressure and speed are not well correlated at low speeds.

Outline description

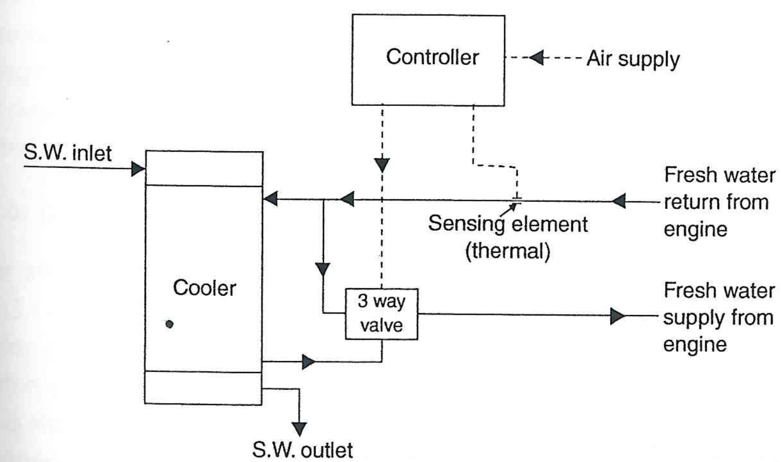
The following is a brief description of one type of electronic bridge control for a large single screw turbine vessel to illustrate the main essentials. Movement of a control lever modifies the output of an attached transmitter (electronic signal 0–10 mA dc). The transmitted signal is passed, via override, alarm and cut out units to the desired flow module which is connected to a time relay and feeds to the controller. The electronic controller compares desired speed with actual speed as detected by a tachometer generator and dc amplifier. The correct controller signal is passed to the manoeuvring valve positioner from which a return signal of camshaft position is fed back to the dc amplifier, thus giving the command signal to the actuator reversing starter. The two control levers are independent and do not follow each other; an engine room override of bridge control is supplied. The rate of valve opening is controlled by the actuators so that too rapid valve opening is prevented by a time delay, full normal valve operation shut to open, or vice versa, occurs in about 1 min, this can be reduced to about 20 s in emergency by full movement of the telegraph from full speed direct to or through stop. A near linear rev/min to control lever position exists. Auto-blast refers to the automatic time delay opening of the ahead manoeuvring valve for a short period after a certain length of time has lapsed – this has an override cut out for close docking, etc.

IC Engine Plant

IC engine plants now utilise general (mainly thermal) auto-control to a great extent and manoeuvring systems, etc., have always involved fairly detailed devices. The efficiency is virtually inherent in the design, however auto-control can still give improvements to efficiency of operation.

Jacket (or piston) temperature control – single element

Refer to Figure 13.8. The sensing element may be on the supply or return line to the engine, and there are certain advantages for each case. The three-way valve (two entry and one exit) varies the re-circulation or supply to the cooler and functions as a mixing valve. This valve may be of the rotary cylinder type or diaphragm operated type. Integral action is usually incorporated in the controller as offset may be appreciable otherwise (up to 9°C). This system is often preferred. An alternative arrangement is to throttle the sea water supply for the cooler. This gives a big capacity lag in the system as one variable (sea water) controls the other variable (fresh water). Valve selection is most important. Maximum pressure and temperature, maximum flow rate, minimum flow rate, valve and line pressure drops, etc. must be carefully assessed so that the valve gives the best results. The controller shown in Figure 13.8 is pneumatic but an



electronic controller with a rotary valve is also common. Engine coolant is shown as fresh water but could equally well be oil. Most loops on a motor ship are single element, for example, temperature control of lubricating oil, fuel, air, water, etc.

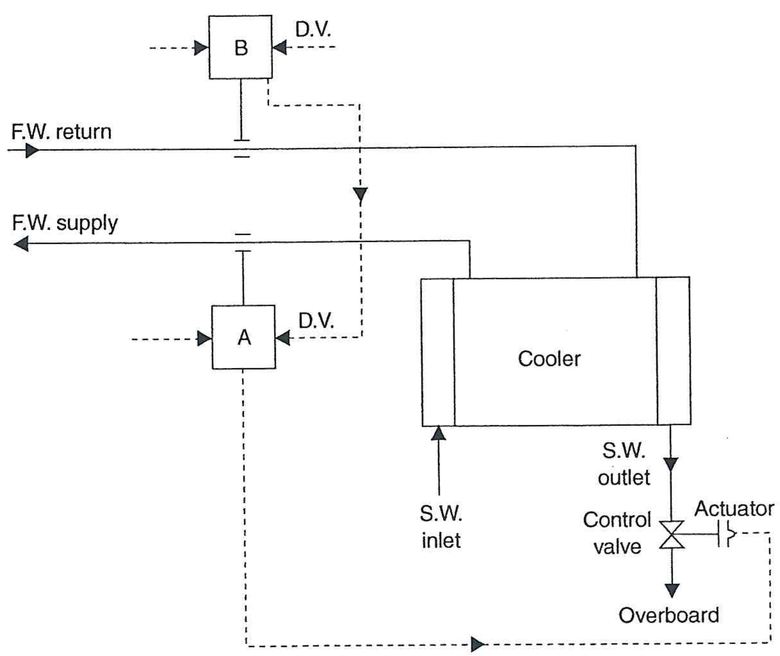
Two-element cooling loop

More sophisticated loops are sometimes found on auxiliary boiler controls (pressure, level, etc.) as well as main engine coolant.

In the latter case a two-element type may be advantageous as heat transfer rates are high and sea water temperature becomes more critical than in smaller single-element loops. The two variables involved are engine load and sea water temperature.

If the engine is considered to be at a fixed load then by reference to Figure 13.9 it is seen that the water inlet temperature is fixed by the set value of the controller A which accounts for water temperature changes (controller A is 'slave').

Now if the engine load changes the inlet water temperature should change, that is, the lower the load the higher the water temperature. This is achieved by changing the desired value of controller A according to the engine load variations. Controller



B (the 'master') provides an indication of engine load by measuring the return water temperature. Controller B signal changes the set value of controller A (cascade control). A fresh water heater may be placed in the engine supply line with heat input controlled by controller A. Split range (level) control allows heat input at low coolant temperatures and cold input at high coolant temperatures. The system works equally well for oil coolants. The actuator can be operated by local or remote controls with the controllers out of operation.

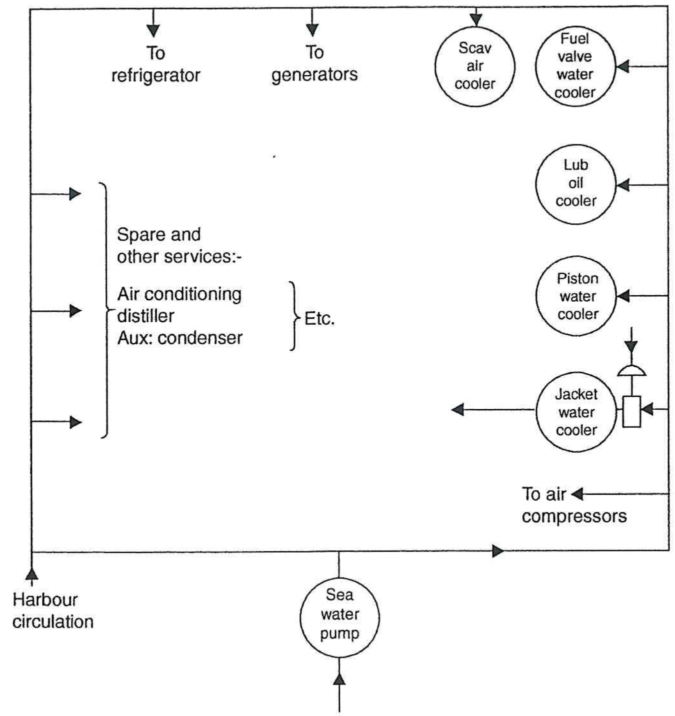
Overall coolant system control

The sketch given previously in Figure 13.8 for jacket (or piston) coolant temperature control is typical. Similar systems can apply almost exactly for lubricating oil, turbo blowers, scavenge air, fuel valve, etc. coolants. Each system is controlled separately so that adjustments can be made to individual systems without affecting the others.

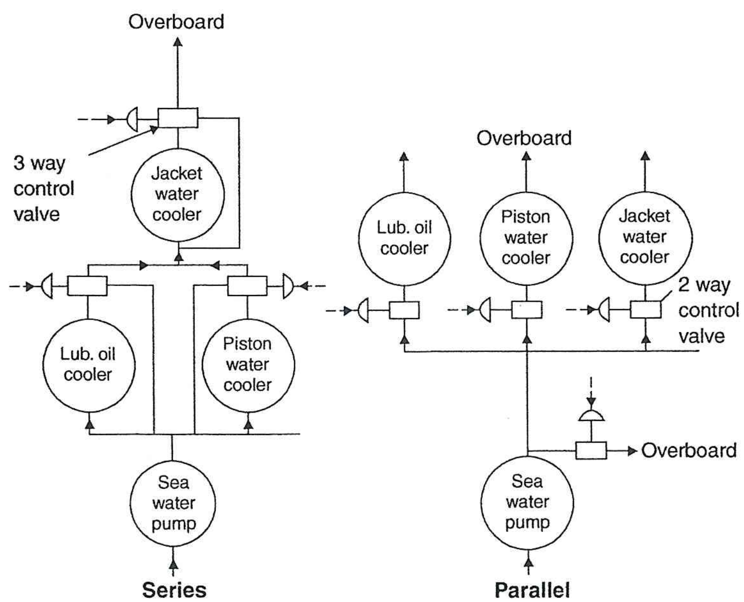
One obvious arrangement for sea water supply is to utilise independent tappings (both main engines and auxiliaries) from a continuously circulated *ring main* – in this design individual sizes and flow rates require very careful investigation. A typical ring main system is given in Figure 13.10. For simplicity no duplication of coolers, indication of individual valves, strum or strainer boxes, etc. is shown. Note that only main engine circuits are controlled in this given case. It is advisable to have say four pumps capable of operation on the main. The pumps preferably should be divided with say two constant speed and two controllable speed to allow flexibility; harbour circulation is best arranged by a feed into the main from say the ballast pump. The individual control is shown on the jacket cooler only for illustration. The control here is direct control of the sea water quantity utilising two-way diaphragm control valves. All main engine circuits shown in the diagram are so controlled and any auxiliary circuits can be similarly controlled if required. Certain discharges, where convenient, can be combined to reduce the number of shipside valves.

Series or parallel arrangement

Another alternative is a series or parallel arrangement. The two diagrams given in Figure 13.11 illustrate series circulation with three-way valves with bypass, and parallel circulation with two-way valves with direct supply; the latter arrangement requires a satisfactory system pressure control such as control flow from pump or dump of excess water (as shown). Grouping of the coolers and choice or combination of systems can be arranged dependent on flow and temperature considerations. No duplication is shown.



▲ **Figure 13.10** *Overall coolant system control (ring main)*

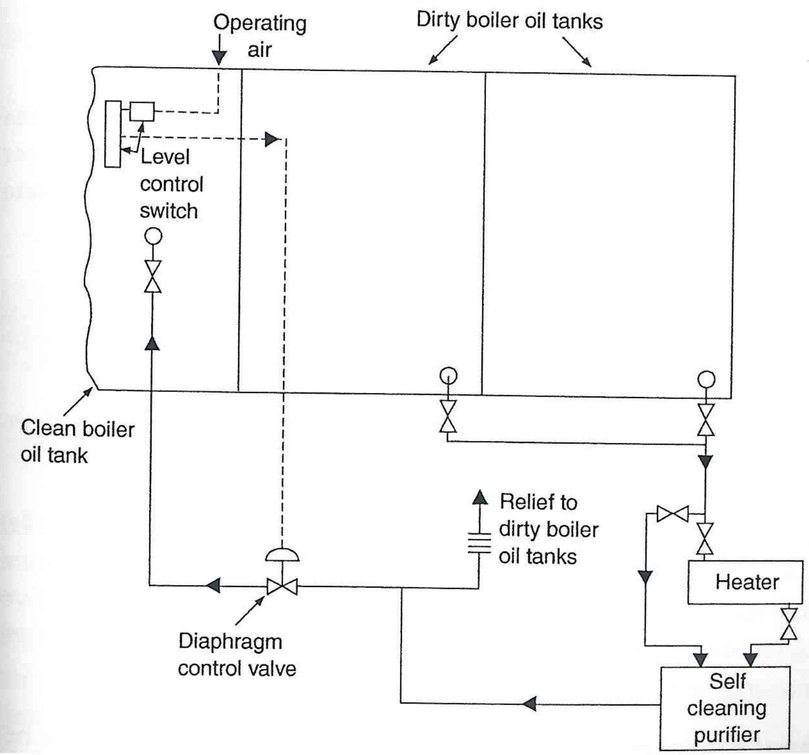


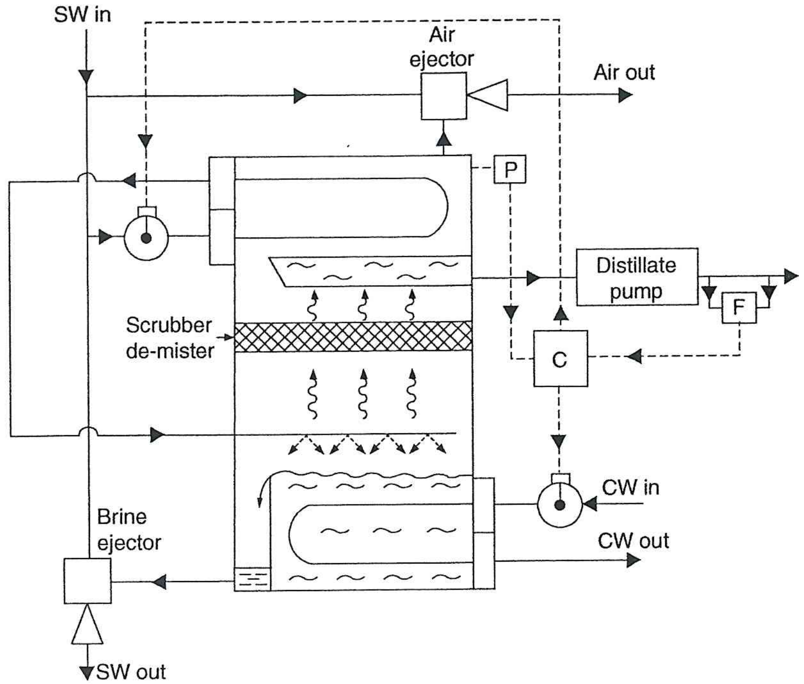
Boiler oil purification control system for IC engine

This system shown in Figure 13.12 is designed to maintain a working level in the boiler oil service tank to the main engine. The oil supply from the dirty oil tanks continuously passes through a self-cleaning purifier. The oil fuel heater (at purifier, or if also provided on the main engine supply rundown) can easily be arranged to give fixed oil temperature or controlled viscosity in a similar manner to the previously described temperature flow control system.

Waste heat flash evaporator control

This system is shown in Figure 13.13. Waste, or low grade heat, from engine coolant has good energy potential. Fresh water jacket (or piston) coolant evaporates sea water in the second stage heat exchanger which is condensed at about 0.1 bar in the first stage pre-heater and removed by the distillate pump.





▲ **Figure 13.13** *Waste heat flash evaporator control*

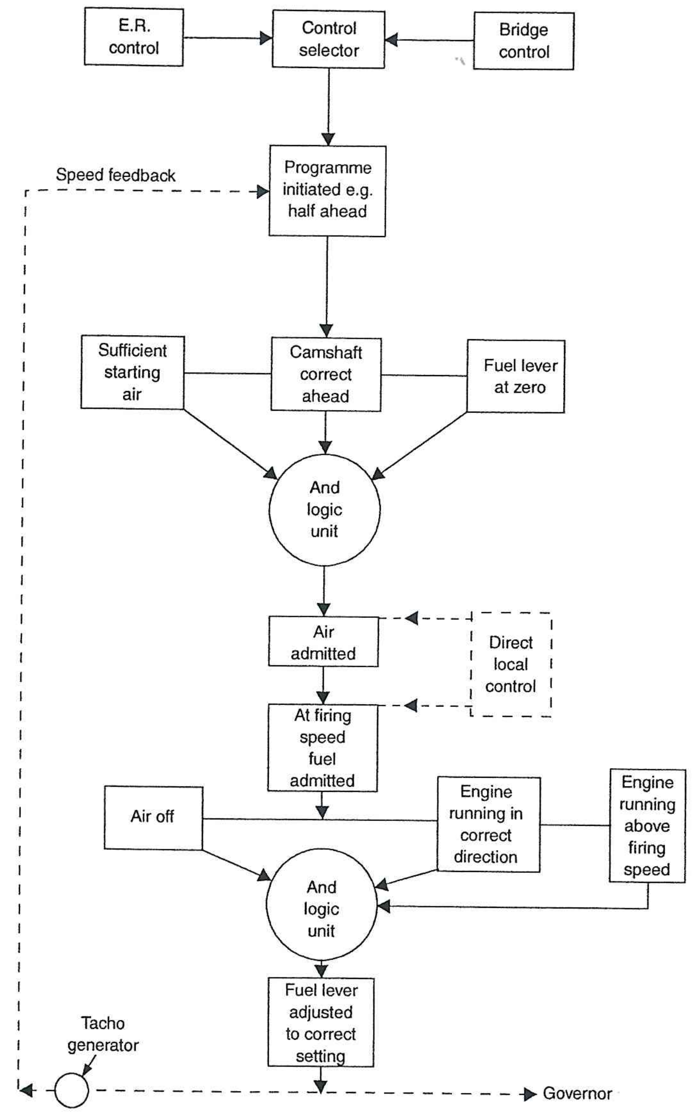
When the pressure sensor-transducer (P) allows the controller (C) to operate the sea and coolant inlet valves, vapour production starts. Control is by measurement of water flow at the flow sensor transducer (F) and the signal allows the controller to regulate water inlet valves accordingly.

Bridge control (direct reversing IC engine)

It is suggested that the section on bridge control for turbines should be read again as there are many obvious similarities.

Instrumentation and Alarms: Minimum usage, suggested alarms (bridge console): (1) Low starting air pressure, (2) lubricating oil discharge pressure and temperature, (3) cooling water discharge pressure and temperature, (4) tank contents level gauge, (5) fuel oil discharge pressure and temperature, (6) scavenge belt pressure. A further six instruments could be provided.

Note: All normal protective devices are assumed, subsidiary control loops are not



▲ **Figure 13.14** *Bridge control (direct reversing Ic engine)*

Refer to Figure 13.14 and consider the following:

Selector: Bridge or engine room control is in the engine room. With one selected the other is inoperative.

Duplication: Both transmission control systems are identical master and slave functions as selected.

Programme and timing relays: Consider say a requirement of 'half ahead'. The programme must satisfy a sequence such as:

1. Fuel admission check for zero.
2. Air lever to ahead.
3. Sufficient air and camshaft direction checks.
4. Air admitted.
5. Adjustable delay period for firing speed.
6. Fuel admitted.
7. Delay checks for air off, correct direction, rotation above firing speed.
8. Adjust fuel for set value speed.

Essential safety locks: Override on timing by such elements as low lubricating oil pressure, low cooling water pressure, etc.

Emergency and local control: Directly at the engine controls themselves. Further protective considerations:

1. Speed governor.
2. Non-operation of air lever during direction alteration.
3. Failure to fire requires alarm indication and sequence repeat with a maximum of say four consecutive attempts before overall lock.
4. Movement of control lever for fuel for a speed out of a critical speed range if the bridge speed selection is within this range.
5. Emergency full ahead to full astern, etc., actions, must have time delays to allow fall of speed before firing revolutions, astern air open, engine stop, correct astern timing and setting.

Outline description

The following is a brief description of one type of electronic-pneumatic bridge control for a given large single screw direct coupled IC engine to illustrate the main essentials. The IC engine lends itself to remote control more easily than turbine machinery.

Movement of the telegraph lever actuates a variable transformer so giving signals to the engine room electronic controller which transmits, in the correct sequence, a signal series to operate solenoid valves at the engine. One set of solenoid valves controls starting air to the engine while a second set regulates fuel supply; the latter, via the manual fuel admission lever, is coupled to a pneumatic cylinder whose speed of travel is governed by an integral hydraulic cylinder in which rate of oil displacement is

With the engine on bridge control the engine control box starting air lever is ineffective and the fuel control rack is held clear of the box fuel lever. Normal fuel pump control is eliminated and fuel pressure, in a common rail, is automatically adjusted to speed and load by a spring loaded relief valve. Engine override of bridge control is provided.

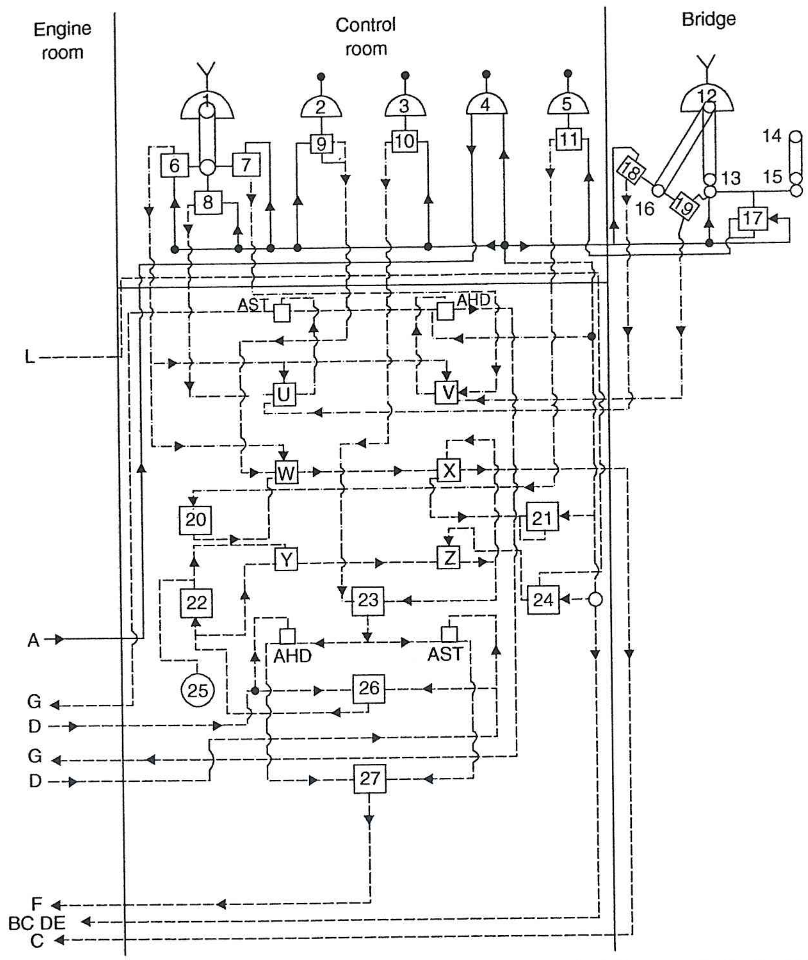
The function of the electronic controller is to give the following sequence for say start to half ahead: Ensure fuel at zero, admit starting air in correct direction, check direction, time delay to allow engine to reach firing speed, admit fuel, time delay to cut off air, time delay and check revolutions, adjust revolutions. Similar functions apply for astern or movements from ahead to astern directly. Lever travel time to full can be varied from stop to full between adjustable time limits of $\frac{1}{2}$ min and 6 min. Fault and alarm circuits and protection are built into the system.

Remote control (detail – Ic engine)

Pneumatic reversing and starting systems have been an integral part of IC engine equipment for many years. Figure 13.14 and description are fairly general and it is now appropriate to extend this to a more detailed arrangement.

Consider Figure 13.15 and the following nomenclature: (1) telegraph receiver; (2) speed lever; (3) start lever; (4) shut off lever; (5) bridge maximum speed lever; (6, 7, 8, 9, 10, 11) selector pilot valves; bridge or control room, ahead, astern, speed, start, maximum speed; (12) telegraph transmitter; (13) speed cam; (14) programme motor drive; (15) speed cam (fine control); (16) control cam ahead-astern; (17, 18, 19) selector pilot valves; speed, astern, ahead; (20, 21, 22) control valves; flow, speed, flow; (24) solenoid valve; (25) timing volume; (23, 26, 27) double check valves. There are ten relay valves, two each ahead and astern plus, U, V, W, X, Y, Z. A is air supply, B outlet lifts engine speed lever handgrip out of gear to allow remote operation, C outlet to speed set, D from and to direction interlocks, E outlet to reversing interlock, F outlet to starting servo, G outlet to reversing servo, L to solenoid valve.

With the engine telegraph set at remote control position the control air at a pressure of about 8 bar flows through line A to all nine pilot valves (as shown full lines) if lever 4 is opened. There is an additional section on the engine telegraph for remote control and also on the control room telegraph for bridge control position. As 4 is open air passes through B, C, D, E to operate interlocks. Air also passes to the upper ahead or astern direction selector relay valves, partly clears speed set relay valve X for operation (via 21) and clears the starting routine at X and Z via 24 (from line L).



▲ **Figure 13.15** *Pneumatic remote control system*

Operation of 1 selects either control room or bridge and output from six loads either U or V accordingly. Ahead lines 7, 19 lead to V and 1 as appropriate feeds top ahead relay valve and goes out via line G to position the reversing servo at the engine (astern similarly through 8, 19 via U and top astern relay valve to other line G).

Speed set from 2, 9 leads to W (directly loaded from control room) (via 6 or from bridge via 17, 5, 11, 20) thence to X and line C to act on speed set servo.

Start from 3, 10 through 23 and either of lower ahead-astern relay valves past 27 via line F to engine starting cylinder servo (depending on loading signal from either line D direction interlocks). 23 is also cleared from 26, through Y and Z, with built in flow and

The engine will, for example, reverse, start and gradually reach selected speed in an automatically programmed sequence. Two fine set speed buttons allow speed variation from fixed values. The system has built in avoidance of critical speed ranges and acceleration to full speed when on bridge control. This pneumatic system is readily adaptable to electronic signals to servos.

General Plant

General control applications have always been used and the ship steering gear control system is perhaps a classic example. Early control components included the safety valve and watt governor. A selection of typical systems is now given.

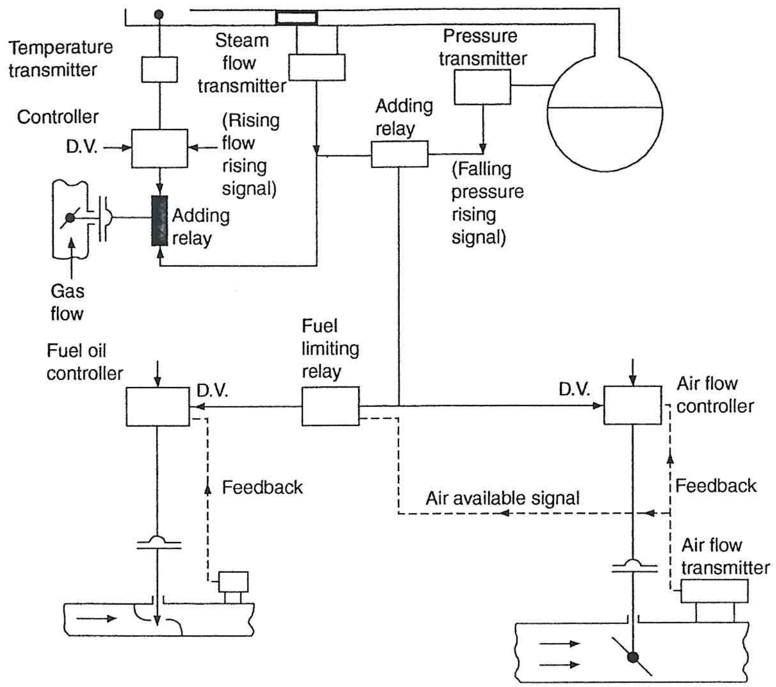
Auto-combustion and attemperator control system

For simplicity no feed water control or similar controls are shown in this system but they are commonly fitted in practice (Figure 13.16).

For main propulsion steam turbine drive wide-range fuel oil burners are a very desirable feature with automatic combustion control as they allow manoeuvring without the need to change burners. Burners must have a high turn down ratio. Mechanical atomisation burners have a turn down ratio of about 2 : 1 whereas steam atomisation assisted burners have a turn down ratio of about 20 : 1. The 'steam' burners however induce a water loss of about 0.75% of steam flow. The ideal arrangement is to utilise 'steam assisted' burners during manoeuvring and 'mechanical' burners at full power. This gives full automatic control over the manoeuvring range. An automatic dumping valve to the condenser ensures a nominal steam flow when the main turbines are stopped. Until recently automatic combustion control was only used at full power, in fact the main need and modern application is when load changes, that is, manoeuvring transients.

For IC engine plant especially in view of the Clean Air Act, combustion control of boilers is virtually essential.

Figure 13.16 is fairly simple in principle and can be considered suitable for auxiliary boiler practice in motor ships. The detail should be taken in conjunction with the following system on the more fully automated auxiliary automatic boiler.



▲ **Figure 13.16** *Auto-combustion and attemperator control system*

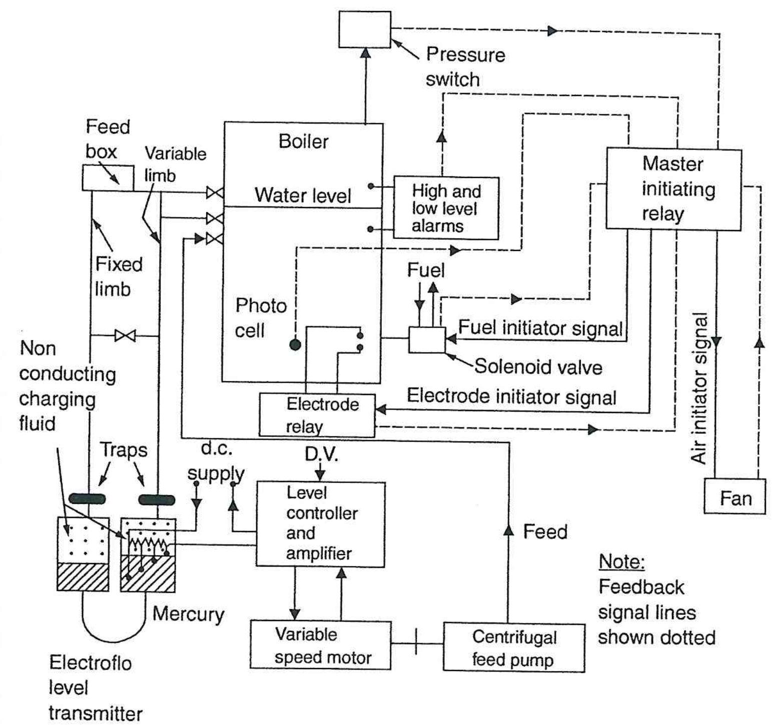
Refer to Figure 13.16. The common signal at the adding relay for pressure and flow adjusts the desired values of both air and oil flow systems. As both these loops are independent sub-groups of a closed nature they will keep changing respective flow rates until the desired value (or set value) is reached. Under normal operation steam flow signals are in advance of steam pressure signals and this anticipatory action is good control practice. The fuel limiting relay in the oil sub-loop (fuel–air ratio control) receives the feedback signal from the air flow rate and compares the required fuel rate with air availability. Fuel rate cannot be increased until an increased air flow signal allows the alteration of the desired value setting at the fuel controller.

Attemperation also has anticipatory control in that the load change signal acts before the temperature change signal. The function of the attemperator is to limit steam temperature at low boiler loads; there is a consistent relation between the two. Command signal choice between fuel pressure, fuel flow, fuel flow/steam flow, etc. show significant variations in response characteristics and research still proceeds.

Auxiliary boiler automatic fuel and control system

Refer to Figure 13.17 for the *lighting sequence.*

1. The pressure switch initiates the start of the cycle. The switch is often arranged to cut in at about 1 bar below the working pressure and cut out at about 1/5 bar above the working pressure (this differential is adjustable).
2. The master initiating relay now allows 'air-on'. The air feedback confirms 'air-on' and allows a 30 s time delay to proceed.
3. The master now allows the arc to be struck by the electrode relay. The 'arc made' feedback signal allows a 3 s time delay to proceed.
4. The master now allows the fuel initiating signal to proceed. The solenoid valve allows fuel on to the burner. The 'fuel on' feedback signal allows a 5 s time delay to proceed (this may be preceded by a fuel heating sequence for boiler oils).
5. The master now examines the photo-electric cell. If in order the cycle is complete, if not then fuel is shut off, an alarm bell rings and the cycle is repeated.



Refer to Figure 13.17 for *emergency devices*: Obviously failure of any item in the above sequential cycle causes shut down and alarm. In addition the following apply:

a. High or low water levels initiate alarms and allow the master to interrupt and shut down the sequential system.

b. Water level is controlled by an electroflo type of feed regulator and controller. Sequential level resistors are immersed in conducting mercury or non-conducting fluid, so deciding pump speed by variable limb level. The fixed limb level passes over a weir in the feed box.

Drum level controller (feed regulator)

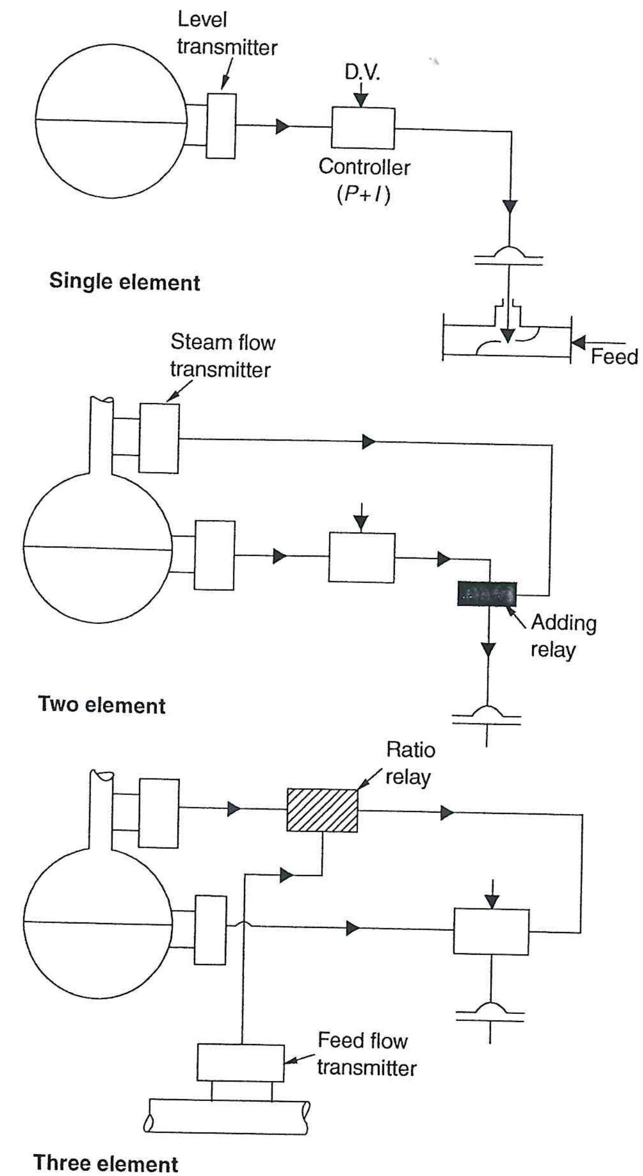
Robot feed regulators are proportional controllers (single term) working on a fairly sensitive proportional band. Due to drum contents 'swell' and 'shrinkage' during manoeuvring load changes, the action is temporarily in the wrong direction. This wrong action is very severe due to the narrow bandwidth and hand operation of feed checks was often necessary. Proportional action is made less sensitive, and this reduces the severity of the short term wrong way action but introduces offset. Offset is got rid of by the addition of integral action, that is, the control is two term for the single element action.

A two-element action is obtained by incorporating a steam flow measurement to reduce severe feed flow variations when manoeuvring. This signal would give an anticipatory action, which is usually desirable in all control systems. The level signal would act as a trimmer and has a wide proportional band so as not to affect the system during swell and shrinkage.

Three-element control gives the highest value of performance. Feed flow is compared to steam flow for the correct 1 : 1 ratio. If the ratio is incorrect then an out of balance signal is given to the controller. Drum level again acts as a trimming device on a wide proportional band with integral action. Single-, two- and three-element actions are illustrated in Figure 13.18.

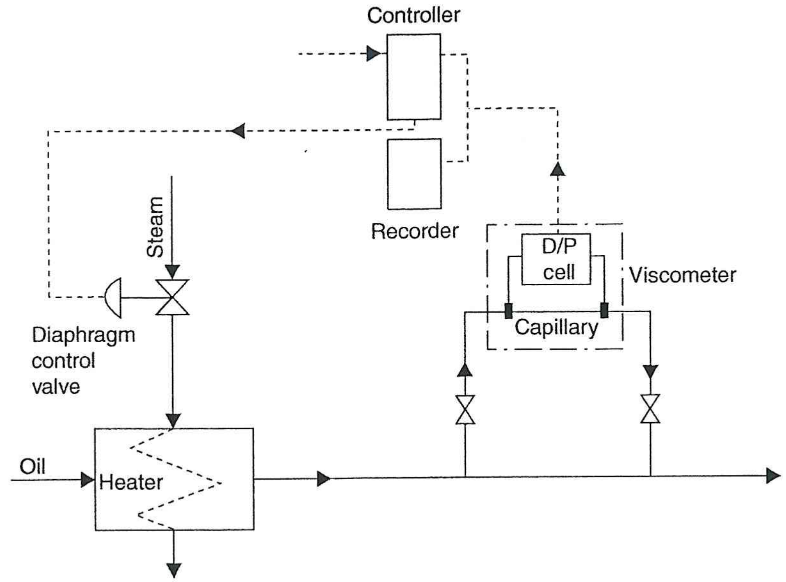
Viscosity control

Refer to Figure 13.19. A continuous sample of oil is passed across a capillary tube. The measurement of viscosity has been considered previously (Chapters 5 and 12). Flow is



▲ **Figure 13.18** *Feed regulators*

difference is sensed by D/P cell transmitter and the signal passed to a controller and recorder. The controller if supplied by air can transmit a direct power signal to operate a diaphragm control valve. This valve controls steam input to an oil fuel heater. P control is generally adequate, and rate and/or reset are easily added. The sensor has been



▲ **Figure 13.19** *Viscosity control*

Refrigeration control

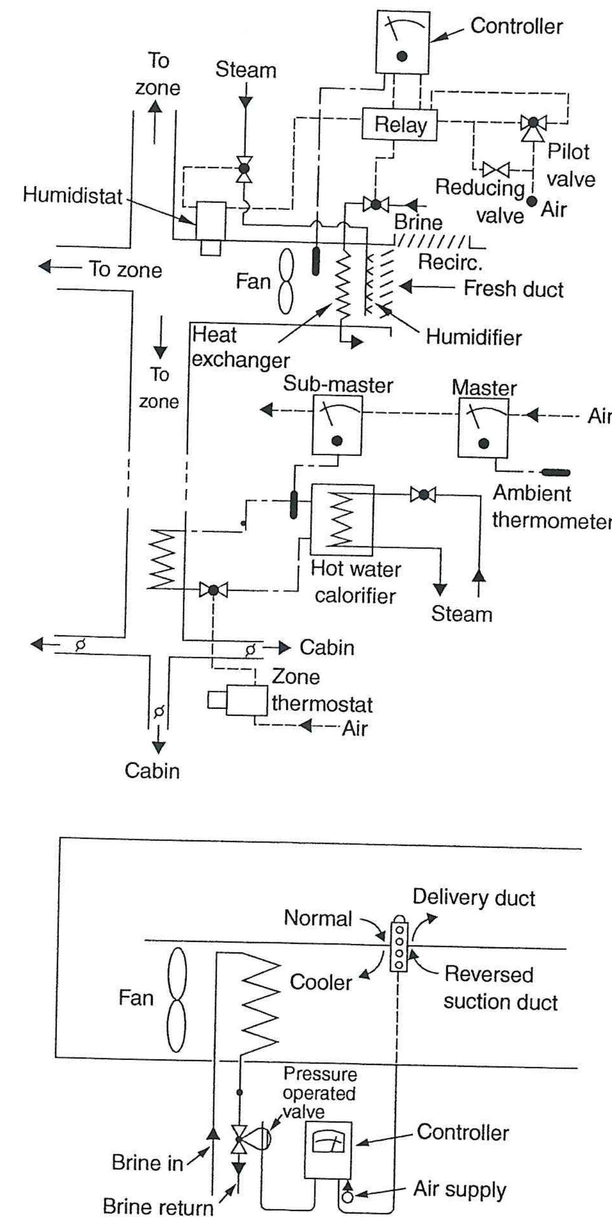
Refrigeration and air conditioning utilise a considerable degree of control. Two examples are now given (Figure 13.20).

Air conditioning (upper sketch). Air (fresh and recirculated) is control sprayed with steam to fix humidity. Air is heated (steam grid) or cooled (brine grid) with steam (or brine) and quantity controlled by temperature. The air is now passed to the various zones where sub-units adjust the air temperature to the thermostat setting of the zone. This is achieved by controlling the steam supply to a calorifier.

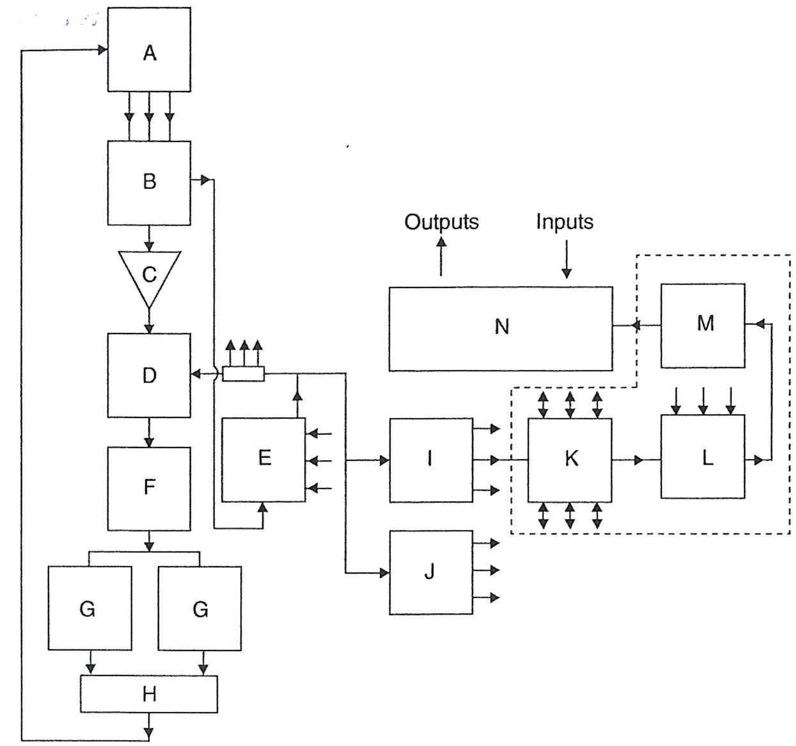
Refrigeration chamber (lower sketch). The brine quantity, for adjusting air temperature, is controlled irrespective of fan direction (suction or delivery) with the controller bulb in bypass pocket sensing air delivery temperature to chamber.

Alternator control

Consider Figure 13.21 and regard this as a main system (A–J) and sub-system (K–M).



▲ **Figure 13.20** *Refrigeration control*



▲ **Figure 13.21** *Alternator control*

Main system (load sharing)

One alternator is shown (A) of say a four-set installation. Current, voltage, power factor sensors give power computation at relay (B) which is amplified (C). Total electrical load is computed at relay (E), fractioned off to enter relay (D). The two signals at *D* are compared for load sharing, error signal triggers (F) and thyristor switches (G) for increase/decrease in speed signal to governor controller (H) with feedback loop to alternator input power. Total computed power is fed to relay (I) which functions to start another alternator at say 75% maximum rated load of those alternators in operation. The signal is also fed to stop relay (J) which is arranged to shut down an alternator when computed load is say 60%, 40% and 20% of maximum rated load for the four alternator unit.

Sub-system (alternator start)

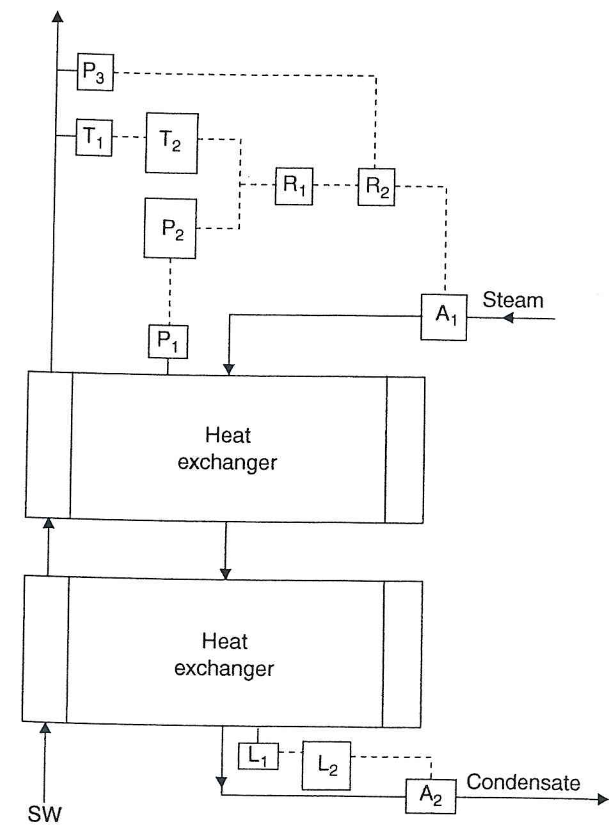
This particular sub-system is shown dotted in Figure 13.21. Obviously a start signal for another required alternator will need a pre-start routine relay (K). This would initiate

Relay (L) will arrange synchronisation (voltage, frequency, phase) and initiate a signal to close circuit breaker (M).

The main control switchboard (N) would be arranged to handle all monitored inputs from individual alternators. Standard protection from engine faults would be provided. Normal electrical protection is required, for example, reverse power trips, overload alarm (105%) and trip, preferential tripping, etc. These could be regarded as module systems within a particular subsystem. Obviously a shut down arrangement sub-system, similar to K–M, is required in which individual alternator off loading takes place and circuit breaker opening at 10% full load occurs.

Butterworth heating control

In Figure 13.22, P_1 and P_3 are pressure sensors, P_2 pressure controller; T_1 temperature sensor and T_2 temperature controller; L_1 level sensor and L_2 level controller; R_1 and R_2

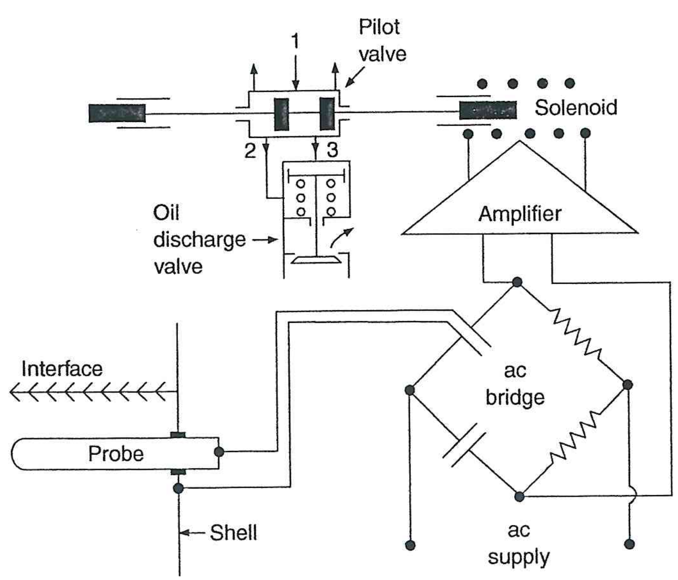


relays; A_1 and A_2 actuators. Signals of pressure and temperature enter R_1 and the lower value is passed on to R_2. Control is inherently on temperature with pressure over-ride. Signals from R_1 and the water-pressure sensor enter R_2 and the lower value is passed on to A_1. This provides protection against water supply failure. Condensate level is controlled as shown. The system can be pneumatic, electronic or a combination of the two. The control problem is essentially the difficulty of handling large quantities of water, with high velocities, maintaining close temperature control and controlling one variable by means of another variable. The system shown has utilised cascade control principles, effectively utilising temperature master reset to pressure slave, the latter being a pressure control system.

Oil–water separator interface level control

The level sensor has been described in Chapter 3. The control system of Figure 13.23 utilises two probes with the lower probe (shown) giving a balanced electrical bridge in water and the upper probe (not shown) giving balance in oil or air.

With the pump started and supplying water to the separator to rise to the lower probe level, the bridge is balanced and the solenoid de-energised. When water rises to the upper probe its bridge is unbalanced and the output signal is amplified which

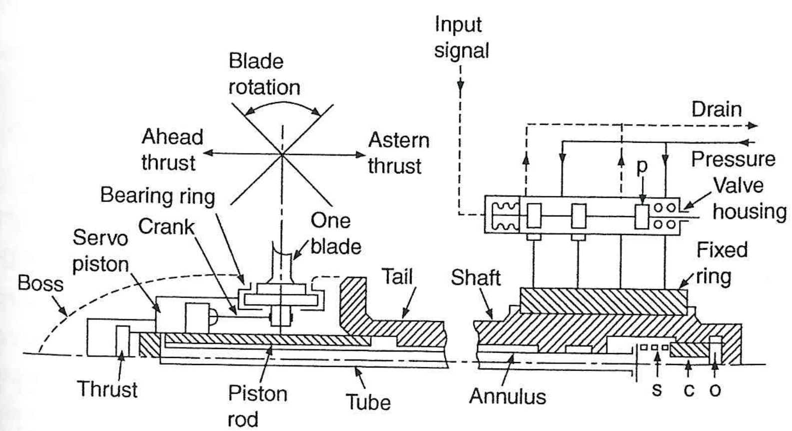


energises a 'left-hand' solenoid (not shown) which moves the pilot valve to the left. This allows clean water pressure to pass (from 1) to close the oil discharge valve (through 2). Shell pressure rises and a spring loaded water discharge valve is opened. As oil build up occurs the oil–water interface moves down, de-energises the left-hand solenoid and then energises the 'right hand' solenoid; the pilot valve moves right (as shown) and water pressure opens the oil discharge valve (through 3) and the water discharge valve closes. Each probe and valve has a signal indicator lamp and an alarm bell operates when the lower probe bridge is unbalanced. A third probe at a low level can be arranged to cut out the pump if oil falls to that point.

Controllable pitch propeller

Use of these propellers has increased with the greater use of unidirectional gas turbine and multi-diesel drives and bridge control. Engine room (or bridge) signal is fed to a torque-speed selector which fixes engine speed and propeller pitch – feedbacks apply from each. Consider Figure 13.24.

The input fluid signal acts on the diaphragm in the valve housing and directs pressure oil via one piston valve through the tube to one side (left) of the servo piston or via the other piston valve outside the tube (in the annulus) to the other side (right) of the servo piston. Movement of the servo piston, through a crank pin ring and sliding blocks rotates blades and varies pitch.

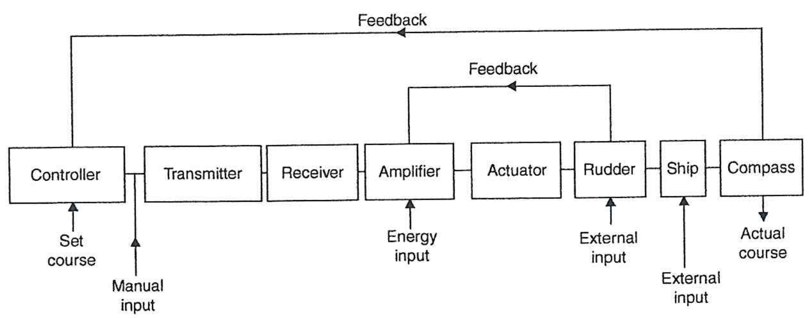


▲ **Figure 13.24** *Controllable pitch propeller*

The feedback restoring signal, to restore piston valves to the neutral position at correct pitch position, is dependent on spring(s) force (i.e. servo piston position) which acts to vary the orifice (o) by control piston (c) so fixing feedback pressure loading on the pilot valve (p) in the valve housing. The central part of the tailshaft includes a shaft coupling and pitch lock device (not shown).

Auto steering (block diagram)

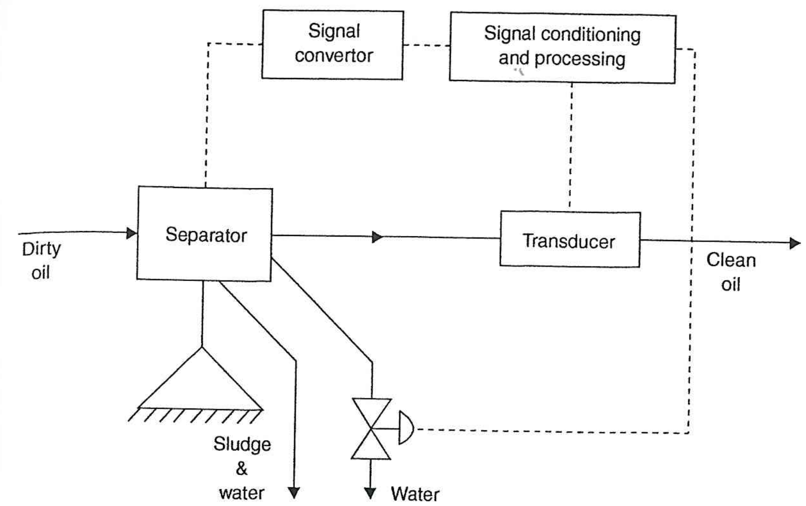
The ship steering gear (on auto pilot) utilises classic control principles best illustrated by a block diagram. Figure 13.25 shows a block diagram for auto steering. The controller will be three term with adjustment for beam sea (or wind) and dead band operation to reduce response to small random signals. Both rudder and ship are acted upon by external forces.



▲ **Figure 13.25** *Auto steering (block diagram)*

'Heavy' fuel oil separation

Consider the circuit in Figure 13.26. The self-cleaning purifier has no gravity disc; sludge and water collect on the bowl periphery. When the water build up leads to some discharging with the clean oil it is sensed (capacitance interface detection) at the transducer which signals the microprocessor. If the time since the last sludge discharge exceeds a pre-set value the sludge and water is discharged from the bowl, otherwise only water is discharged through the water valve. Essentially it works as a separator but works also as a purifier to discharge water via a paring disc when the water valve is opened.



▲ **Figure 13.26** *'Heavy' fuel oil separation*

Test Examples

1. Sketch and describe a fully automatic, oil fired, packaged steam boiler.

   Explain how it operates.

   State what attention is needed to ensure safe operation.

   Give two advantages possessed over conventional boiler installations.

2. Sketch and describe a system of control for manoeuvring a main engine from the bridge.

   Explain how control is transferred to the engine or control room upon failure of bridge control.

3. Describe with a block diagram, an automatic combustion control system, with particular reference to the methods of measuring each of the following data:

   (a) pressure,

   (b) level,

   (c) flow,

   (d) temperature.

4. Define the terms:
   (a) cascade control,
   (b) split level control.

   Discuss the application of such principles within a description, utilising sketches, of an automatically controlled lubricating oil system and an automatically controlled cooling water system of the type used in auxiliary diesel driven generators.

14 KINETIC CONTROL SYSTEMS

Kinetic Control System

A control system, the purpose of which is to control the displacement, or the velocity, or the acceleration, or any higher time derivative of the position of the controlled device. (It should be noted that forces and torques are involved in the above definition.)

Servo-Mechanism

An automatic monitored kinetic control system which includes a power amplifier in the main forward path. (Includes continuous, discontinuous, on-off, multi-step, etc. actions.)

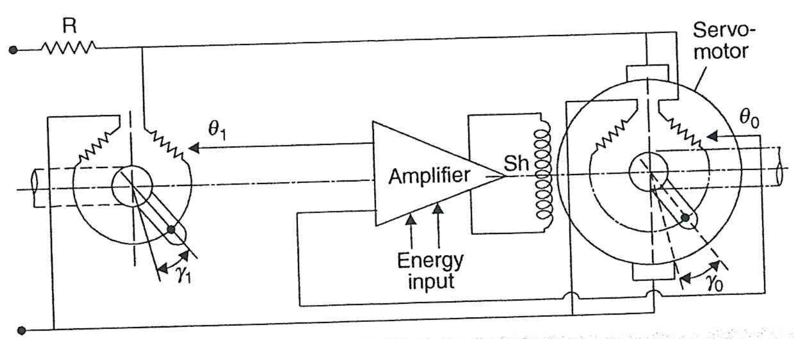
Position Systems

The control of position (displacement) in a system (linear or angular).

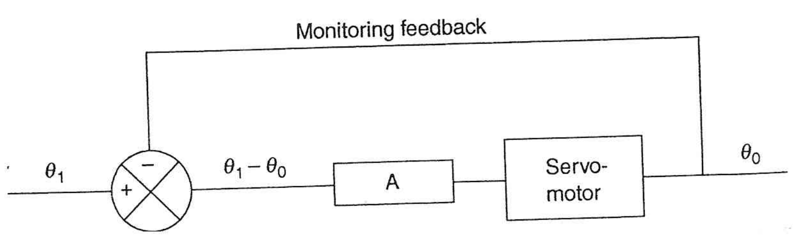
Position control servo-mechanism (dc)

Consider the electrical example of Figure 14.1 in which the output servo-motor shaft is required to follow the input shaft. As long as there is a difference of angular position between these shafts, measured by toroidal potentiometers, a difference of potential will cause current to flow in the required direction through the amplifier and servo-motor shunt field. The armature of the servo-motor carries a constant voltage (stability with large ballast resistor R) and hence a torque and motor shaft rotation occurs as soon as the field is excited.

(A constantly excited motor series field can additionally be connected across the mains if the inertia of the servo-motor is high.) As an alternative shunt field current can be arranged to be constant and armature voltage varied. The system as considered however has the true relationship that torque is proportional to actuating signal and is generally independent of rotational speed. The principle is effectively used in practical electrical ship steering gears. (Ward-Leonard system). Reference should also be made to Figure 14.2, the simple block diagram of the system. While a detailed analysis of the dynamics of such a control system is given later, at this stage it is clear that output torque is proportional to deviation, that is, proportional action. Field control is used for



▲ **Figure 14.1** *Position control servo-mechanism (dc)*



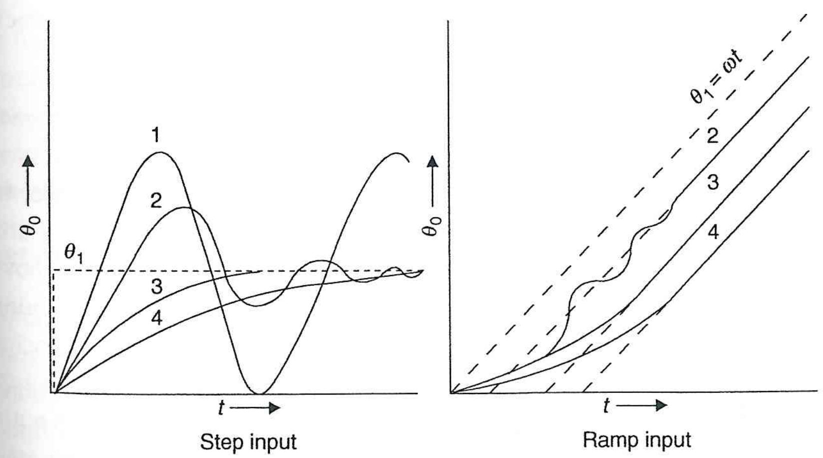
small powers and armature control for higher powers, the latter utilising a series ballast resistor (dc) or rectifier from ac supply.

System response

The effects of energy transfer including inertia, friction, etc. need to be considered whether mechanical or electrical, and so on. Descriptive analysis is often presented electrically for any system due to ease of diagram circuitry and the mechanical, or the other such, equivalents can then be simply derived.

Consider a *step input* to the servo-mechanism of Figure 14.1. At the instant of applying input, maximum deviation exists, and the servo motor first accelerates rapidly; when deviation ceases the motor stops. This is an ideal situation because in practice inertia of the motor causes *overshoot*, reverse current and rotation, with oscillation. Such oscillation would continue but for the frictional effects, static and viscous, which damp out oscillation.

Referring to Figure 14.3 for the step input θ_i curve one represents the undamped oscillation (of natural frequency ω_n) response at output θ_0. Curve two represents light damping (damping factor $k < \omega_n$), curve three critical damping (minimum time without oscillation to equilibrium $k = \omega_n$) and curve four heavy damping, that is, aperiodic ($k > \omega_n$). As a first assumption viscous friction is assumed to account for all frictional effects, with resisting friction force (or torque) proportional to velocity. Damping, to prevent overshoot, limits amplifier gain and speed response.



Also given in Figure 14.3 is output response θ_0 to *ramp input* θ_i for the three types of damped condition. θ_0 does not equal θ_i in the steady state, that is, error with output lagging input. This is a position lag (offset) due to velocity (rate) which is termed a *velocity misalignment* (friction proportional to velocity). A ramp input of position equals a step input of velocity.

Overshoot

This occurs with proportional control and is due to inertia effects. It can be reduced by any of three methods.

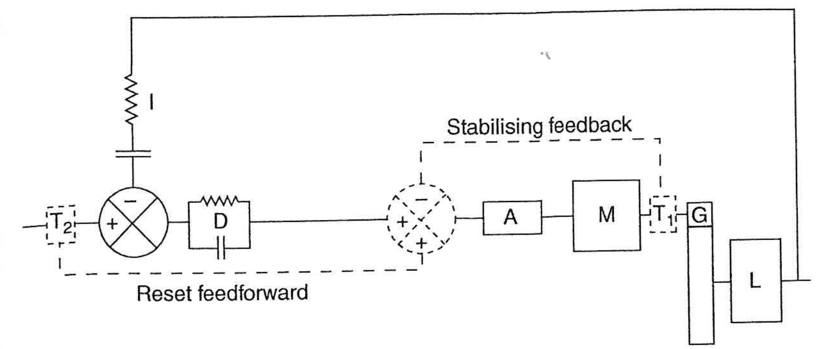
Friction damping

Friction is non-linear and regarded in two components: static (Coulomb) and viscous; the former gives steady state error and is usually neglected in simple analysis. Viscous friction is proportional to velocity and gives damping due to absorption of kinetic energy. Such damping, when utilised, is achieved by increasing load torque as velocity is increased. Damping devices employing viscous friction are not often used because of the following disadvantages: increased response time to achieve steady state, increased losses, increased energy input with a larger velocity misalignment to produce this input. Viscous and static friction are of course always naturally present to an extent.

Stabilising feedback

Modifying feedforward or feedback is to minimise any tendency to oscillate. In principle the object is to decrease servo-motor output torque as speed increases by tachogenerator feedback, which is preferable to viscous friction damping as the effect is more linear and no extra energy input is required.

Refer to Figure 14.4 in which the relevant part of the loop is shown dotted; also shown is a gearing (G) and load system (L). Stabilising feedback from the tachogenerator (T_1), proportional to velocity, reduces the voltage input error signal, proportional to deviation, hence the amplifier input is reduced. No input will exist before the shafts are aligned due to the tachogenerator feedback voltage reduction. Inertia will move the output shaft and reverse amplifier output will provide a braking torque to bring the motor to rest. For a ramp function (linear displacement variation, constant velocity) velocity misalignment exists because with the input shaft stopped in position a zero input voltage to the amplifier is required before shaft alignment. This requires an equal ... available in the main



▲ **Figure 14.4** *Block diagram (three-term position)*

Rate network

From the above it is obvious that a voltage proportional to deviation plus a voltage proportional to rate of change of deviation (i.e. velocity) is required, which is $P + D$ action. The derivative (rate) circuit has been covered in detail for process control. It is indicated at D and is a phase advance whose voltage requires extra amplifier gain.

For a step input, peak overshoot, settling time and rise time tolerances may be specified as a given percentage of the step change. With a ramp function it is necessary to specify allowable velocity misalignment (function of rate of input change) as a given percentage of maximum input velocity (rate of ramp input).

Offset

This has been discussed in process control and is a characteristic of proportional control at different loads. Obviously in the position system being discussed it is velocity misalignment. This is a steady state position error due to viscous torque (proportional to velocity) additive to load so that offset must occur. It can be eliminated by one of two methods; the former is preferred.

Reset network

A voltage proportional to deviation plus a voltage increasing with time at a rate proportional to deviation is required, that is, $P + I$ action. This has been discussed previously and the reset network (I) is shown on Figure 14.4. The capacitor stores enough delayed charge to feed the amplifier to reduce deviation to zero against the extra loading of frictional effect. It can assist against initial inertia and friction

Reset feedforward

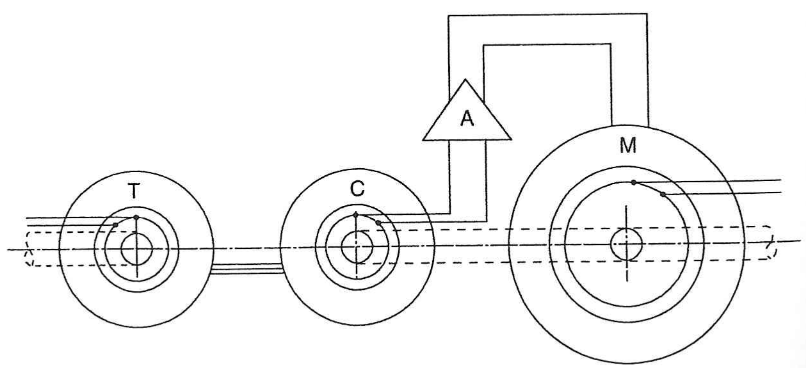
A second identical tachogenerator (T_2) is driven off the input shaft and supplies a feedforward additive voltage proportional to speed. In the steady state for a ramp function the feedforward from T_2 balances the feedback from T_1. Hence there is no velocity misalignment and the offset is removed. When the input shaft stops (T_2 zero output) the following output shaft rotation provides the usual stabilising feedback rate action from T_1 to reduce overshoot.

Position control servo-mechanism (ac)

Generally limited to small powers and although cheap have a low start torque and relatively low performance. Essentially two categories of ac system exist, that is, demodulator-modulator and all ac. The principles involved in the former have been discussed in Chapter 7, and with application to electronic controllers in Chapters 11 and 12.

The input shaft is connected to the single phase ac rotor of the transmission (T) and the output shaft to a similar rotor of the controller (C) whose excitation is amplified (A) for supply to the main drive servo-motor (M) to bring the shafts into alignment. Three-phase stators of transmitter and controller are directly connected. A null point of relative rotor positions exists, variation from which gives a proportional voltage in one phase or the other. This is *not* a synchro-transmission link because the output shaft rotor is *not* mains excited.

The two transducers are three-phase induction motors (as synchros-magslips) and usually the main output servo-motor is two phase (one fixed, one control).



Stabilising feedback can be used with an ac tachogenerator. The stator is wound with an input reference field which acts as excitation and an input field wound at right angles. Rotor cutting of reference field induces an emf in the output field proportional to speed and in phase and frequency with input signal.

Hydraulic position control servo-mechanism

Hydraulic position control has many applications. A typical example would be an electro-hydraulic steering gear. The control function acts to vary plunger travel, often by radially displaced or 'swash plate' piston-operated devices, and so give a variable delivery pump. Linear or angular operation to any position control system, particularly hydraulic, is easily arranged.

Speed Systems

The control of speed (velocity), linear or angular, in a system. Most of the principles of speed control are applicable to position control, which has been already been discussed, so only a brief analysis is given here.

Speed control servo-mechanisms (dc)

The essential equations relating to dc motors are:

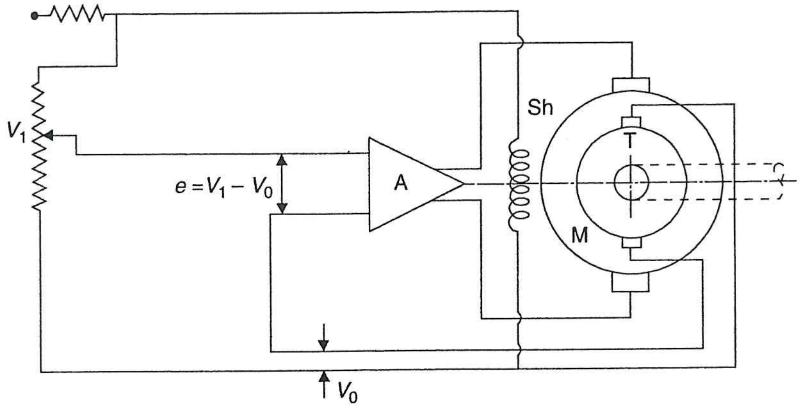
$$I \propto \Phi I_a$$

where T is output torque, Φ flux and I_a armature current.

$$N \propto \frac{V - I_a R_a}{\Phi}$$

where N is speed, V applied voltage and R_a armature resistance.

$$P \propto V I_a - I_a^2 R_a$$



▲ **Figure 14.6** *Speed control servo-mechanism (dc)*

Obviously speed control can be effected by varying armature voltage or field flux, the former being shown on Figure 14.6.

A speed set voltage from the input potentiometer has an input volts signal V_1. Monitored feedback from the tachogenerator gives a voltage (V_0) proportional to output shaft speed. The error signal $e = V_1 - V_0$ is amplified and fed to the main drive motor.

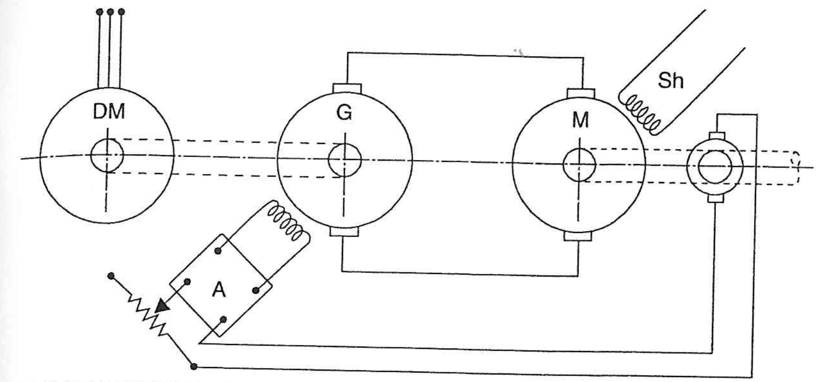
A derivative (rate) and integral (reset) circuit could be added. Rate of velocity change is acceleration. Gyros measure acceleration in many control systems.

Speed control (ac)

This has become common practice due to advances in power electronics and computer technology. Thyristor based dc motor speed controllers were developed into ac speed controllers based around the synchro-converter arrangement (covered under propulsion control) but these systems were insufficiently flexible to match many of the features of dc machines. Improvements in field effect transistor (FET) technology and the development of the insulated gate bipolar transistor (IGBT) enabled the development of pulse width modulation (PWM) or voltage source controllers which could match all of the commonly desired advantages of dc motor speed controllers.

Ward-Leonard speed control

Consider the arrangement shown in Figure 14.7: Terminology and principles of



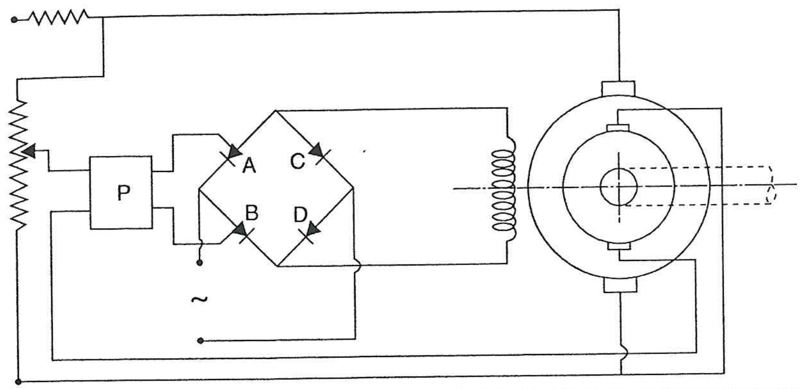
▲ **Figure 14.7** *Ward-Leonard speed control*

induction motor with standard dc generator (G) and servo-motor (M), the latter with a constant excitation field. Generated supply voltage to the servo-motor depends on the error signal between desired and measured speed values, as proportional signal voltages, which is amplified and fed to the generator control field. For small powers a valve or transistor amplifier could be used with extension to medium powers with a magnetic amplifier. High gain and stability from drift can be obtained utilising such ac amplifiers suitably modulated and demodulated. For high powers it is usually necessary to replace these amplifiers with either a rotating amplifier – auxiliary generator (exciter) or amplidyne (metadyne, cross-field dc generator) – or modern thyristor control amplifier.

Controlled rectifier units can also be used. The mercury arc rectifier and ignitron can be used for large powers but are generally limited to specialist applications. Thyratron devices, with transformer coupling to anode and grid, have been successfully used. Thyristors are being increasingly used either directly or as field control devices.

Thyristor Speed Control

Consider the arrangement shown in Figure 14.8. The sketch has the familiar layout used previously, field control is by thyristor. The bridge circuit consists of two rectifiers C and D and two controlled rectifiers (thyristors) at A and B. The gate of each thyristor is triggered by pulses from *P* representative of error speed input signal. Alternating current supply is rectified for output to field by passage through B and C on one cycle and D and A on the other cycle. This yields fullwave rectification with thyristor trigger



▲ **Figure 14.8** *Thyristor speed control*

Governor Systems

Many engines, turbine and reciprocating, are still fitted with trip devices to allow full energy supply under normal conditions. If revolutions rise about 5% above normal the energy supply is cut off until normal conditions are restored – lock out occurs at about 15% excess which can only be unlocked by hand. Aspinall types come into this category.

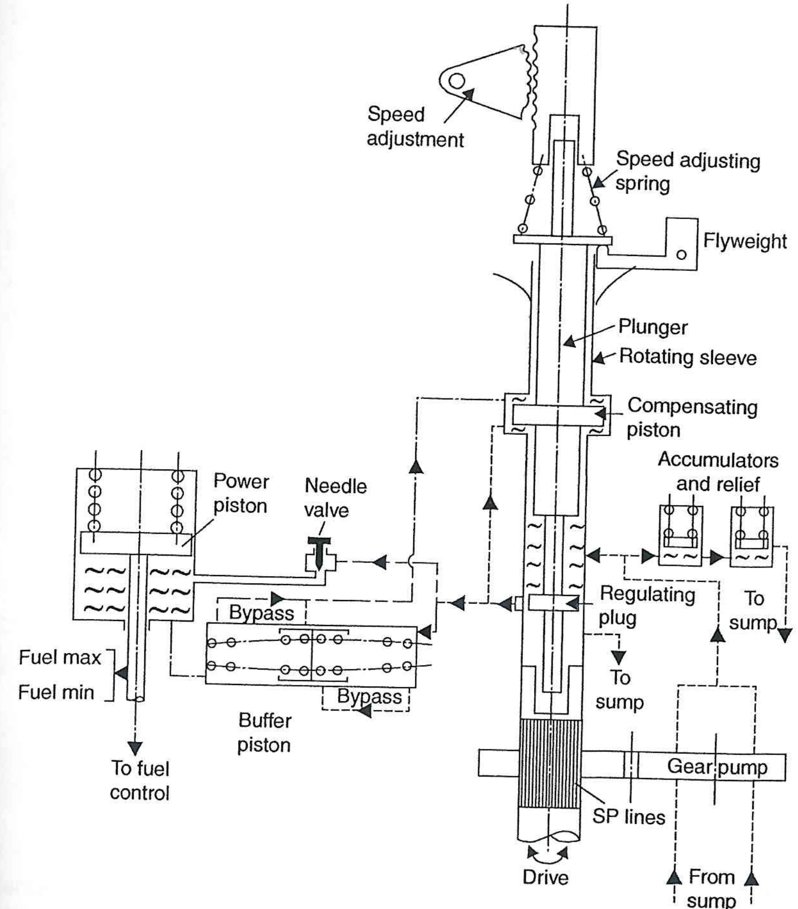
Smaller engines, such as electric generator drive, often use centrifugal governors based on the Watt principle, the Hartnell governor is typical. Control is essentially proportional action with sensed output (rotation speed) controlling energy input and offset (exemplified by no load to full load speed droop) occurs.

Modern engine governors are isochronous devices but reset action is applied to eliminate hunting. One such *P* + *I* governor has already been described in Chapter 12.

Two designs will now be considered: mechanical-hydraulic (similar in principle to Figure 12.2) and electrical-hydraulic (utilising principles discussed previously in this chapter).

Mechanical-hydraulic speed control servo-mechanism

Many engine units employ these servo devices incorporating inbuilt safety for oil or



▲ **Figure 14.9** *Mechanical-hydraulic speed control servo-mechanism*

Refer to Figure 14.9. With the engine running at constant speed under a steady load the up-force due to centrifugal force from the flyweights is balanced by the down-force of the speeder spring. The plunger is central with the regulating plug covering the regulating ports in the sleeve. The plunger moves vertically but does not rotate and the opposite applies to the sleeve. The power piston is stationary and the buffer piston is central under these conditions.

Consider a load *increase* on the engine for which condition Figure 14.9 is applicable. Speed reduces and the plunger moves down with pressure oil flow to the right of the buffer piston, which moves left. The power piston will move up and admit more fuel to the engine. Pressure oils also acts on the compensating piston under side which will exceed the pressure on this piston upper side so that the plunger will be restored up the needle valve, to restore equal pressures on each side of the buffer piston and compensating piston. The buffer piston is returned to mid-position by the springs. This gradually reduces the up-force on the compensating piston but the increasing engine speed is also increasing this up-force due to centrifugal force. The compensating piston is designed to be balanced gradually so that the rate of leakage at the needle valve (unloading) equals the rate of loading due to extra centrifugal force caused by higher engine revolutions. The engine will now run at normal speed but with increased load and higher fuel setting. The needle valve should be screwed in sufficiently to prevent hunting but without making the operation sluggish. Oil pressure is shown dotted with the slightly lower pressure chain dotted.

The bypass arrangement ensures that for a large speed change the power piston only moves as far as the bypass. Pressure oil flows directly to the power cylinder without further increasing the D/P on the compensating piston. After sufficient governor movement and speed return to near normal the D/P acts as usual.

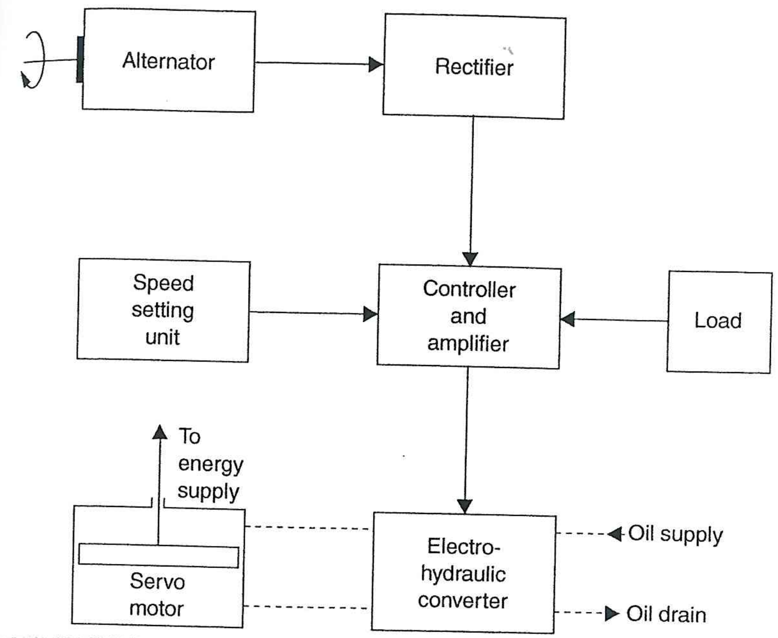
In the event of a large load decrease the power piston is at fuel minimum and blocks the needle valve connection. This gives a higher speed setting than normal and reduces a tendency to under speed.

Electrical-hydraulic speed control servo-mechanism

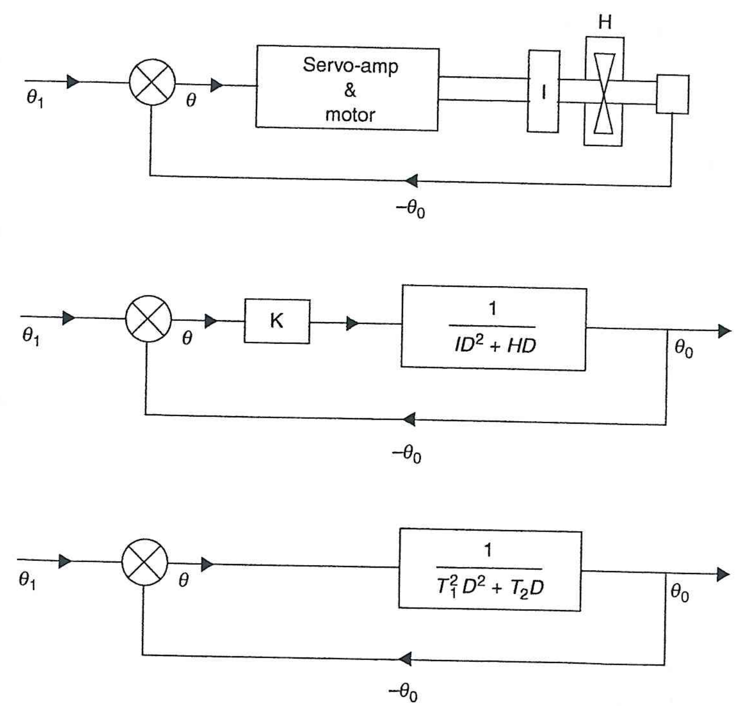
Rotational speed is sensed by tachogenerator, or as shown in Figure 14.10 by ac alternator, with frequency pulses converted in the rectifier to dc voltage proportional to speed. Set value is applied to the controller and the two input voltages, opposite in polarity, are compared. Error signal (if any) is amplified, converted to hydraulic signal and operates to alter energy supply through a servo-motor. To reduce hunting, and offset, the controller has reset (integral) action via a feedback loop. The unit is also anticipatory, that is, load changes are fed back to the controller to amend energy input *before* speed change occurs – speed control is virtually a fine trimming operation.

Mathematical Aspects

Figure 14.11 is introduced now to link illustrative block diagrams (as Figure 14.2) and the analyses of transfer functions for open and closed control loops (Chapter 15). It is a position control servo-mechanism with unity feedback. Torque proportional to



▲ **Figure 14.10** *Electrical-hydraulic speed control servo-mechanism*



Torque applied to load ($I\alpha$) = $K\theta - H\omega$

$$K\theta = I\frac{d^2\theta o^2}{dt^2} + H\frac{d\theta}{dt}$$
$$= \theta_0(I\dot{D}^2 + HD)$$

Open-loop transfer function

$$\frac{\theta_0}{\theta} = \frac{K}{ID^2 + HD} = \frac{1}{T_1^2D^2 + T_2D}$$

Closed-loop transfer function

$$\frac{\theta_0}{\theta_i} = \frac{(T_1^2D^2 + T_2D)^{-1}}{\{1 + (T_1^2D^2 + T_2D)^{-1}\}} = \frac{1}{T_1^2D^2 + T_2D + 1}$$

where $T_1 = \sqrt{\frac{I}{K}}$ and $T_2 = \frac{H}{K}$ are time constants. See now pages 265 and 266.

Test Examples

1. Explain the difference between 'open-loop' and 'closed-loop' systems of control.

   Draw a circuit diagram for a system in which the Ward–Leonard arrangement with feedback control is used to regulate the speed of a dc motor and explain the mode of operation.

2. **(a)** Sketch a clearly labelled circuit diagram for a simple electrical remote position control servo-mechanism with zero damping.

   (b) With the aid of waveform sketches describe the action of the system when subjected to a step input.

   (c) Compare any advantages and disadvantages of the following methods of damping such as a servo-mechanism:

   (i) viscous friction, **(ii)** output velocity feedback.

3. Describe a thyristor control arrangement for the speed control of a large electrical fan. Show how a zener diode can be used to stabilise the voltage supply to load and include the necessary protection to safeguard the diode against overload.

4. **(a)** Distinguish between the following control system terms:

   (i) Error

   (ii) Offset

   (iii) Monitored feedback

   (iv) Deadband

   (b) For a remote position control servo-mechanism: (1) State the effect of adding integral action. (2) Describe velocity feedback damping. (3) Sketch waveforms to illustrate response to step input.

15

CONTROL SYSTEM ANALYSIS

This subject is generally complex and the objective of this chapter is to introduce the basic principles so as to allow an initial appreciation which could be further developed, if required, at a later stage. Consideration is given to the systems approach, the order of linear systems, performance of systems, component interaction and adjustment.

The Systems Approach

System

A general definition is: a functional assembly with components linked in an organised way and affected by being within, and changed if removed from, the boundary.

A system may exist as a sub-system within a larger system so that a hierarchy exists, for example, the biological cell, within the heart of an animal, within a social human system of a universe.

State

A system may be discrete, that is, exist in only one clearly defined state at a given time, or

or probabilistic with random change or subject to external influence. A closed system always tends to seek equilibrium and ideally has no energy transfer with surroundings outside the boundary whereas an open system tends to approach a steady state of balance with the surrounding environment. The *black box* philosophy is applicable, internal form unknown and sealed, with the only factors of interest being the output and input variables and their relation.

Essentially the systems approach can be detailed as:

1. Specify aims and objectives of the problem or analysis.
2. Establish system and sub-system boundaries.
3. Devise functional conceptual models of the problem leading to block diagrams with attendant mathematical models (equations) allowing for interaction between component units, feedback analyses, etc.
4. Scale system variables and construct circuit diagrams or firmware models.
5. After evaluation and iteration the synthesis can be established and tested for final appraisal.

(The reader is strongly advised at this stage to consider Figure 16.1A of the next chapter for a systems approach applied to a very basic engineering mechanism, that is, the simple pendulum and its analogue.)

Development to transport, banking, education, manufacturing, community systems, and such is an essential part of general systems theory and philosophy based on the techniques outlined in principle above.

Systems considered in this context are linear, that is, equations with constant coefficients. In practice non-linearities exist but unless complicated theory is utilised are difficult to analyse. Many cases exist where the effect of non-linearity can be negligible by correct design so that linear theory can be applied.

A systems approach requires a unification between similar quantities and should result in a generalised mathematical model whose equations are applicable for simulation and evaluation.

Analogues

It should be remembered that rate of change of a variable with respect to time can be written d/dt in calculus notation...

change of translational displacement with respect to time) can be written *dx/dt*; an alternative is $\dot{x}$. Similarly, for example, rotational (angular) acceleration is *dw/dt*, or $\dot{\omega}$ and as this is the second-rate derivative of rotational displacement it also equals *d2θ/dt2*; or $\ddot{\theta}$. Numerous other examples can be quoted.

Variables in common use include force, torque, voltage difference (drop), pressure difference (drop), displacement, velocity, acceleration, current, flow rate, etc. *Parameters* include stiffness, damping coefficient, mass, inertia, resistance, capacitance, inductance, etc. (see Table 15.1).

Consider the following:

$$\text{Generalised impedance } (Z) = \frac{\text{Across variable } (X)}{\text{Through variable } (Y)}$$

Through variables are velocity, current, flow rate, etc.; across variables are force, voltage, pressure, etc.; impedance parameters are inertia, resistance, capacity, etc. Elements in a system can be classified as dissipative, where $X \propto Y$ (such as resistors); as *storage*, where $\dot{X} \propto Y$ (such as capacity); or as *storage*, where $X \propto \dot{Y}$ (such as inductance).

Table 15.1 compares the translational mechanical, electrical and fluid systems. Extension to pneumatic, thermal and rotational mechanical systems gives a similar result in every case. *B* is damping coefficient, *S'* spring stiffness, the dot above variables indicating rate of change, for example, $\dot{v}$ will be acceleration.

Gen.	X	Y	$Z=\frac{X}{Y}$	$Z=\frac{\dot{X}}{Y}$	$Z=\frac{X}{\dot{Y}}$
Mech.	Force	Velocity	Damper $B=\frac{F}{v}$	Spring $S'=\frac{\dot{F}}{v}$	Mass $m=\frac{F}{\dot{v}}$
Elec.	Voltage	Current	Resistor $R=\frac{V}{I}$	Capacitor $\frac{1}{C}=\frac{\dot{V}}{I}$	Inductor $L=\frac{V}{\dot{I}}$
Fluid	Pressure	Flow	Resistor $R'=\frac{p}{f}$	Capacitor $\frac{1}{C'}=\frac{\dot{p}}{f}$	Inductor $I'=\frac{P}{\dot{f}}$

System Order

The response of any system, or a component within the system, can be described by a mathematical equation. The order of the equation, and hence the system or component, is fixed by the highest derivative 'power'. In a mechanical system, for example, velocity is the first derivative of displacement, that is, $v = dx/dt$ so that a system with such a term as the highest derivative is classified as a *first-order* equation. All such first-order systems are defined by having *one* form of energy storage component. Similarly acceleration as the second derivative of displacement, that is, $a = d^2x/dt^2$ is a *second order*. All such second-order systems are classified by having *two* forms of energy storage components and *one* form of dissipative energy component. It is often convenient to write *D* for *d/dt* in calculus notation (note D^{-1} is a first *integration*).

First-order systems

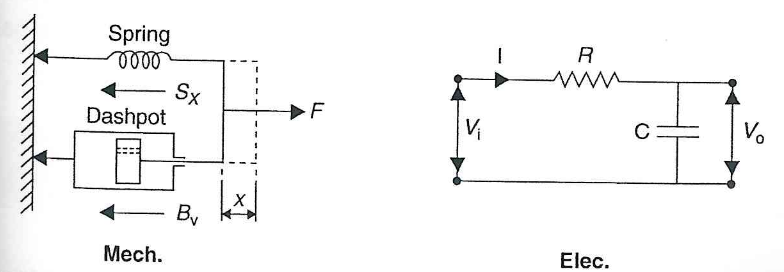
Consider the mechanical translation system of Figure 15.1.

The constant applied force (*F*) is resisted by the spring force (*S'x*) and the dashpot damping force proportional to velocity (*Bv*). Now $v = dx/dt = Dx$ so that the equation is written as:

$$F = S'x + BDx$$

and after re-arrangement this becomes

$$\frac{B}{S'}Dx + x = \frac{F}{S'}$$



which is a typical first-order equation, time constant $\tau = \frac{B}{S'}$

For the electrical resistance–capacity system shown:

$$\tau DV_o + V_o = V_i$$

(in charge terms, $\tau DQ + Q = V_i C$ when DQ is current) where $\tau = RC$ is a time constant.

For a heat resistance-capacity system:

$$\tau D\theta_E + \theta_E = \theta_F$$

where $\tau = RC$, θ_E and θ_F element and fluid temperatures.

For a fluid restrictor-capacity tank system:

$$\tau Dp_0 + p_0 = p_i$$

(in quantity terms, $\tau Dq + q = p_i C'$ where Dq is flow rate) ('head', $\tau Dh = h = \dot{q}_i R$ 'τ = Area × R).

Such equations can be extended to economic, management, etc. systems and a general equation arising from the above analogous cases may be written down as follows (y any variable):

$$\tau DY + y = bf(t)$$

where τ is the system time constant and b is a constant. It is possible that the input may not be a step or ramp form of constant, such as a dc voltage, but may be varying with time, such as an ac voltage. The $f(t)$ forcing function term is a general way of writing an input function dependent on time, for example sinusoidally, to allow for such variations. If the input is not so varying the right-hand side of the equation is a constant, as covered in the analogous cases given above, when y is made up of a constant and a variable, with a system time constant in the solution, that is, $y = b(1 - e^{-t/\tau})$. b is usually the gain constant.

Transfer function

... ratio of its output signal to its input signal.

$$\text{Transfer function} = \frac{\theta_o}{\theta_i}$$

Its use, together with block diagrams, simplifies analysis using s-plane in place of differential equations. In some cases output is merely amplified or attenuated input, for example, gearbox, while in other cases the signals may be in different physical form with different amplitude and phase.

Consider the RC network of Figure 15.1:

$$V_i = IR + V_o$$
$$I = C\frac{dV_o}{dt} = CDV_o$$
$$V_i = RCDV_o + V_o$$

(this is the first-order equation given previously)

$$V_i = V_o(RCD + 1)$$
$$\text{Transfer function} = \frac{V_o}{V_i} = \frac{1}{1 + \tau D}$$

This result will be characteristic of all such first-order equations.

Closed-loop transfer function

Figure 15.2 illustrates the usual arrangement. The transfer function of forward elements is G and of feedback element F.

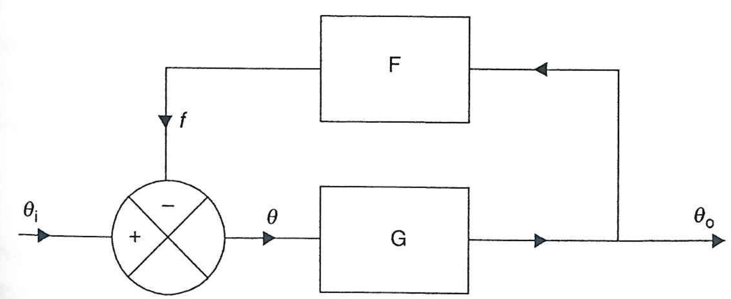


Figure 15.2 *Closed-loop transfer function*

1. Consider the loop of feedback to be opened and let $F = 1$ (direct feedback):

$$\text{Open-loop transfer function } (G) = \frac{\theta_o}{\theta}$$

for forward path elements.

2. Consider closure of the feedback loop and $F = 1$:

$$\text{Closed-loop transfer function} = \frac{\theta_o}{\theta_i} = \frac{\theta_o}{\theta + \theta_o}$$
$$= \frac{\theta_o/\theta}{1 + \theta_o/\theta} = \frac{G}{1+G}$$

3. Consider the system as sketched:

$$f = F\theta_o;\quad \theta = \theta_i - f;\quad \theta_o = G\theta_o$$

combining these equations gives:

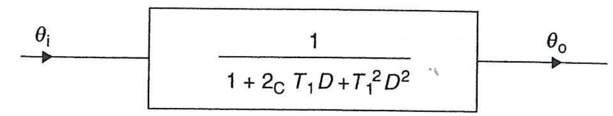
$$\theta_o/G = \theta_i - F\theta_o$$
$$\theta_i = \theta_o\left(\frac{1+FG}{G}\right)$$

$$\text{Closed-loop transfer function} = \frac{\theta_o}{\theta_i} = \frac{G}{1+FG}$$

$$\text{Open-loop transfer function} = \frac{f}{\theta} = FG$$

1. The open-loop transfer function is very useful in stability testing (see later).
2. Note that for direct feedback the closed-loop transfer function is (open-loop transfer function) divided by (one plus open-loop transfer function). Increasing gain G reduces offset.
3. This is a simple example. Practical cases with more involved transfer functions are more complicated and difficult to solve.

The response output θ_o will depend on the form of the input signal. For elements in series the overall transfer function is the product of the individual transfer functions, ... there is no interaction between the components, that is, $G = G_1G_2$. There will



▲ **Figure 15.3** *Transfer function block diagram*

need to algebraically add phase shifts. With elements in parallel $G = G_1 + G_2$. For a given open-loop transfer function there is only one closed-loop transfer function, hence open-loop analysis gives closed-loop analysis automatically. This fact is very important as an unstable closed system cannot be measured but by opening the system an analysis is possible allowing stability compensation to be made. The open-loop transfer function, independent of where the loop is opened, is *FG*.

If the system under consideration was itself a component of a control system it could be represented as one block diagram enclosing its transfer function. Such a second-order system is shown in Figure 15.3.

Second-order systems

Such systems have analogues as detailed previously. An in-depth review is not required but three examples are given:

1. *RCL series electrical network.*

$$\text{Applied volts} = \text{Resistor volts} + \text{Capacitor volts}$$
$$+ \text{ Inductor volts}$$
$$V = IR + \frac{1}{C}\int I\,dt + L\frac{dI}{dt}$$
$$\frac{dV}{dt} = R\frac{dI}{dt} + \frac{I}{C} + L\frac{d^2I}{dt^2} \quad \text{(after differentiation)}$$

2. *Mechanical translational damper, spring, mass assembly (see Figure 15.10).* Applied force = Damping force + Spring force + Inertia force.

$$F = Bv + S'x + ma \qquad (B \text{ damping coefficient})$$
$$F = B\frac{dx}{dt} + S'x + m\frac{d^2x}{dt^2} \quad (S' \text{ spring stiffness})$$
$$\frac{d^2x}{\ldots} + 2k\frac{dx}{\ldots} + \omega^2 x = \frac{F}{\ldots}$$

$2k = \frac{B}{m}$ where k is the damping factor

$\omega_n^2 = \frac{S'}{m}$ where ω_n is the natural (undamped) frequency

$$D^2x + 2c\omega_n Dx + \omega_n^2 x = \frac{F}{m}$$

$$c = \frac{k}{\omega_n} \text{ where } c \text{ is a damping ratio}$$

(constant changing is purely for mathematical convenience, m is the effective mass of the spring and its load).

3. *Position control servo-mechanism.*

Reference should be made to Figure 14.1 in Chapter 14, showing this unit. Output drive torque is dependent on inertia, viscous and stiffness (static and load error) torques.

$$K_1K_2K_3(\theta_i - \theta_o) = I_\alpha + H\omega$$

(I is the moment of inertia, H is damping coefficient)

$$\frac{1}{K_1K_2K_3}D^2\theta_o + \frac{H}{K_1K_2K_3}D\theta_o + \theta_o = \theta_i$$

(Inertia torque is for motor and load, direct drive assumed, viscous torque proportional to angular velocity: K_1, K_2, K_3 are respectively potentiometer bridge, motor torque-current and amplifier *scaling constants*.) Simplifying constants gives:

$$(D^2 + 2cw_nD + w_n^2)\theta_o = w_n^2\theta_i$$

$$\text{Transfer function} = \frac{1}{T_1^2D^2 + 2cT_1D + 1}$$

where $T_1 - 1/\omega_n$ con and is periodic time of undamped natural oscillation divided by 2π. For sinusoidal input the transfer function is as above but with $i\omega$ replacing D, that is, if $\theta = Ae^{i}\omega^{t}$ then $D\theta = i\omega\theta$; both D and i are operators.

Note: A general second-order equation allowing for input variation with time is:

$$(D^2 + 2c\omega_n D + \omega_n^2)y = a\omega_n^2 f(t)$$

where a is a constant, y the variable and $f(t)$ the forcing function.

Higher order systems

Commonly arise but are not considered here except to note three-term control action ($P + I + D$), action factors K_1, K_2, K_3.

$$V = -K_1\left(\theta + \frac{K_2}{K_1}\int \theta dt + \frac{K_3}{K_1}\frac{d\theta}{dt}\right)$$
$$= -K_1\left(1 + \frac{1}{SD} + TD\right)\theta$$

S integral action time, T derivative action time, V controller output. Application to say a position control servo-mechanism gives a transfer function of a third-order character obtained by equating the system second-order equation in terms of θ_o to the above equation in which $\theta = \theta_i - \theta_o$.

System performance

Ideal response is where output is identical to input command. This cannot be obtained because viscous friction and measure delays result in output lag. Effects such as inertia produce oscillation about the steady state. The mathematical approach gives an estimate of likely performance which can be improved by experimentation rigs, usually electrical analogues. The objective is attainment of desired value fairly quickly and accurately, with stability. Response to simple input signals has already been covered in the text but a quick resume can now be presented.

Performance results are in two distinct parts:

1. Transient response when the system is responding and may hunt.
2. Steady state response when the transient has died away.

the object in presenting the work is purely to introduce techniques and a general appreciation is all that is required.

Step input response

For a first-order system

$\tau Dy + y = bf(t)$ is the general equation

$\tau D\theta_o + \theta_o = \theta_i$ in control terminology

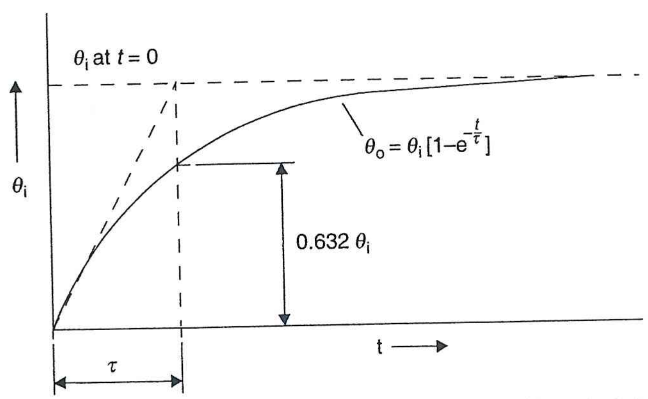
The steady state is a constant step input θ_i. The transient solution is $\theta_o = -\theta_i e^{-t/\tau}$. The complete solution is $\theta_o = (1 - e^{-t/\tau})$. To 'slow down' such a system and increase resistance, damping or capacity but decrease stiffness.

Reference to Figure 15.4 shows an exponential curve. The time constant T is the time to reach steady state if the initial slope was maintained. Output actually only reaches 63.2% of this value. There is no overshoot. The curve is characteristic of the *LR* electrical 'growth' response. The *CR* electrical circuit gives a decay characteristic.

For a second-order system:

$$(D^2 + 2c\omega_n D + \omega_n^2)y = a\omega_n^2 f(t) \quad \text{general}$$
$$(D^2 + 2c\omega_n D + \omega_n^2)\theta_o = \omega_n^2 \theta_i \quad \text{control}$$

Again the steady state is a constant step input θ_i (after dividing through by ω_n^2). The transient solution is obtained by setting the left-hand side to zero and three resulting



equations are possible depending on roots obtained. Reference should be made to Figure 14.3 of Chapter 14 where curves representing the three-solution equations are shown. Curve 2 ($c < 1$) is one solution for under-damped and is oscillatory. Curve 3 ($c = 1$) is another solution for critical damping. Curve 4 ($c > 1$) is the third possible solution for over-damped (aperiodic). The oscillatory case illustrates overshoot, a settling time, etc. For the electrical (series) system critical damping occurs when $R = 2\sqrt{L/C}$, for mechanical (translational) when $B = 2\sqrt{S'm}$.

Ramp input response

For a first-order system: The complete solution is $\theta_o = \omega[t - \tau(1 - e^{-t/\tau})]$. Reference to Figure 14.3 of Chapter 14 would indicate exponential approach to an inclined line representing steady state θ_o, similar to curve 3 (or 4), starting at τ on the horizontal axis (see Figure 9.2, Chapter 9).

For a second-order system: Reference to Figure 14.3 of Chapter 14 illustrates response for under-damped ($c < 1$ curve 2), critical damping ($c = 1$ curve 3) and over-damped ($c > 1$ curve 4). The inclined line of steady state θ_o, as appropriate to the degree of damping, will start at $2cT_1$ on the horizontal axis (see also Figure 9.2 of Chapter 9).

Sinusoidal input response

This is the case with the forcing function related to time, that is, $f(t)$ such as $\theta_i \sin \omega t$ or $\theta_i \cos \omega t$, for say an alternating voltage input; first- or second-order equation. The system is subjected to varied frequency sine wave inputs and response output is noted in magnitude and phase. Such analysis is also useful for evaluation of higher-order systems.

The first-order solution to $TD\theta_o + \theta_o = \theta$ (cos ωt) results in one equation (magnitude) and another equation (phase).

$$\frac{\theta_o}{\theta_i} = \frac{1}{\sqrt{1+\omega^2 T_1^2}}$$
$$\tan\phi = \omega T_1$$

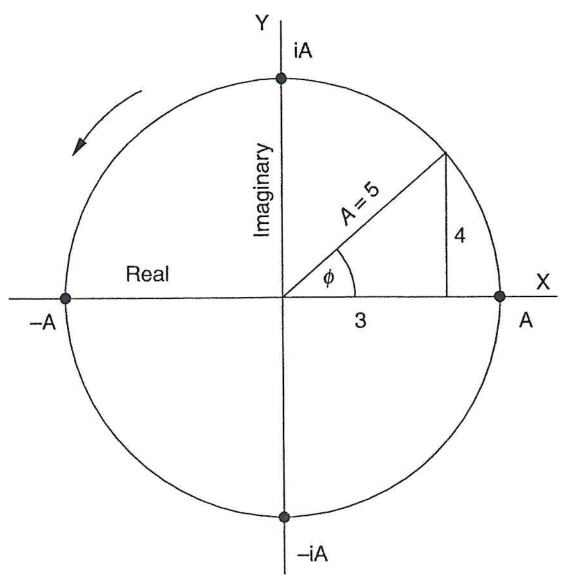
Consider as example the *RC* network covered previously (Figure 15.1) with the object of obtaining the steady state sinusoidal response. [illegible] inputs, is to replace operator D by $i\omega$ using the *complex number* notation which is very useful in ac networks or *polar* control plots. Symbol i (sometimes j) is an *operator* which rotates a vector by 90° in an anti-clockwise direction, without altering its length, its numerical value is $\sqrt{-1}$; i^2 is 180°, *numerically* −1; i^3 is 270°, numerically $-\sqrt{-1}$, that is, $-i$; i^4 is 360°, or 0°, that is, numerically 1. This is illustrated by an Argand Diagram (Figure 15.5) which shows a vector A whose *modulus*, or *amplitude* (length), is 5, that is, $\sqrt{3^2+4^2}$ and *arguement* (phase angle) is $\tan^{-1} 4/3$. The vertical (Y) axis is referred to as *imaginary* and the horizontal (X) axis as *real* and the number composed of real and imaginary parts is called a complex number. The diagram is often called the 's-plane'.

From the circuit of Figure 15.1:

$$I = \frac{V_i}{Z} = \frac{V_i}{R - i/\omega C}$$

(Capacitive reactance (Xc) is $i/\omega C$ *where* $\omega = 2\pi f$, impedance Z.) ($-i$ indicates 270°, that is, a current leading voltage, capacitive effect.)

$$V_o = \frac{(-i/\omega C)V_i}{R - (-i/\omega C)}$$



▲ **Figure 15.5** *Argand diagram*

The next step is to evaluate the transfer function:

$$\frac{V_o}{V_i} = \frac{-i/\omega C}{R - i/\omega C} = \frac{1/i\omega C}{R + 1/i\omega C}$$
$$= \frac{1}{1 + i\omega CR} = \frac{1}{1 + i\omega\tau}$$

This is the typical first-order transfer function with D replaced by $i\omega$. This expression gives the modulus, which is termed magnitude ratio (M) in control terms, and argument (phase shift).

$$M = \frac{1}{\sqrt{1^2 + \omega^2\tau^2}}$$

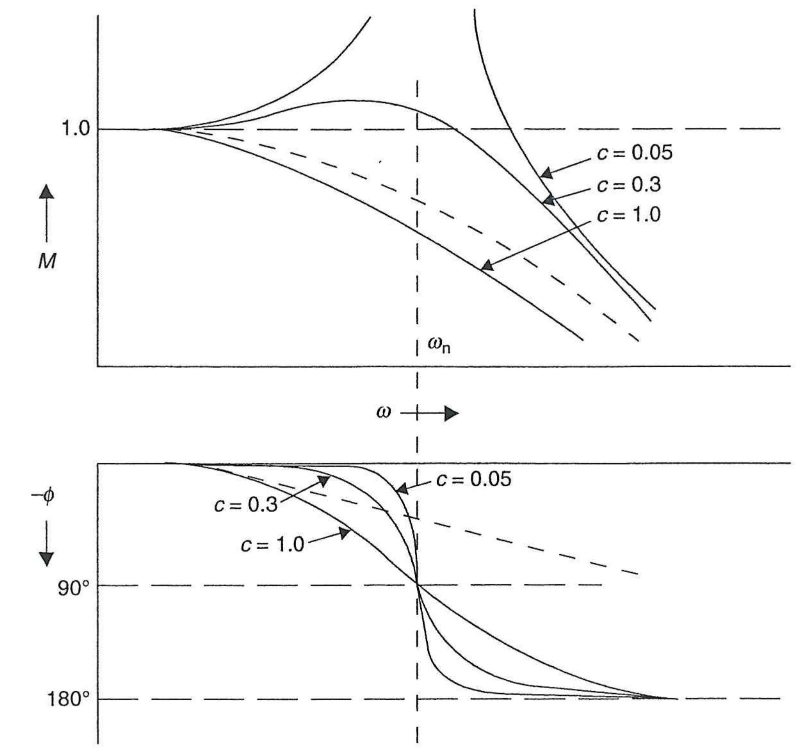
$$\varphi = \tan^{-1}\omega\tau \text{ phase of } V_o \text{ relative to } V_i.$$

Transfer function second order is $1/(1 + i\omega\tau_1)(1 + i\omega\tau_2)$.

The second-order solution is more complicated but shows the same characteristic result of a response of different magnitude and phase to input. Solutions are best illustrated graphically, Figure 15.6 (also Figures 9.3, 15.8). Measurement is by a cathode ray oscilloscope or transfer function analyser. Resonant frequency is $1/2\pi\sqrt{LC}$ electrical (series) and $1/2\pi\sqrt{S'/m}$ mechanical (translational).

Frequency response analysis

Refer to Figure 15.6. The dotted line shows first-order response and the full lines are second-order response curves for various degrees of damping. In the former case magnitude ratio (M the ratio between output and input amplitude) decreases steadily with input frequency and phase lag (φ) increases, due to the output being unable to follow input. In the latter case results are similar with a high resonance peak at ω_n for low damping which reduces with increased damping, as does ω_n^2. Lag is inherent in control systems. A frequency response diagram of $M \sim \varphi$ with M a logarithmic axis or decibels (dB = $20 \log_{10} \theta_o/\theta_i$), both effectively the same, is used in open- and closed-loop analysis. Response diagrams as in Figure 15.6, but with a logarithmic ω (or ωT_1) axis and a logarithmic M axis (or dB), are similarly used. Such diagrams if used in open-loop frequency response analysis are called Nichols and Bode diagrams respectively. On a Bode diagram, for stability, then (1) when dB are zero φ should be under −180°



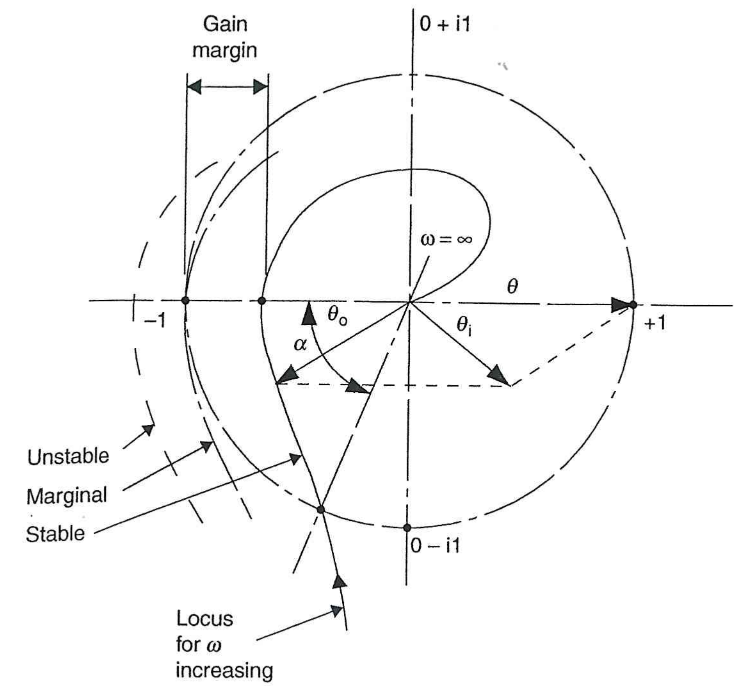
▲ **Figure 15.6** *Frequency response curves*

(e.g. −135°) for a positive phase margin and (2) when φ is −180° dB should be negative for a positive gain margin.

Stability response

The main aim of a frequency response test is to assess stability. One common method is to open the feedback loop and inject a small sinusoidal constant magnitude input signal (θ) to the forward path elements only and obtain a polar plot of this open-loop frequency response. The input is usually made unity and the polar plot is a Nyquist diagram obtained by measuring magnitude ratio and phase angle of output for increasing values of frequency from zero to infinity. M and φ could of course be calculated but this is obviously pointless, at least at this stage. A typical Nyquist diagram (open-loop polar plot) is shown in Figure 15.7.

The *Nyquist stability criteria* is that the *closed-loop system* is stable if the *open-loop* frequency response locus traced from $\omega = 0$ to $\omega = \infty$ does not enclose (pass to the



▲ **Figure 15.7** *Nyquist diagram*

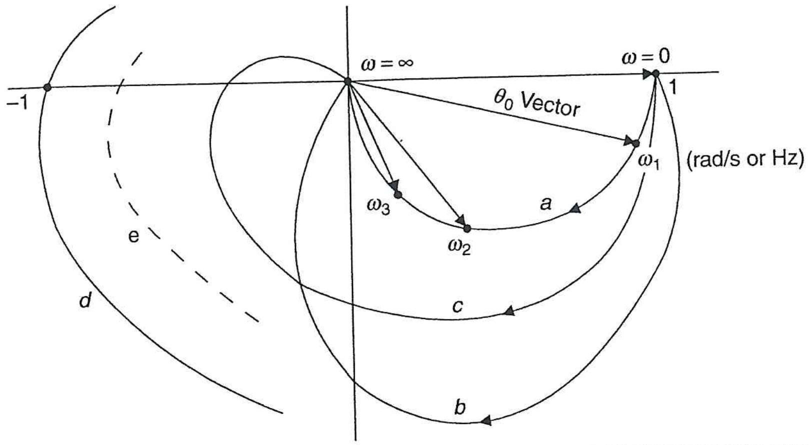
the diagram. The marginal state through (−1, 0) is shown, with instability beyond that. The circle is for unity gain ($M = 1$). Locus curves starting at various gain values on an extended axis to the right (where ω = o) can be plotted for fixed gain with increasing values of ω to determine correct gain for complete stability.

(Also shown is a vector sum $\theta_i = \theta_o + \theta$ being the input signal to give $\theta = 1$; this gives indication of *closed-loop* response but is not part of the normal Nyquist criteria.)

If $\theta_o = 1$ and φ = 180° for a particular frequency ω, *i* tip of θ_o locus vector is point (−1, 0) then θ_i to a closed-loop system is zero. This is unity feedback (positive) and slight change of θ_i will cause oscillation which would grow with increased amplitude oscillations for open-loop gain over unity. This is the basis of the Nyquist criteria.

Phase margin (α), as shown, should exceed 30° (usual range 30°–60°) and *gain margin* (sometimes expressed in dB) should exceed 0.3 (usual range 0.3–0.6) to ensure stability. The curve of Figure 15.7 is a typical fourth order.

Illustrative response curves are shown in Figure 15.8 ($M = 1$); *a* is typical of a first-order system, *b* second order, *c* is combination of *a* and *b* (third order effectively). Curve

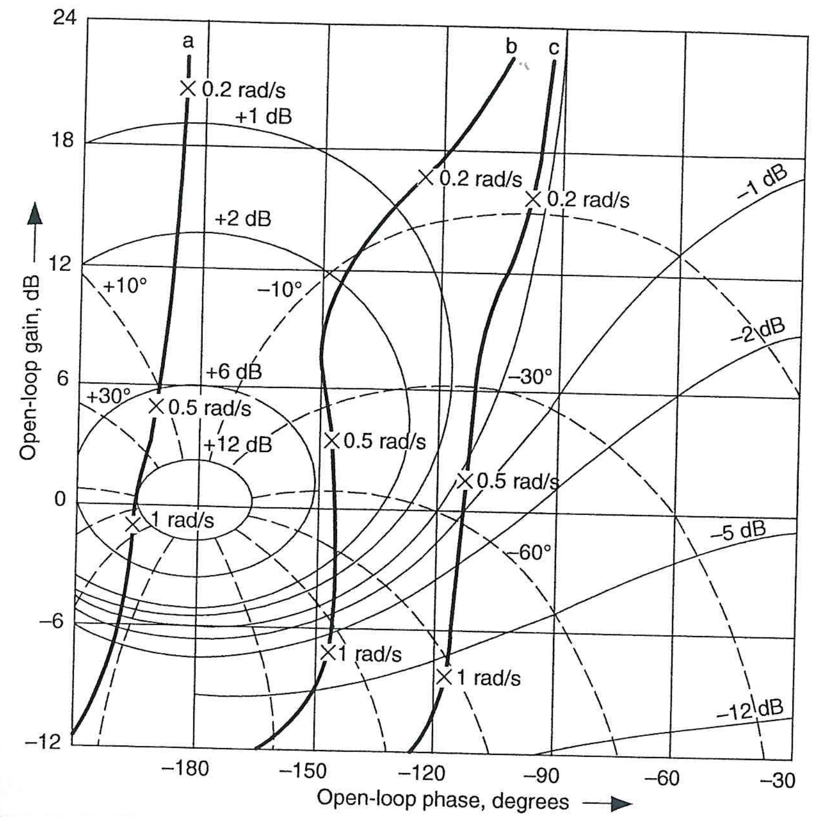


▲ **Figure 15.8** *Response curves (Nyquist)*

is plotted in the first quadrant with output leading input and amplitude increasing as ω increases.) The first- and second-order systems sketched cannot be made unstable with an increase in gain factor but the third-order system can (as can the fourth order of Figure 15.7). In such a case feedback from a tachogenerator would induce oscillation and increasing amplitude, that is, instability. This can be prevented by reducing the gain, which however reduced accuracy. Other methods of stabilisation include adding a passive phase lag network ($P + I$) or phase lead network ($P + D$) or a stabilising feedback *CR* circuit with the object of increasing stability without reducing gain. Curve *d* moved to curve *e* of Figure 15.8 illustrates the objective. It should be noted that good stability and high accuracy are incompatible and a compromise between the two is desired for best performance.

Contours of open-loop response (Figures 15.7 and 15.8) can be used to evaluate closed-loop response. For any point on the open loop, vector values of output to input ratio and the angle between them give points for the closed-loop response. These can be plotted on another similar harmonic response diagram or on a frequency response diagram.

The Nichol's chart (Figure 15.9) is frequently used in frequency response and stability analysis. Point plots of constant gain and phase (derived by calculation due to the logarithmic scale) give contours on which open-loop response can be plotted; minimum contours are shown on the sketch to simplify the illustration. Note that $20 \log_{10} 1 = 0$ dB, $20 \log_{10} 10 = 20$ dB and $20 \log_{10} 0.1 = -20 \log_{10} 10 = -20$ dB for the dB plotting so that negative dB corresponds to amplitude ratio (gain) of less than 1 (attenuation). Magnitude ratio 0.7–0.4 (−3 to −8 dB) corresponds to gain margin 0.3–0.6 (3–8 dB)



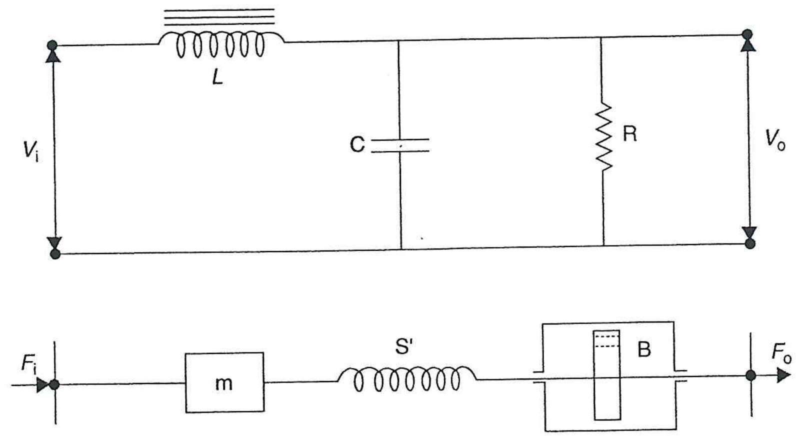
▲ **Figure 15.9** *Nichols chart*

On Figure 15.9 curve a (unstable) shows 15° phase lead at 11 dB, curve b (stable) 32° phase lag at −3 dB and curve c (improved stability 63° phase lag) at −8 dB, the operating frequency to be near 1 rad/s.

The closed-loop characteristic (direct feedback systems) can be derived on this chart from the intercepts of the open-loop locus with the gain and phase contours. This requires a simple calculation for values (which includes a feedback fraction).

Further analogues

In some cases approaches are used in which electrical components in parallel are regarded as equivalent to mechanical components in series, and vice versa. Certainly current, which has a common value through series electrical components, is analogous to velocity, which has a common value through linked mechanical translational com



▲ **Figure 15.10** *Second-order systems*

The lower diagram of Figure 15.10 illustrates a mechanical translational damper, spring, mass system which has been described previously (the left-hand end is often fixed, $F_i = 0$, $F_o = F$). It is typical of an anti-vibration mounting in which high frequency input oscillations will be damped out (see analogue circuit diagram, Figure 16.2).

The upper diagram of Figure 15.10 is the equivalent electrical circuit. High impedance to high frequency inputs results in a filter system in which only low frequency components are applied to R. The ratio of V_o to V_i can be found at a given frequency.

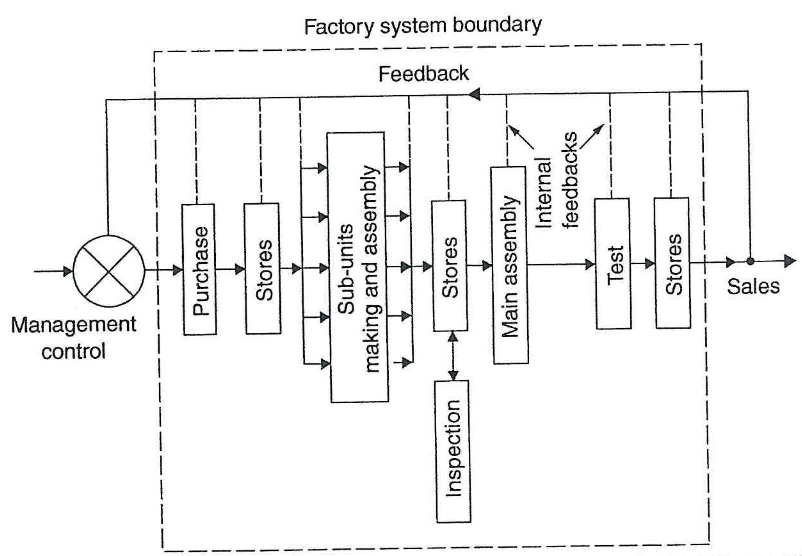


Figure 15.11 illustrates the systems approach to a factory management system which should be self explanatory. Management includes finance, development, etc.; sub-systems and techniques include critical path analysis, O.R., quality control, O&M, queuing theory, etc.

Component interaction

By this heading is meant the internal interference within a controller of the interference effect between connected controllers.

Internal interaction

The three-term equation for a controller has been considered and the equation in transfer function terms is:

$$V = -K_1\left(1 + \frac{1}{SD} + TD\right)\theta$$

This assumes each control action can be generated separately which may not be possible, and resulting interaction can occur which will affect output signal. Let consideration be applied to a three-term pneumatic controller – see Figure 10.7 of Chapter 10. As stated the position of integral and derivative adjustment can affect output in various ways. Consider the derivative adjustment placed at *X* in Figure 10.8. The control output signal can be shown to be:

$$V = -K_1\left[\left(1 + \frac{2T}{S}\right) + \frac{1}{SD} + TD\right]\theta$$

Integral or derivative adjustment affects controller gain (K_1). In this case K_1 is altered by the interaction factor (1 + *2T/S*). Analysis of each controller design is required to establish the exact output signal. Similar remarks apply for electronic controllers. Bode diagrams can be utilised to obtain characteristic plots of controller response which exhibit gain and phase variation.

External interaction

Consider two first-order controllers in series with *no* interaction between stages. The overall transfer function becomes:

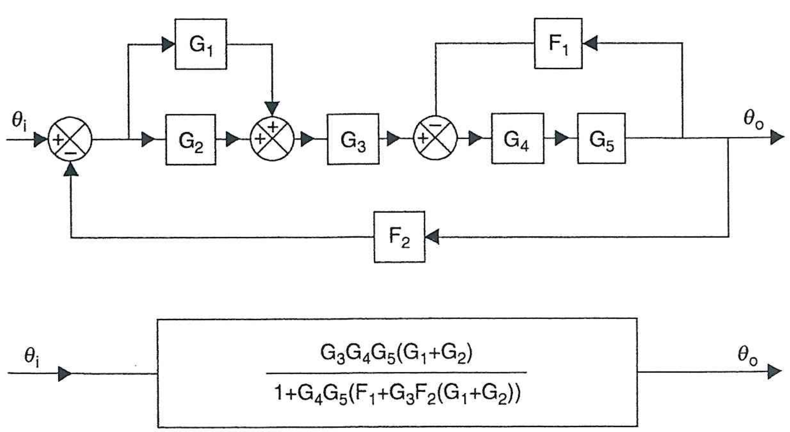
$$\frac{\theta_o}{\theta_i} = \frac{1}{(1+\tau_1 D)(1+\tau_2 D)}$$

which is a non-oscillatory type of second-order function. For a sinusoidal input the gain (attenuation) of individual elements are multiplied and phase angles added algebraically, utilising as usual $i\omega$ for D.

The control units must be non-interacting otherwise the transfer function of one controller will be modified by the loading of a following controller. This is usually avoided by inserting buffer amplifiers (or stages) of unit gain, without phase shift, between the controllers in series.

Block diagram reduction

A single (overall) transfer function can be obtained from a complete system consisting of individual (non-interacting) units each with its own transfer function. Block diagram algebra is used with two basic rules applied to each block pair in successive reduction: (1) parallel blocks are summed (2) series blocks are multiplied. A system and solution are detailed in Figure 15.12.



Component Adjustment

The adjustment of controllers, especially $P + I + D$, in a plant is usually done empirically by generally well-established experience criteria.

Adjusting controllers to plant

For initial commissioning the controller must be set up exactly to the manufacturer's instructions and all maintenance must follow similarly from maker's advice. Integral resistance is usually set at maximum and derivative resistance at minimum. Proportional band can now be set for minimum stabilisation time. Derivative resistance can now be increased to reduce this time a little, integral resistance now being adjusted to the same as derivative resistance. There is a definite relation between T and S settings; even with independent settings T can never exceed S.

We now consider setting and adjustment in more detail. This is a skilled operation requiring time and knowledge of plant characteristics so that the following, for a $P + I + D$ controller, is obviously a condensed simplification.

The object is to critically damp the signal to rest in the minimum time without overshoot and oscillation. Instability may occur for too narrow a proportional band, too short integral action time or too long derivative action time. Stability with underdamping gives oscillation with too long a stabilisation time; overdamping gives no oscillation but too long stabilisation time usually due to a too wide proportional band, too large integral action time or too short derivative action time.

For proportional action band only, it is best to start at say 200% bandwidth and move the dial away from and then back to the set value, noting the settling time. This is repeated at step reductions of bandwidth until the oscillations do not reduce to zero (too much reduction would cause instability with increasing oscillations). A slight increase in bandwidth now gives the correct value for minimum offset and stabilisation time.

For $P + I$ the proportional band would be set as for P action above with integral action time at maximum. The integral action time is then reduced (using big steps initially) until hunting oscillation starts. A slight increase in integral action time now gives the correct value and stability.

For $P + D$ the proportional band is narrowed as above until hunting is occurring and it is held at that value. The derivative action time, which had been set at minimum, is increased to remove hunting. The proportional band is again narrowed slightly and hunting removed by adding to the derivative action time. This process is continued until hunting cannot be removed by the derivative action time. The proportional band is now widened slightly for correct setting.

$P + I + D$ controllers may be adjusted in practice as for a $P + D$ controller, noting the derivative action time, adding the same integral action time then adjusting for minimum offset. Interaction is always rather a problem except in a well-designed controller, well matched to plant characteristics.

Empirical setting method ($P + I + D$)

I and D terms are reduced to zero and proportional band is narrowed until continuous cycling occurs. This may require a small step input on the desired value setting to start the oscillation. Although, continuous oscillation at constant amplitude is taking place the periodic time is measured. With this proportional band (W) and periodic time (T_0) it is possible to empirically set the controller thus:

$$\%\ \text{Bandwidth for } P = 2W$$

$$\%\ \text{Bandwidth for}(P + I) = 2.2W$$

$$S\,\text{for}(P + I) = \frac{T_0}{1{\cdot}2}\text{minutes}$$

$$\%\ \text{Bandwidth for}(P + I + D) = 1{\cdot}67W$$

$$S\ \text{for}(P + I + D) = \frac{T_0}{2}\text{minutes}$$

$$T\ \text{for}(P + I + D) = \frac{T_0}{8}\text{minutes}$$

Signals are then trimmed for optimum performance. The above is satisfactory for a continuous process but not for auto-start with no overshoot.

Test Examples

1. Sketch the harmonic response (Nyquist) diagram for frequency response tests on:
   (a) a stable system,
   (b) a critically stable system,
   (c) an unstable system.
2. A thermocouple at 10°C is placed in a fluid at a temperature of 60°C and the reading after 4 s is 40°C. Assuming exponential delay response evaluate the time constant of this instrument. If the thermocouple were then used to measure a temperature rising steadily at 2°C/s what would be the steady-state error of the reading? (4.37 s, 8.74°C)
3. A step change of 2.5% is applied to the input of a $P + I$ controller and the output gives a sudden step change of 5% and after 2 min the total output change is 12.5%. Determine proportional bandwidth and integral action time. A ramp change of 1% linearly is applied to the input of a $P + D$ controller and the output gives a sudden step change of 5% and after this the output changes linearly at 3% per minute. Determine proportional bandwidth and derivative action time (50%, 80 s, 33%, 100 s).
4. Detail a 'trial and error' (or 'hunt') method of setting a three-term controller. Show on a diagram the effect of (1). too long (2) too short and (3) correct setting of integral action time for a $P + I$ controller.

16 LOGIC AND COMPUTING

This chapter will be concerned with analogue computers, switching logic circuits, digital computers, data processing and computer control. Each section is obviously a specialist area of study and only an introduction is attempted in this chapter.

Analogue Computers

Many of the principles involved have already been covered in the immediately preceding chapters. The information is now summarised here.

Analogue

That which has correspondence or resemblance (analogous) to something else which may be otherwise entirely different in form. Analogies between resistance-damping, inductance-mass, capacitor-spring, etc. have been considered. This is extremely useful for simulation. Similar relationships exist for pneumatic, thermal and fluid systems.

Analogue computer

These are essentially devices to represent continuous measures of physical quantities

fast, inherently graphical and reasonably accurate simulation and investigation of mathematical models of dynamic systems. In most cases electrical analogues are used with voltage signals representing system variables.

Basic elements

These have been considered previously in Chapters 10 and 11 utilising both pneumatic and electrical transducer analogues of variables. Elements include the operational amplifier, feedback, averaging, ratio, summer, scalar multiplier, inverter, proportional-derivative-integral devices, flow, power, root extraction components, etc. incorporated as required to form a simple analogue computer.

Control engineering

The application of basic elements to form control devices in process and kinetic systems is a direct application of small analogue computation. These aspects have been described in previous chapters.

System analysis

The analogue computer solves mathematical equations for system analysis making it a valuable engineering tool. Variables can be varied readily and simulated results quickly obtained. Outputs may be conveniently recorded on pen chart devices, two axes plotters ($X \sim Y$), cathode ray oscilloscopes and ultraviolet recorders. Obviously the mathematical analysis is usually complex but the selection of a simple analysis of a basic equation should effectively illustrate the principles which can be applied to complex systems.

Simple pendulum analogue simulation

The procedure is an example of the systems approach to engineering problems. The procedure can be analysed from objectives as follows:

a. Sketch the *conceptual* (physical) *model* (see Figure 16.1).
b. Evaluate the *mathematic model* equations, in this case:

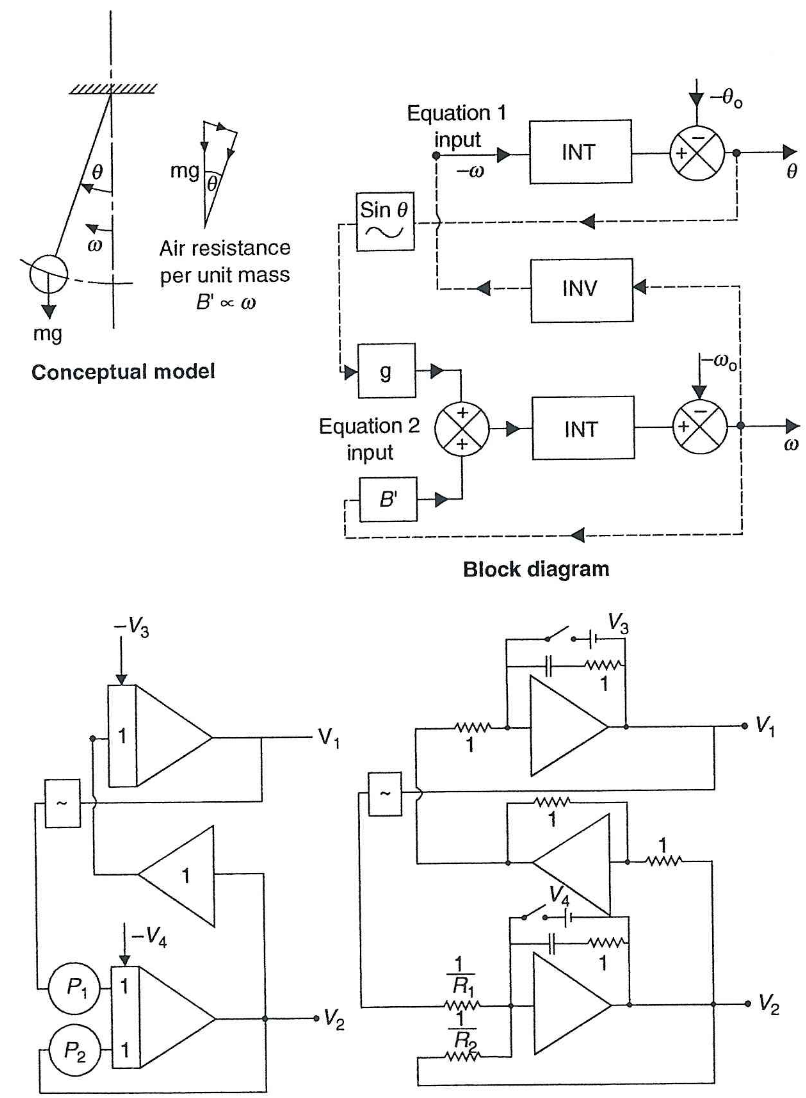
$$m\frac{d\omega}{dt} = -mg\sin\theta - mB'\omega$$

1. $$\frac{d\theta}{dt} = \omega$$

hence $\theta = \int_0^t \omega\, dt + \theta_0$, that is, $\theta = \theta_0$ at $t = 0$

$$2.\ \frac{d\omega}{dt} = -g\ \sin\theta - B'a$$

hence $\omega = -g\int_0^t \sin\theta\, dt - B'\int_0^t \omega\, dt + \omega_0$



Conceptual model

Block diagram

Analogue programme

Circuit diagram

c. Draw a *block diagram*, starting with integrators, adding scalars, etc. as required.
d. Scale variables, draw *analogue programme*, draw *circuit diagram* (see Figure 16.1).
e. Analyse by test for interaction, evaluation, synthesis; refine-iteration.

On the block diagram the multipliers, summers, integrators, inverter will be noted. It should be remembered that there is a sign change on the operational amplifier. Interconnection of components is termed *patching*.

On the scaled analogue programme (patching) diagram, sometimes called flow diagram, the unity ratio of amplifier resistance is indicated by a 1. Voltages V_1, V_2, V_3 and V_4 are analogues proportional to θ, ω, θ_0 and ω_0 respectively. Potentiometers are shown as *P*. Adjustment of potentiometer tap ratio *k* and amplifier gain *K* gives required ratio scaling for gain (*F* and *G* in feedback terms).

On the circuit diagram all factors should be clear from preceding diagrams. Resistances R_1, and R_2 are proportional to g and B'.

$$Dx = -ax + f(t)$$
$$x = -\int ax - f(t)dt$$

This is readily patched with two potentiometers (one input, one feedback) and an inverter integrator with suitable scaling for constants in a similar way to that shown in this pendulum example.

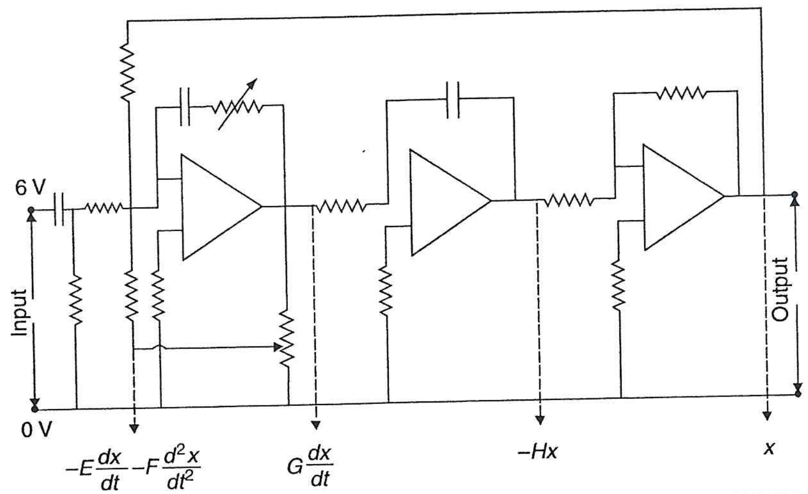
For a typical second-order diagram:

$$(D^2 + B'D + g)\theta = 0 \qquad \text{Pendulum}$$
$$(D^2 + 2c\omega_n D + \omega_n^2)x = a\omega_n^2 f(t) \qquad \text{general}$$

The approach is similar utilising, for example, three operational amplifiers in series (Figure 16.2). This example is an analogue for an anti-vibration mounting involving linear damper, spring, mass. The model diagram has been considered previously (see Figure 15.10) and the equation, linear simple harmonic second order, was derived in Chapter 15 and is repeated in a different form (pendulum and general case).

Net force (mass times acceleration) acting in the opposite direction to displacement and velocity is given by:

$$m\frac{d^2x}{dt^2} = -S'x - Bv$$



▲ **Figure 16.2** *SHM analogue circuit diagram (linear vibration damper)*

where S' is spring stiffness and B is damping coefficient,

$$\frac{d^2x}{dt^2} = -Cx - D\frac{dx}{dt}$$

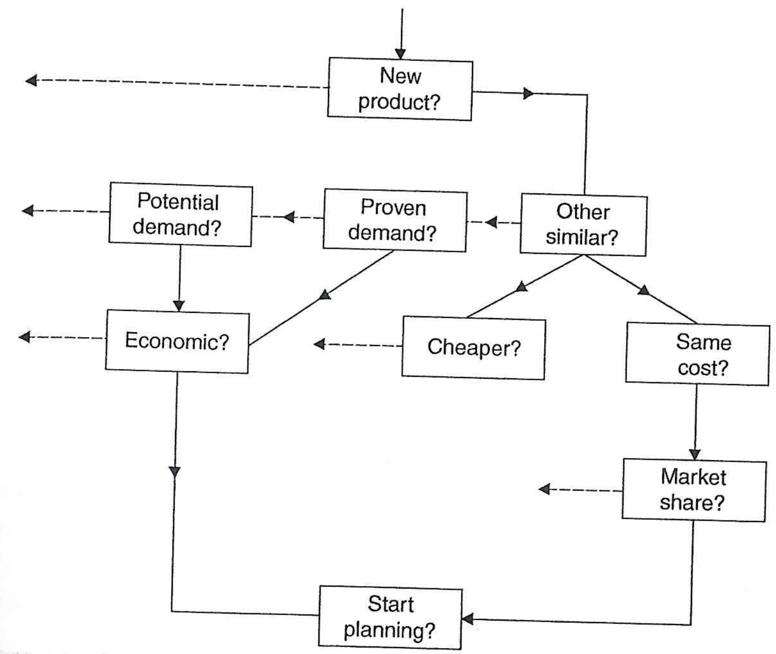
$$x = -E\frac{dx}{dt} - F\frac{d^2x}{dt^2}$$

Refer to Figure 16.2. The components used will decide values of constants. Scale factors have to be analysed relating voltage to variables – time constants allow real time scale variations. Output must equal input, so both are connected. An input force is necessary to provide essential acceleration and for this analogue is usually by square wave pulses from a signal generator. Output characteristics are analysed on an oscilloscope. Analogue variations (damping/mass and spring force/mass) are arranged at potentiometer and feedback resistor respectively. The circuit can be regarded from input to output as integration, or differentiation in the other direction. The last amplifier is a scale factor (which can also be arranged to have adjustable feedback).

It is now convenient to move from analogue (frequency domain) to digital (time domain) as used in logic circuits.

Logic Circuits

Logic is a systematic approach to problem solving and decision making. Figure 16.3 illustrates a simple logic flowchart analysis for marketing a new product (yes decisions full lines, no decisions dotted lines, that is, binary algorithm).



▲ **Figure 16.3** *Logic flow chart (algorithm)*

Boolean algebra

Is the algebra of logic. Has tended to the development of set theory, Venn diagrams, sentence logic and the logic of switching circuits. It is not possible to consider these aspects in any detail because notation and laws require involved consideration. However, Table 16.1 indicates the obvious relationship and the logic of switching circuits can be developed somewhat to illustrate principles. Only two input variables (A and B) are considered for simplicity but