FLUID PRESSURE MECHANISMS

BY

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LONDON

SIR ISAAC PITMAN & SONS, LTD.

1949

First published 1949

SIR ISAAC PITMAN & SONS, LTD.
PITMAN HOUSE, PARKER STREET, KINGSWAY, LONDON, W.C.2
THE PITMAN PRESS, BATH
PITMAN HOUSE, LITTLE COLLINS STREET, MELBOURNE
27 BECKETTS BUILDINGS, PRESIDENT STREET, JOHANNESBURG

ASSOCIATED COMPANIES
PITMAN PUBLISHING CORPORATION
2 WEST 45TH STREET, NEW YORK
205 WEST MONROE STREET, CHICAGO
SIR ISAAC PITMAN & SONS (CANADA), LTD.
(INCORPORATING THE COMMERCIAL TEXT BOOK COMPANY)
PITMAN HOUSE, 381-383 CHURCH STREET, TORONTO

MADE IN GREAT BRITAIN AT THE PITMAN PRESS, BATH
D9—(T.487)

PREFACE

THIS book deals with the mechanism of fluid pressure equipment and systems—that is, hydraulic and pneumatic machinery. It is not concerned with detail design nor constructional features of the equipment, but with the essential principle of the various devices, how they work and how they differ from each other. It is believed that a proper understanding of the fundamental mechanisms involved is necessary if a hydraulic or pneumatic engineer is to be able to design and use such equipment. An attempt has, therefore, been made to classify the various devices on a logical basis and to standardize nomenclature and the methods used in the drawings. As there is a certain amount of overlapping between various similar components or systems, cross referencing frequently occurs, and in general, a reader may find that a component described in early chapters is mentioned again in later chapters when the component occurs in a particular system.

It is true that many of the devices described in the text have not been found to be satisfactory for reasons not referred to; in other cases the mechanism may have never actually been made and tested. No apology is offered for these shortcomings which arise owing to the author's limited knowledge. Considerable research has been necessary to find and classify the 500 odd devices which are described and illustrated; to test and investigate all the unusual or obscure ones would indeed be a formidable task quite outside the author's abilities, or facilities. It is hoped that the descriptions will stimulate interest and thought, and that a hydraulic engineer setting out to design a relief or transfer valve, for example, may at least be made aware of the state of the art if he is not already acquainted with it.

In order for a book of this nature to be useful it must be accurate and comprehensive. In the event of future editions being found possible omissions will be repaired, additions made, and mistakes, let it be hoped few in number, corrected. To this end comments and suggestions will be valuable and will be most gratefully received.

Sources of information have been many and varied, too many for individual acknowledgment. Special thanks are due, however, to Mrs. H. M. Wheeler for an exceptional job of tracing the author's diagrams.

H. G. CONWAY

January, 1949

CONTENTS

PART I: MECHANISM

CHAPTER 1

INTRODUCTORY

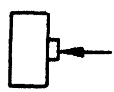
The scope of the book covers hydraulic, pneumatic and compressed gas systems of high, medium and sometimes low pressure, but not specifically such items as household water supply, which is hardly a "fluid pressure" system.

1.1. Conventions. All diagrams have been simplified as far as possible to exclude all details of design irrelevant to an understanding of the mechanism involved. To avoid unnecessary repetition on each diagram of the direction of flow of fluid, etc., by description, the following simple convention of signs has been adopted.

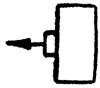
(*a*) Single direction flow is indicated by a single arrow on the pipe line thus—

(*b*) Two-direction flow is indicated thus—

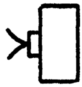
(*c*) The connection from the pump or source of pressure on a component is indicated by inward flow thus—



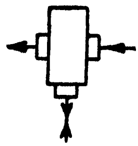
(*d*) The connection back to tank, etc., is indicated thus—



(*e*) In the case of air systems, where the air is exhausted to atmosphere, the exhaust connection may be indicated thus—



(*f*) The connection from a component (e.g. a selector) to a jack, etc., involving two-way flow, is indicated with a double arrow, thus—



1.2. Pressures. Although the pressure in a fluid-actuating system has fundamentally little effect on the operation of the system, a brief statement of present-day practice may be of interest.

As regards pneumatic systems, rotary compressors are reliable up to about 150 lb/sq in. and vehicle braking systems are in use at about 100 lb/sq in. maximum pressure. Two-stage piston compressors are available for aircraft purposes giving at least 1 000 lb/sq in. and pneumatic systems are in operation at this pressure on aircraft. There is evidence that compressors for aircraft at as high a pressure as 2 500 lb/sq in. will be available in the near future. For normal industrial purposes pneumatic equipment in the machine shop, etc., is generally limited to the 120–150 lb/sq in. of the shop air system, although often as low as 80 lb/sq in. Some shops, however, are provided with 1 000–2 000 lb/sq in. air lines, since high pressure three-stage industrial compressors are available.

Hydraulic equipment development has been encouraged by aircraft requirements. Although one or two power-press applications use pressure as high as 5 000–6 000 lb/sq in. (suitable industrial pumps being available), normal industrial and machine-tool applications are still using 500–800 lb/sq in. On aircraft 2 500 lb/sq in. is standardized in Britain, 3 000 lb/sq in. is current in the U.S.A. and in France, while a few aircraft have been fitted with 4 500 lb/sq in. equipment. There are indications that 4 000–5 000 lb/sq in. will be widely used in future aircraft.

Vacuum system pressures are obviously limited by atmospheric pressure. A good vacuum pump can produce — 10 lb/sq in. reliably, with a maximum of — 12 to — 13 lb/sq in.

1.3. Fluids. Any fluid, liquid or gas, could probably be used in a pressure actuating system, but for obvious reasons the common fluids are air and oil. Water can be used, but is now rare in industrial actuating systems.

Air is commonly used since it is so readily obtainable and can be exhausted directly to the atmosphere. It is comparatively hygroscopic (i.e. absorbs water), and because the moisture may be released as the air expands through the selector, valves, etc., condensation and even freezing in cold weather may result. To prevent this on aircraft and vehicle braking systems, the usual procedure is to inject or otherwise saturate the air with alcohol vapour, which depresses the freezing point sufficiently to eliminate this trouble.

Liquids used are light mineral oils with low setting points (with synthetic rubber seals, etc.) or special castor oil base fluids (with natural rubber glands). These fluids are generally castor oil—alcohol or equivalent solvent mixtures, the castor oil being too viscous and having too high a setting point by itself. Other liquids are blended in for special reasons, for example, to increase

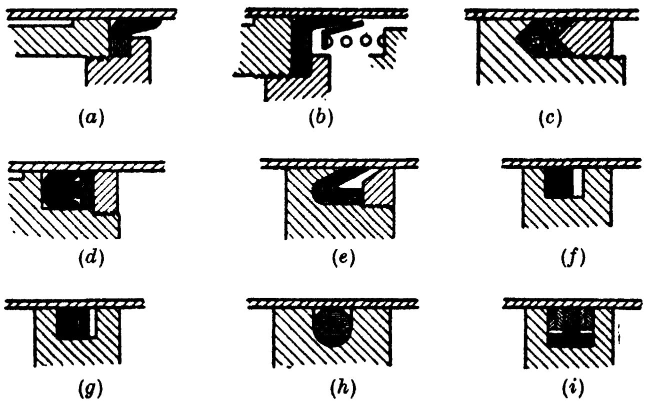


FIG. 1. TYPES OF MOVING SEAL

the miscibility of the oil and alcohol, or to enhance lubrication properties.

Ethylene glycol base fluids are sometimes used owing to their low freezing points, but they have very poor lubrication properties. Mixed with water, glycol may make a good non-inflammable fluid.

Synthetic fluids, such as "Silicone Oil" are being developed with remarkable characteristics including very small change of viscosity with temperature.

In certain chemical plants unusual liquids, such as molasses, coco-nut oil, etc., have been used in hydraulic systems.

1.4. Seals. All pressure actuating and actuated equipment uses seals of one type or another to prevent loss of fluid (gas or oil). Metal-to-metal seals are made either by compression of the two members with, for example, a gasket between them, or by close fitting parts, lapped together, on a slide or plate valve.

Fixed seals can also be made with rubber or other flexible joints (i.e. a copper asbestos gasket) or by such device as flexible pressure filled rings (Wills rings), or by any of the moving seals described below.

The main types of moving seal are shown in Fig. 1. Initial sealing against low pressure requires the glands to be fitted with

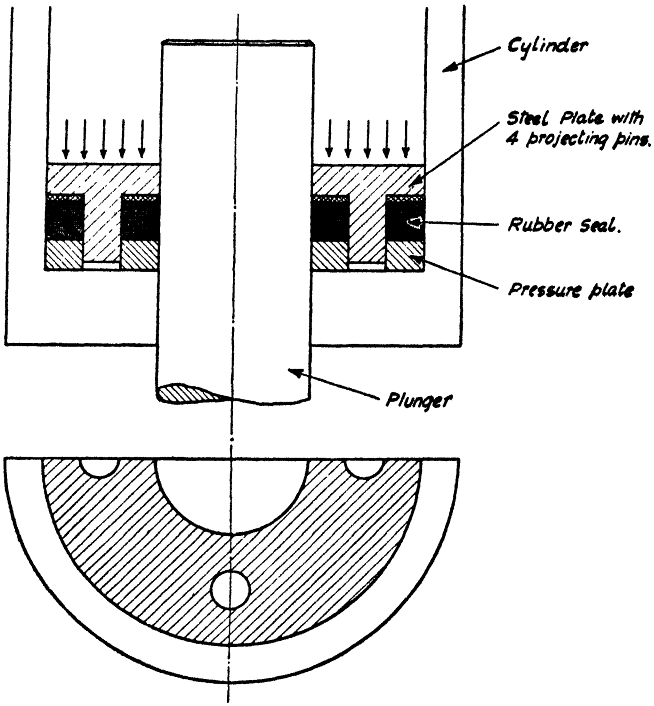


FIG. 2. ULTRA HIGH PRESSURE SEAL

The effective sectional area of the rubber seal is less than that of the pressure plate; this reduction is due to the area occupied by the four pins. Thus the hydrostatic pressure in the seal is greater than the hydraulic pressure.

radial pressure between the seal and moving wall; in the case of types (*a*), (*d*) and (*e*), this is obtained by a bending stress in the seal due to its lip being larger initially than the cylinder bore; in type (*b*) any bending stress is augmented by a spring loaded expander cup. In the case of type (*c*), and sometimes with types (*d*) and (*e*), the radial stress is obtained by tightening the gland longitudinally.

In types (*f*), (*g*), (*h*) and (*i*) the seal is elastic (e.g. rubber), and the initial stress is obtained by the ring being larger than the cylinder before fitting, so that when fitted the section of the seal is squeezed out sideways. Type (*f*) is of simple rectangular section; type (*g*) is similar, but has two rings, the outer harder than the inner, the hard one for wear and the soft one for resilience. Type (*h*) is the same as type (*f*) but of greater section, while type (*i*) is of inverted tee section, with metal, plastic, or even leather support rings on either side of the narrow part of the tee in contact with the moving wall. Similar support rings are known with types (*f*) and (*h*).

Under pressure the seals, types (*a*) to (*e*), are pressed firmly on to the moving wall by virtue of the unbalanced area exposed to the pressure. In the case of types (*f*) to (*i*) the material being flexible, when the side face of the seal is exposed to the fluid pressure, the seal is put in a state of hydrostatic stress and the radial pressure between the seal and the moving wall increases accordingly. A ring, such as the O-seal of type (*h*), which does not fit the groove side face closely, will be distorted under pressure until it fits the contour of the groove closely. The permissible clearance behind the ring between piston and cylinder depends on the flexibility of the seal. In the case of rubber seals, the material will extrude down any appreciable clearance.

The friction of the seal is a function of the coefficient of friction of the seal material, the radial pressure and the area of the seal in contact with the cylinder; narrow rings have obviously less friction than wide ones.

Fig. 2 illustrates an interesting seal where the hydrostatic stress in the rubber is actually greater than the fluid pressure, by virtue of a difference in areas on the pressure plate. This seal operates reliably at pressures of 60 000 lb/sq in.

CHAPTER 2

VALVE ELEMENTS

THE essential mechanism of all valves and equivalent devices is incorporated in almost all hydraulic or pneumatic equipment. Applications range from simple non-return valves, where usually a valve element by itself can be used, to selectors where several valve elements are combined and operated deliberately to transmit fluid in various predetermined directions.

Owing to the ubiquity of these elements they should be fully understood before the practical applications in the various components of a pressure system are studied.

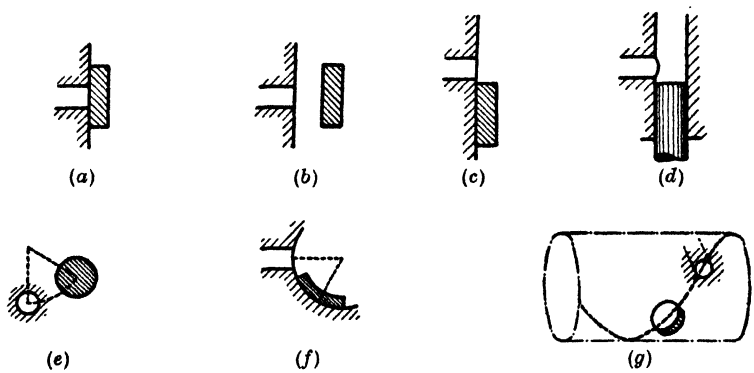


FIG. 3. THE BASIC VALVE ELEMENTS

2.1. The Basic Valve Elements. All valves can be analysed to consist of a fixed body containing a hole or port, covered by a valve which moves away from the port to permit flow of fluid, gas, or oil through it. The valve can move along the axis of the port or more or less at right angles to it. Fig. 3 (*a*) shows the basic lift valve mechanism as already defined; (*b*) illustrates opening of the port by motion of the valve in the axial direction; this is the motion of most poppet or ball valves of usual type. Included in this category is the hinged type of valve which folds away along an arc of a circle, since the instantaneous motion is axial.

In (*c*) is illustrated the slide type of valve where the motion of the valve is normal to the axis of the port. An equivalent mechanism is the plunger slide type shown at (*d*), where the valve section is circular; (*c*) can be considered as a particular case of (*d*), where the diameter of the plunger is infinite. The so-called sleeve valve is a particular case where the valve is annular in section instead of circular.

Diagram (*e*) illustrates a further variation of (*c*), where the valve moves on a circular arc; this is used in the rotary disc type valve. Again type (*c*) may be considered as a particular case of (*e*), where the radius of operation is infinite.

Diagram (*f*) illustrates another variation where the valve moves in a circular arc in the other plane, i.e. over a cylinder. Strictly speaking, it need not be a circular arc, but non-cylindrical surfaces have no practical interest. Again (*c*) can be considered as a particular case of (*f*).

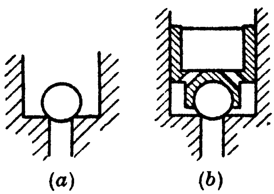


FIG. 4. BALL VALVES
(*a*) = Unguided. (*b*) = Guided.

As a final simplification, it will be seen that all cases, except (*a*), are particular cases of *helical* motion of the valve element (*g*), the radius of operation sometimes being infinite and the helix angle being 90° or zero. The helical solution itself, where the valve turns and slides is known but has little practical value.

Using the terminology of kinematics, the general solution (*g*) is that using "screw pairs", cases (*e*), (*f*) are "turning pairs", and cases (*c*), (*d*) "sliding pairs".

2.2. Axial or Lift Valve Elements (Type (a)—Fig. 3). The common valves of this type are—

The ball valve.
The poppet valve.
The inverted poppet valve.
The flat or plate valve.

These exist in more or less complicated variations and are either automatic in the sense that they open automatically, due to the flow of fluid in one direction, closing in the reverse direction (i.e. the "non-return valve"), or are opened deliberately by application of external forces. In the latter case they then become cocks, or selectors, as will be explained later.

BALL VALVES. The common ball valve (Fig. 4) is so widely known as to need little description. Diagram (*a*) is the usual free ball, usually spring loaded into its seat, while (*b*) is a variation where the ball is guided along the axis of the port, although it can usually, though not necessarily, turn round.

Ball valves are popular because of the cheapness of the ordinary steel ball-bearings. They are, however, not used on high quality equipment, because their inherent freedom is, in fact, not an advantage. The working face of the ball in contact with the valve seat becomes marked in use, and when the ball turns round this mark lies across the seat, setting up a leak. The guided ball valve is better in that sideways wobble of the ball is prevented, wobble being serious when the valve is just open. Stainless steel, or non-ferrous balls, are sometimes used.

Non-automatic ball valves are usually lifted off their seats by means of a small spindle or rod which passes through the port

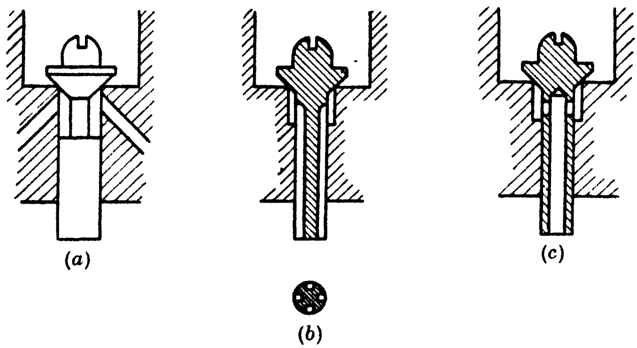


FIG. 5. POPPET VALVES
(a) = Necked. (b) = Fluted. (c) = Hollow.

and lifts the ball. The guided ball type can be lifted off its seat by means of the guide.

POPPET VALVES. The principal types of poppet valve are shown in Fig. 5. Type (*a*) is the common mushroom valve, with a chamber under the valve head formed by relieving the valve stem or by counter-boring the valve throat. The fluid passages under the valve are formed in the port body.

Type (*b*) is similar, except that the fluid passages are formed in the stem by slots or flats.

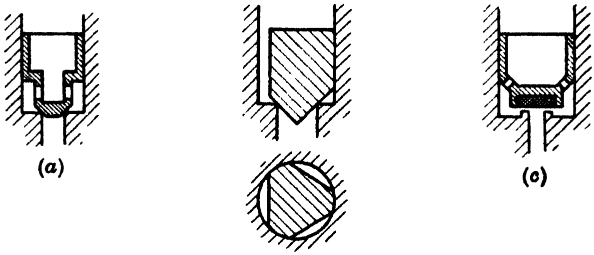
Type (*c*) is a third variation using a hollow stem to admit the fluid.

Although the valves illustrated have 45° face angles, the usual angle for hydraulic valves, 60° is also common, and 30° is often used for gas valves, as, for example, on internal combustion engines. True mushroom valve with a flat head (angle = 0) was, and is, met with occasionally.

The main advantage of a poppet valve of this type is that the valve is fully guided with two degrees of freedom only, i.e.

axial lift and rotation. Thus when the valve becomes marked, no trouble with leakage occurs as with a ball valve unless the valve seat is not of constant width (e.g. eccentric); in this event the marking of the valve face will not be of equal width, and should the valve turn round the wide mark may overlap the narrow seat and allow leakage. Although it is possible to constrain the valve against rotation this is likely to interfere with the free motion of the valve, and it is better practice to take care that the valve seat is concentric and of regular width.

A second advantage of the poppet type is that, while it is not difficult to manufacture the port and valve accurately and with sufficient quality of surface finish, it is relatively simple to lap



(a) (b) (c)

FIG. 6. INVERTED POPPET VALVES
(a) = Hollow. (b) = Fluted or with flats. (c) = Inserted seating.

or grind away any remaining imperfections or those resulting from wear in use during assembly. Slots or holes for lapping purposes are usually provided in the head.

Although for gas valves seat widths measured along the valve face are generally of the order of 0·05–0·1 in., for high pressure hydraulic work "knife edge" seats not wider than 0·02 in. give best results.

Poppet valves are generally lifted by pushing on the valve stem, although it is possible to lift them from the head.

INVERTED POPPET VALVES. The inverted poppet valve differs from the ordinary poppet valve in that it is guided beyond the valve face (Fig. 6).

Type (a) is a typical one, holes being drilled in the valve to admit the fluid to the centre of it and thus away. Type (b) is an alternative, with flats (or slots, etc.) on the edges of the valve to pass the fluid; if adequate area is provided there is a danger that the guiding of the valve will suffer. For this reason also a small even number of slots should be avoided.

As with the normal poppet, face angles of 60°, 45°, 30°, and

0° are common; angles as high as 75° are known for special purposes.

Type (*c*) shows a common type of flat valve with zero face angle. In this case a soft seat (usually rubber or fibre) is used in conjunction with a fairly large seat area.

The problems of manufacturing inverted poppet valves are those of the normal poppet, except that for proper guiding the valve seat and guiding diameter should be in the same component

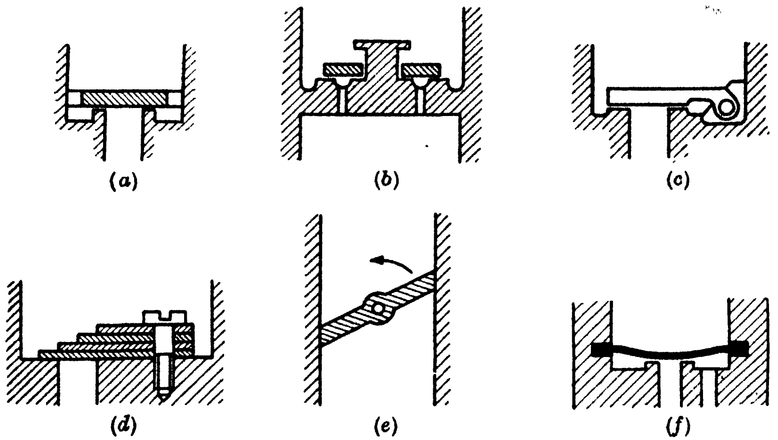


FIG. 7. FLAT VALVES

(*a*) = External guide.
(*b*) = Internal guide.
(*c*) = Hinged.
(*d*) = Laminated.
(*e*) = Butterfly.
(*f*) = Flexible diaphragm.

to ensure concentricity, and this means that the seat is at the bottom of a hole. Thus the inverted poppet assembly is slightly more difficult to manufacture than the normal type.

The inverted poppet is opened deliberately in the same way as a ball valve by a push rod, which masks the valve throat area to some degree. Lifted valves are common, particularly solenoid-operated types.

FLAT OR PLATE VALVES. Various typical plate valves are shown in Fig. 7. Type (*a*) is a plate guided at its edges; (*b*) on a central spigot; (*c*) is hinged; type (*d*) is formed by a laminated spring, and type (*e*) is the common carburettor butterfly valve, which however may be considered as a type of rotary valve. There is a whole range of valves where the valve element is held in, or formed by, a diaphragm, as type (*f*), instead of being guided mechanically, but these are referred to later under "Cocks".

Flat valves are of interest for special applications only, owing to relative difficulty in making them tight against high pressure.

Lapped metal valves are known, but rubber or fibre faced valves are more common, since the elasticity of the valve covers imperfections of the seat. The normal household tap comes in this category.

Hinged valves, type (*c*), are rare nowadays, as they are imperfect, for unless the seat is flexible the valve faces only coincide with the valve in one particular plane, and errors in manufacture or wear tend to throw the two planes of valve and seat out of coin-

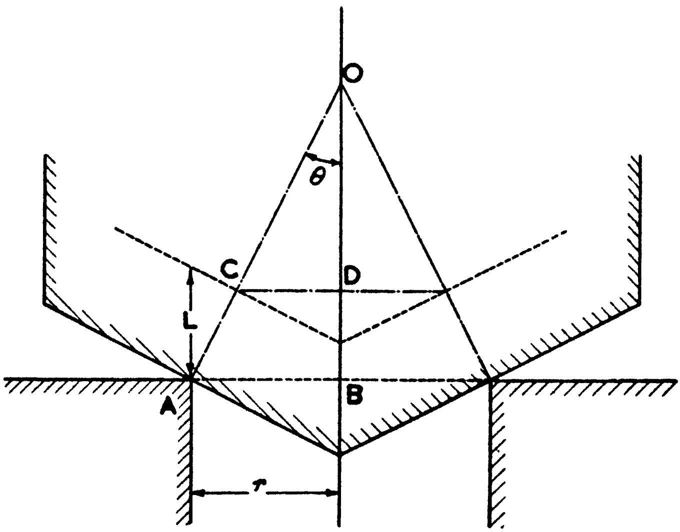


FIG. 8. THE DEGREE OF MOVEMENT REQUIRED ON A LIFT VALVE

Area through valve = Area AC.
= Area of cone OAB − area of cone OCD.

$$\therefore L = \frac{1 \pm \sqrt{1 - \sin\theta}}{\cos\theta \sin\theta} r$$

If $\theta = 60°$, $L = 1{\cdot}48r$ = 74 per cent of throat diameter.
= $45°$, $L = 0{\cdot}92r$ = 46 per cent of throat diameter.
= $30°$, $L = 0{\cdot}67r$ = 34 per cent of throat diameter.
= $0°$, $L = 0{\cdot}5r$ = 25 per cent of throat diameter.

cidence. Looseness at the hinge is therefore necessary. This type of valve, however, where screw pressure, etc., effects sealing, and a loose hinge is used to allow the valve to be swung aside, is common on tank or vessel filler caps, etc. Plate valves to be opened deliberately generally require push rods through the valve throat, except perhaps in the case of the hinged type.

2.3. Degree of Movement Required on Lift Valves. The information in Fig. 8 gives the required lift of a poppet, or plate valve, such that the area through the open valve is the same as the throat area of the valve port, assuming that the valve seat face is thin and ignoring any restriction of the valve throat due to a valve stem, etc. A ball valve can be taken as a 45° poppet as a first approximation, although the exact value depends on the relative sizes of ball and valve throat; usually the ball sits on the seat at about 45° seat angle, i.e. the ball is $\sqrt{2} \times$ the throat diameter, and in this case the required lift is about 38 per cent of the throat diameter.

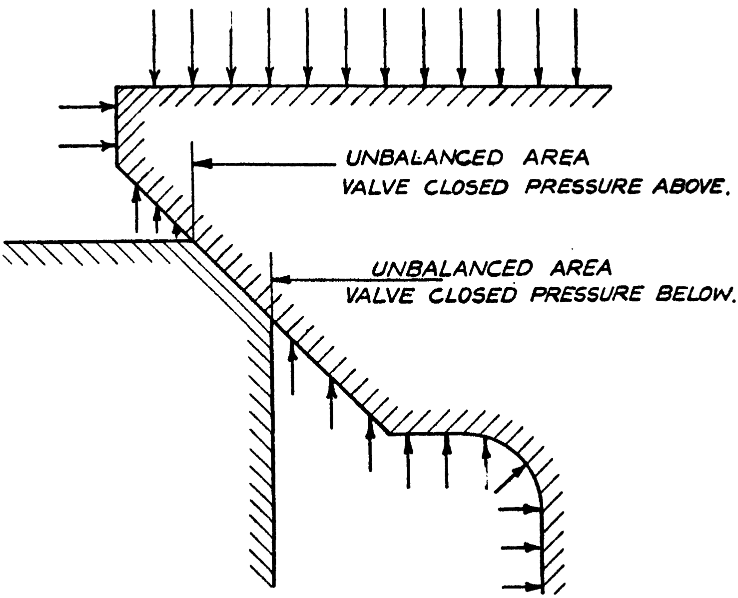


FIG. 9. THE FORCES ACTING ON A LIFT VALVE

The advantage of a relatively flat valve angle or large ball from the point of view of lift will be seen, although as manufacturing difficulties increase as the angle is reduced, 45° is a reasonable compromise for metal valve seats which must be lapped or ground in. In the case of soft (e.g. rubber) seats, full advantage of the flat valve can be taken.

2.4. Balance of Forces on a Lift Valve. If the forces acting on a lift valve when under pressure are considered, it will be seen that the net force required to open the valve against pressure is equal to the pressure multiplied by the unbalanced area (see Fig. 9). With a seat of finite width, the unbalanced area is the outside circle of the valve seat, whatever the angle of the seat. In the case of pressure below the valve, the unbalanced area tending to lift the valve is the inner circle of the valve seat.

Therefore with practical valves the force required to lift the valve against pressure is greater than to hold it against pressure.

Fig. 10 illustrates the general case of fluid flowing through a valve with pressure p_1 above the valve dropping to p_2 as it passes through the valve, as in a restriction valve. In many cases p_2 is zero, or almost zero when the pressure is released through the valve. Whatever the value of p_2 there must be a drop in pressure through the valve of the type shown diagrammatically in the illustration, and this will cause a mean upward thrust on the

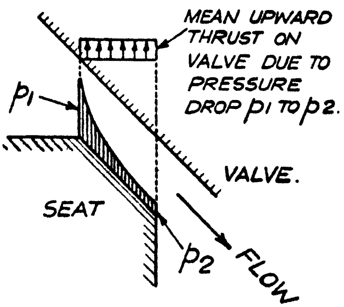


Fig. 10. Balance of Forces on a Lift Valve—Inward Flow

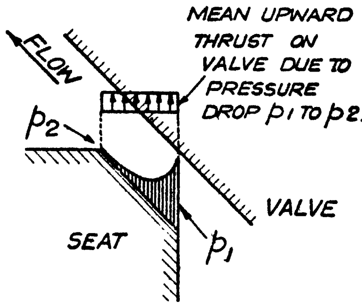


Fig. 11. Balance of Forces on a Lift Valve—Outward Flow

valve, its magnitude depending on the ratio of p_2 to p_1, and the length of the seats, etc.

Thus the balance of forces on the valve is—

Downwards: $p_1 \times$ outer seat circle.
Upwards: $p_2 \times$ inner seat circle + mean upward thrust on valve seat area.

It is thus obvious that even where p_2 is zero, it requires less force to hold the valve open than to open it. This fact is important as will be seen later when considering relief valves.

Fig. 11 illustrates the same conditions when the direction of flow is from underneath the valve, lifting it.

Again in this case the balance of forces is—

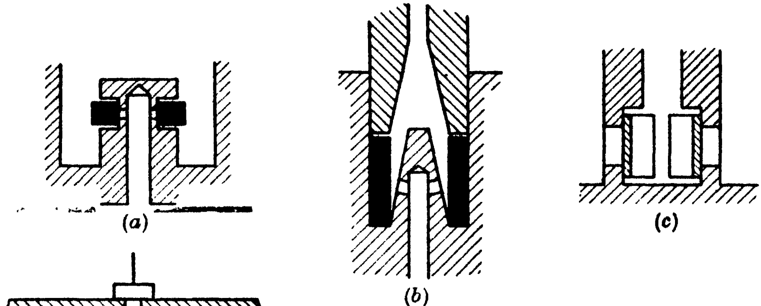
Downwards: $p_2 \times$ outer seat circle.
Upwards: $p_1 \times$ inner seat circle + mean upward thrust on valve seat area.

It is again obvious that even where p_2 is zero, it requires more force to close the valve once it has opened than to open it.

This effect on a valve is sometimes referred to as the pressure "differential" of the valve.

2.5. Other Types of Lift Valve. These may be specified as follows—

Radial Expansion Valves. Although several lift valve types could be arranged to move radially away from a port in a cylindrical body, the most useful application is that shown in Fig. 12 (*a*). In this a rubber ring fitted with radial tension is blown outwards by the pressure, allowing the fluid to escape sideways. This is the mechanism of the ordinary cycle valve. Type (*b*) is a valve closed by the compression of the rubber ring or tube, and type (*c*) a valve (usually a suction valve), which is formed by a



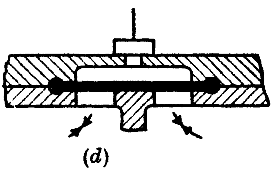


Fig. 12. Radial Expansion Valves

(*a*) = Flexible and expanding. (*c*) = Spring.
(*b*) = Flexible and contracting. (*d*) = Diaphragm.

thin spring steel ring which collapses under inwardly directed flow.

Type (*d*) shows another expansion valve which by lifting the diaphragm between two series of radial slots, offers restriction to flow in either direction proportional to any pressure which may exist in the chamber outside the rubber sleeve. It is obviously suitable for special applications only, for example, as a relief valve, which has to pass fluid containing a certain amount of solids.

Pilot Valves. A most useful device in practice is to use two valves together, one acting as pilot valve for the other. Fig. 13 (*a*), (*b*) show two common types, and it can be seen that by opening the small pilot valve, using a small force, the pressure on the main valve is released so that it can be easily opened. However, since a larger flow through the pilot valve will cause a build-up in pressure due to the restriction through its small throat, there is a practical limit to the ratio of the pilot valve area to the main valve area. Type (*c*) is an inverted pilot valve which is lifted from its seat. Types (*d*) and (*e*), which together form a three-way selector, are interesting pilot-operated relay valves, without the disadvantage of flow restriction, since opening the pilot valve unbalances the valve and opens it automatically. There is a servo action with this type of valve which therefore differs from the simple types in this respect.

Double or Dual Seal Valves. Two valves are sometimes used in series for special reasons. In Fig. 14 (*a*) is a type of valve

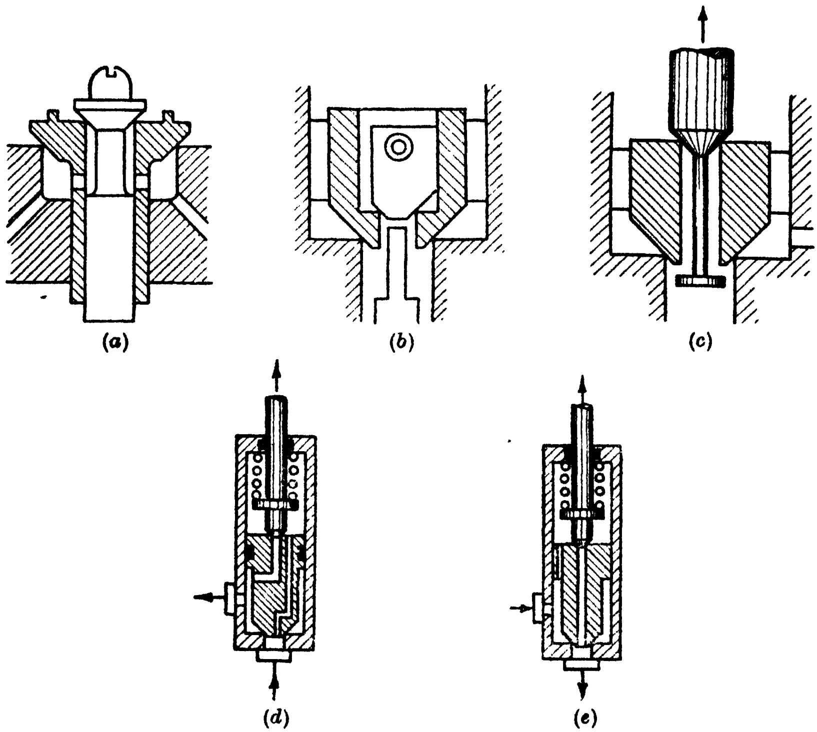


Fig. 13. Pilot Valves

where the normal poppet valve is used for sealing purposes, and a small lift ball valve is used to meter the flow through the assembly. Since very small poppets are difficult to make, a small $\frac{1}{8}$ in., or even $\frac{3}{32}$ in., ball forms an excellent metering substitute. In (*b*) is shown a modification of the valve illustrated in Fig. 5 (*c*); the ports communicating with the inner passages are drilled below the edge of the counter bore, and thus before any appreciable flow can take place the valve must open comparatively widely. This is supposed to prevent erosion of the main seat.

In (*c*) is shown the combination of a metal poppet valve and a rubber seal which enters the valve body as the valve shuts. This can be made absolutely leak-proof, provided due care is taken that the seal does not tear as it enters the recess.

Another construction has a conical valve seating on a wide conical seat, a circular or rectangular section seal being carried in a groove in the valve cone, and sealing the two mating surfaces of valve and seat. The sealing is probably mainly due to the sealing ring since the valve seating is very wide and

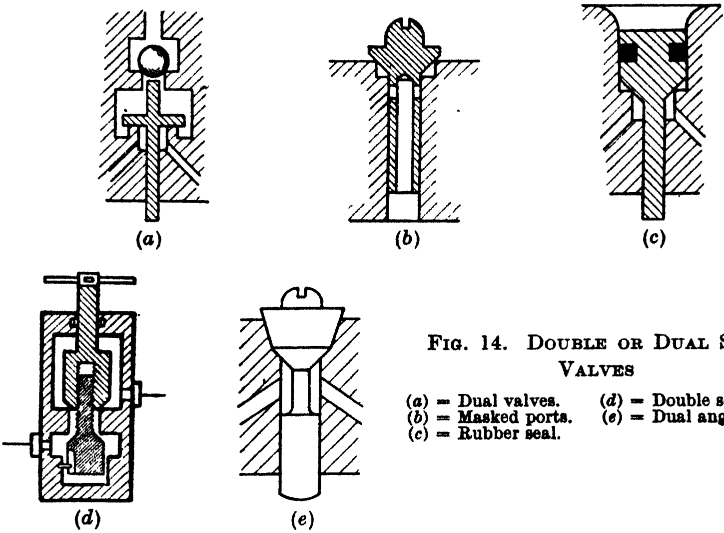


FIG. 14. DOUBLE OR DUAL SEAL VALVES

(*a*) = Dual valves. (*b*) = Masked ports. (*c*) = Rubber seal. (*d*) = Double screw-up. (*e*) = Dual angle.

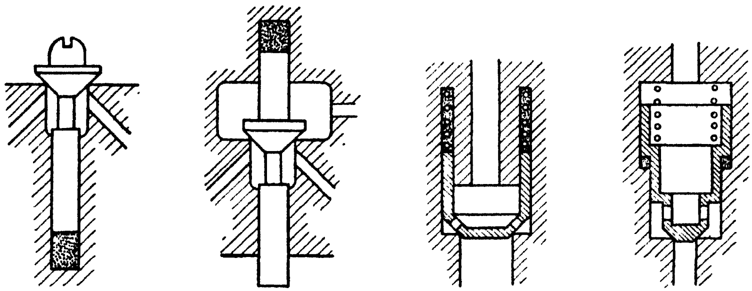
thus such constructions are scarcely worthy of classification as "dual-seal".

Type (*d*) is a double valve, the two parts of the valve being clamped tightly to either side of the diaphragm forming the seat by a screw thread. Type (*e*) is not strictly a dual seal valve, but performs the same function as type (*b*); the normal cone acts as a sealing surface, and the pointed cone permits more precise metering of the flow; the design is obviously difficult to realize.

Numerous other combinations exist and will suggest themselves.

DAMPED VALVES. Valve motion is sometimes damped, some of the means being shown in Fig. 15. The methods of (*a*) and (*b*) allow fluid to seep up the valve stems into a chamber in which the stem forms a plunger, and thus action will depend on the rate of loss of fluid from the chamber. Illustrations (*c*) and (*d*) show other solutions used on poppet or non-return valves, the damping being against opening in the case of (*c*), and closing in the case of (*d*).

The efficiency of any damper of this type is doubtful since it depends on the volume involved being full of oil when needed,



(*a*) (*b*) (*c*) (*d*)

FIG. 15. DAMPED VALVES

and this is not necessarily so. Damping action against opening may be desirable on such applications as reducing valves, while damped non-return valves have been used to prevent excessive shock loads as the valve seats.

BALANCED VALVES. A balanced valve is one where the pressure acts on the valve on a net area much less than the valve seat or

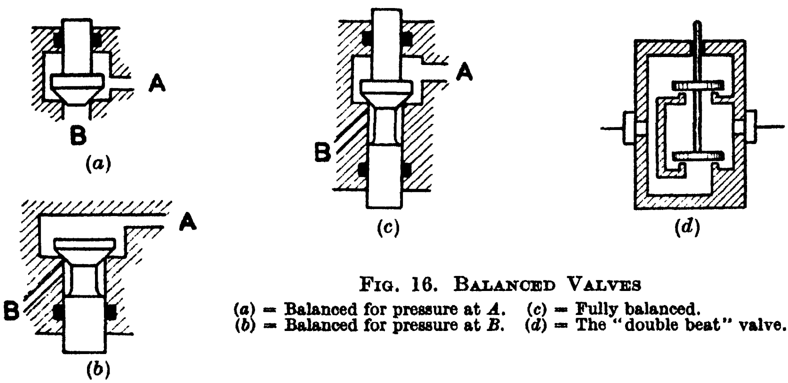


FIG. 16. BALANCED VALVES

(*a*) = Balanced for pressure at *A*. (*c*) = Fully balanced.
(*b*) = Balanced for pressure at *B*. (*d*) = The "double beat" valve.

throat area, and other things being equal it requires much less force to operate (or even, if perfectly balanced, almost no force). This is accomplished by using a sealed area producing a force opposing the pressure force on the valve.

Fig. 16 (*a*) shows a valve balanced for pressure at *A* only, by sealing the guide diameter, which is the same size as the valve throat. Apart from friction or spring loads the valve is thus fully balanced. It should be noted that once this valve is opened, it is overbalanced and has to be forced shut (see below).

The solution of (*b*) is similar to (*a*) except that it is balanced for pressure under the valve (at *B*). In (*c*), however, is shown a valve balanced for pressure at either connection. The preferred construction for this type of valve has the upper spindle articulated to the head of the valve so as not to constrain the valve from seating centrally.

Type (*d*) is the "double beat" valve of steam engine practice, and is more or less fully balanced. The area of the lifting rod will approximately balance the area of the seat for flow around

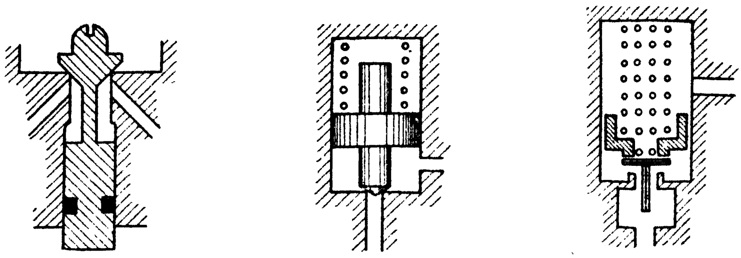


FIG. 17. OVERBALANCED VALVES

(*a*) = Overbalanced on stem. (*c*) = Overbalanced once open.
(*b*) = Overbalanced once open.

the lifting stem, but if the seat width is narrow the valve will be balanced for flow in the other direction also.

In actual practice normal valves are usually not fully balanced, but the balance diameters are chosen so that a reasonable net force remains between valve and seat, yet leaving the opening load reasonable.

OVERBALANCED VALVES. Overbalanced valves are used for special purposes, as will be seen later. Fig. 17 shows three solutions. That in (*a*) is to enable pressure to be applied underneath a valve without opening it automatically. The device of (*b*) is a special version of Fig. 14 (*a*), such that as soon as the valve opens, due to pressure underneath the valve, the fluid is exposed in the large piston area and opens the valve widely. This presupposes that there is a restriction or equivalent at the valve outlet. The use of this valve will be seen in cut-out valves, described later.

The element of (*c*) is a valve held to its seat by two springs. As soon as the pressure underneath the valve overcomes the total spring load, the valve assembly lifts and then the pressure separates the upper plate from the valve poppet, the plate being a good fit in the bore of the spring housing. Under these conditions a pressure exists between the two valves, balancing the outer spring load and allowing the poppet valve to remain relatively widely open.

2.6. Miscellaneous Valve Elements. Two other valve elements of special type are worth mentioning. Fig. 18 (*a*) is the well-known piston seal cup used on early pumps, modern car brakes,

(*a*) (*b*)

FIG. 18. MISCELLANEOUS VALVE ELEMENTS

(*a*) = Collapsing cup. (*b*) = Rectangular section transfer valve.

and other gear. In the forward direction the piston seal acts normally and leakage is prevented. In the reverse direction the cup collapses slightly to allow fluid to flow past the cup. It thus acts as a non-return valve and piston seal at the same time.

In (*b*) is shown a rectangular section seal which acts in one direction as a piston seal, covering up the small holes in one side of the groove, and in the other direction allows fluid to flow past the ring and out the holes in the other side of the groove, which are not masked by the ring. A circular section seal can be used in place of a rectangular seal, and may in fact be preferable since it will roll from one position to the other easily.

CHAPTER 3

SOURCES OF PRESSURE—PUMPS

FUNDAMENTALLY most pumps suitable for pumping hydraulic fluid will operate with air or gas, although most oil pumps make poor air compressors, and some compressors will not operate with incompressible fluid without modification.

3.1. General Considerations. The main requirements of a hydraulic pump capable of delivering sufficiently great a pressure to be of practical use, are—

(*a*) Means for controlling fluid leakage either by special seals or close working clearances.

(*b*) A substantially uniform, non-pulsating delivery.

(*c*) The absence of trapping of fluid during rotation, since the fluid is for all practical purposes incompressible.

(*d*) Mechanical balance to a sufficient degree to enable the pump to be run as fast as possible, to reduce bulk for a given output.

(*e*) Adequate provision for minimizing the effect of distortion of the working parts, casings, etc., due to the internal pressure in the pump.

(*f*) Low clearance volumes so that the pump will act momentarily during priming, or with aerated fluid, as a compressor, to discharge the air through the delivery line, instead of leaving it trapped in the pump.

(*g*) Adequate inlet valve or port provision to enable the pump to create a good suction, or alternatively, as in the case of aircraft, to operate at high altitude, and with relatively thick fluids. It should be noted that the time of opening of the inlet valve or port is almost as important as the size of the port.

The main requirements of a good compressor are—

1. The smallest possible clearance volume compatible with mechanical clearance; this is because any gas compressed in the clearance volume will expand again instead of being discharged, and will thus adversely affect not only the output of the pump, but the maximum pressure that can be achieved.

2. Except on low-pressure pumps, two or more stages of compression with as much intercooling between stages as possible. This is desirable, firstly, to reduce the wastage of work and output due to the gas being compressed adiabatically instead of

isothermally, and, secondly, to reduce the overall loss of output due to clearance volume.

3. Pre-compression of the air in the pump up to the pressure of the delivery line. Although the normal reciprocating pump with an automatic or spring-loaded delivery valve will ensure this, some of the rotary compressors have to be ported specially to obtain pre-compression, and with some it is not possible at all. Without pre-compression, as soon as the delivery port opens, the gas in the delivery pipe expands back into the pump, and the whole volume has again to be compressed. A rotary compressor with pre-compression cannot be operated with hydraulic fluid.

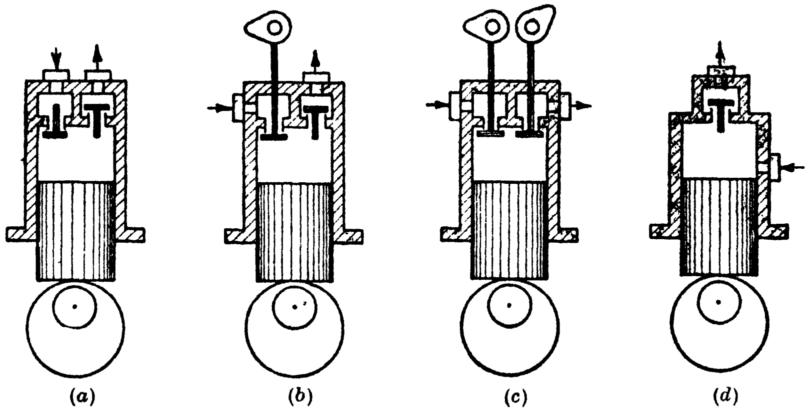


FIG. 19. ELEMENTARY TYPES OF PISTON PUMP

4. Provision for air or water cooling of the cylinder so that the temperature of the air in the compressor may be kept as near isothermal as possible, and the volume of air being delivered correspondingly increased. This is a feature not normally necessary on a hydraulic pump.

5. Means for controlling gas leakages during compression either by piston rings on a reciprocating piston compressor, or fine clearances on the rotary types.

6. Adequate inlet valve area to prevent throttling of the incoming air, with the consequent loss of volumetric efficiency.

The main types of pump and compressor use pistons, meshing gears or rotating vanes, these having been found to give good high pressure performance. Other types of pumps, such as centrifugal pumps or turbo-compressors, are not used for pressure systems as they do not produce high enough pressure to be useful.

Piston pumps are capable of at least 10 000 lb/sq in. (e.g. Diesel injector pumps), gear pumps 2 000–2 500 lb/sq in., and vane types 1 000–1 500 lb/sq in. when operating on oil. When pumping air the maximum pressures are about 3 000, 75, and 150 lb/sq in. respectively.

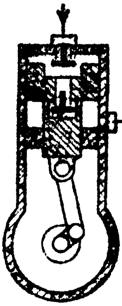


FIG. 20. A TWO-STAGE COMPRESSOR

3.2. Piston Pumps. The mechanical features of piston pumps are well known and require little description. Fig. 19 shows the principal elements. Type (*a*) has both valves spring loaded so that they open automatically. The restriction of the inlet valve spring pressure is objectionable on an air compressor, and undesirable on a hydraulic pump, particularly for high suction lift or high altitude work. Type (*b*) is better in that the suction valve is mechanically opened, while type (*c*) has both valves mechanically opened, perhaps an unnecessary refinement on a pump, although important for a motor. Type (*d*) shows the principle of the Diesel injector pump and other hydraulic pumps, the suction ports being exposed by the piston. This gives satisfactory performance, except at high speed or high altitude due to the relatively short time of opening of the valve.

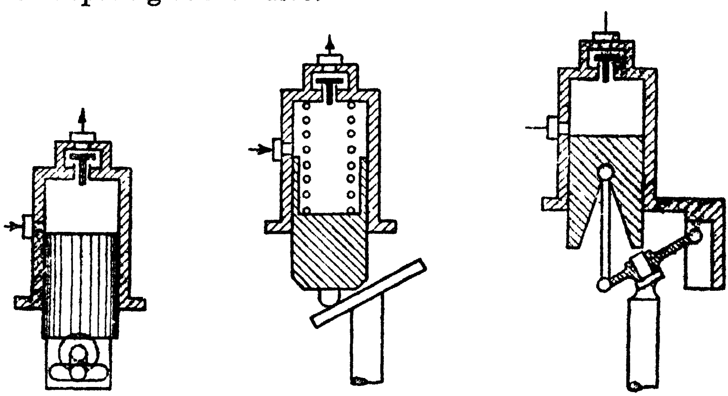


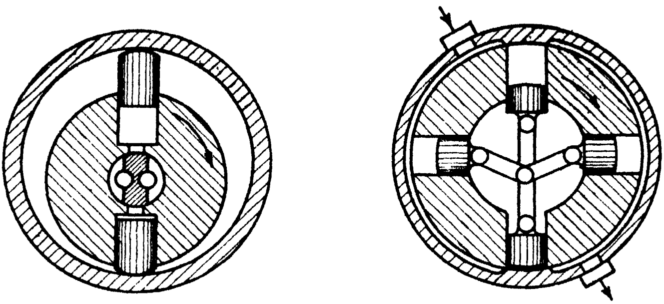
FIG. 21. PISTON RECIPROCATION MECHANISMS
(*a*) = Crank. (*b*) = Swash plate. (*c*) = Z crank or wobble plate.

Although the diagrams show poppet valves, any other type of valve can be used (e.g. slide or rotary valves).

Hydraulic pumps operating at 5 000–6 000 lb/sq in. on these principles are known, and they use sealless pistons, fitted closely by lapping, etc., to the cylinder bores.

Fig. 20 shows a simple method of obtaining two-stage air compression with a double piston, the full volume being used for the first stage and transferred to the annular high pressure volume through a valve in the piston. Various other methods of obtaining two or more stages of compression are used, the most usual being to couple two or more compressors of different sizes together.

RADIAL PUMPS. These are formed by using several cylinders disposed radially around a common camshaft, or, for example, a crank and slider as in Fig. 21 (*a*). Similarly, piston reciprocation can be obtained by means of a swash plate (*b*), or a *Z*-crankshaft wobble plate (*c*), in these cases several pistons being disposed in



(*a*) (*b*)

FIG. 22. ROTARY PUMPS
(*a*) = Eccentric ring. (*b*) = Crank and connecting rods.

parallel axes around the axis of the drive shaft; again various forms of valve gear can be used.

ROTARY PUMPS. These are formed by rotating the cylinders and pistons about a fixed shaft or camshaft, the motion of the pistons being obtained by an eccentric casing forming a cam-track for the pistons, Fig. 22 (*a*), or by an eccentric crankshaft (*b*).

In the first case the central shaft usually acts as a distribution shaft, being drilled with suitable suction and delivery ports, although an end face can be used instead. In the second case the outer edge of the cylinder rotor and the inner edge of the casing can form the distribution surfaces by suitable porting.

Although the eccentric crankshaft of this type of radial pump means that all main piston forces are dealt with on this component, in the case of type (*a*) the centrifugal load as well as the pressure load on the pistons, together with friction forces, come on the outer ring. Various designs have been evolved to deal with these loads, including rollers at the end of the piston, slide slippers, and a loose thrust ring carried on bearings and rotating with the pistons. It is noteworthy, however, that in the latter case the velocity of the pistons about the axis of the outer ring is not quite uniform, due to the eccentricity, and that there

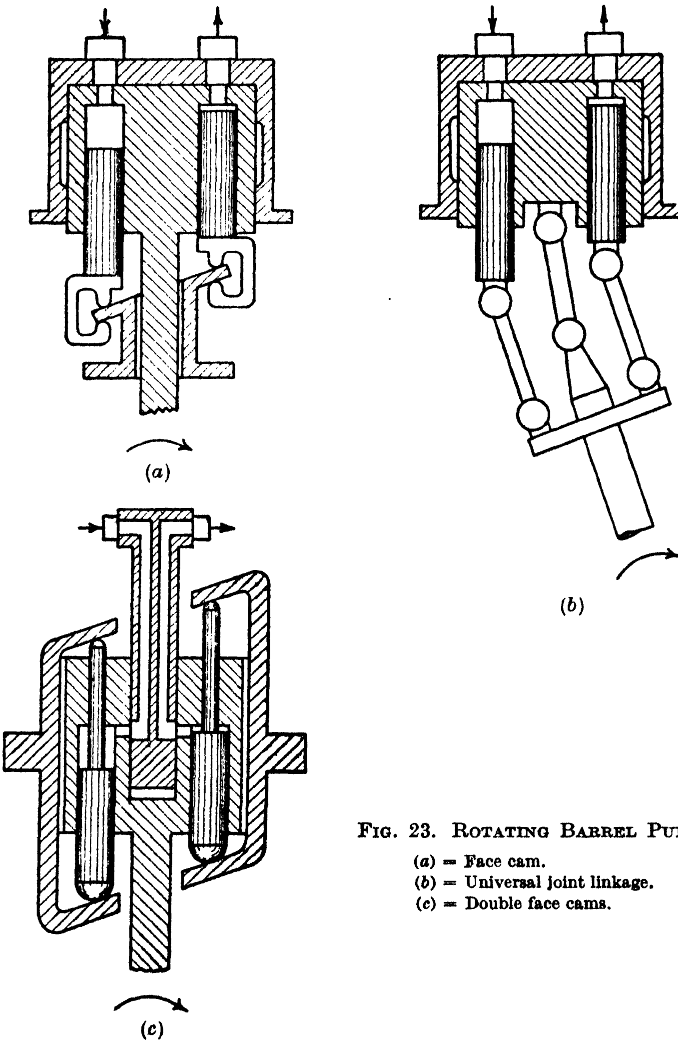


Fig. 23. Rotating Barrel Pumps

(a) = Face cam.
(b) = Universal joint linkage.
(c) = Double face cams.

is always a small amount of rubbing between the pistons and the thrust ring, which owing to the distribution of pressure loading may even tend to go round *faster* than the pump rotor.

A further variation of the rotary pump combines the centre shaft of Fig. 22 (*a*) with the connecting rods of type (*b*), but connected to the outer ends of the pistons. This is an inversion of the normal rotary type, the direction of pumping being inwards instead of outwards.

Rotating barrel pumps are shown in Fig. 23. These are characterized by the pistons working in parallel cylinders bored around the axis of the barrel, which rotates complete with pistons. Type (*a*) uses an inclined plate cam, with the pistons contacting

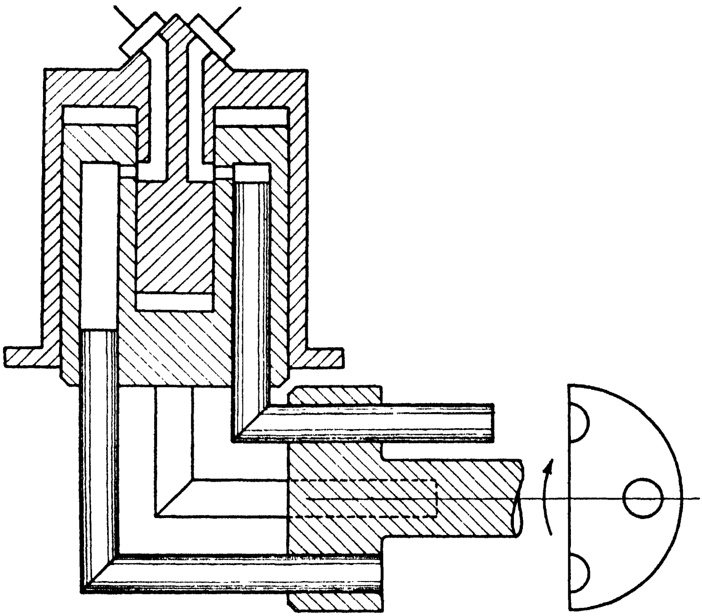
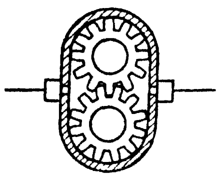


Fig. 24. The L-crank Pump

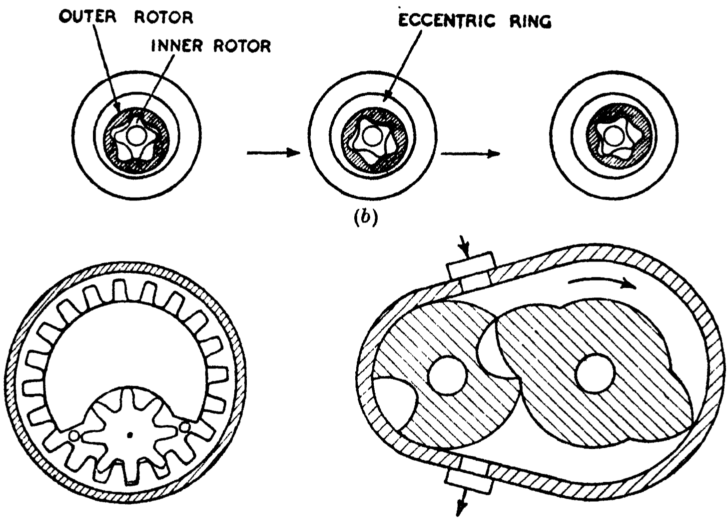
both sides, or one side, using return springs. Valving is usually by a face distributor on the end of the barrel, although a central shaft can be used (see type (*c*)). Type (*b*) obtains the reciprocating motion of the pistons by means of an inclined driving plate with ball-ended connecting rods, and a pair of universal joints to drive the barrel. Note that to obtain constant annular velocity in the barrel the usual requirement of equality of angles in the Hooke's joint components is necessary. Type (*c*) is a modification of type (*a*), using differential pistons and two cams to obtain positive action in both directions. In this case a central valve shaft is shown.

A further type is shown in Fig. 24, which uses the L-crank right-angle drive mechanism. It is perhaps more of a mechanical novelty than a practical proposition, but pumps of this type have been made. It is, strictly speaking, a special case of Fig. 23 (*b*) with the angle 90°, the drive being obtained by the rigidity of the piston stems.

3.3. Gear Pumps. A gear pump consists essentially of a pair of intermeshing toothed gears carried in a close-fitting casing, and so arranged that oil or gas is carried around by the teeth from inlet to outlet, but is prevented from returning by the intermeshing gears.


(*a*)

The best known type is that shown in Fig. 25 (*a*). For oils the tooth profiles are generally involutes cut by normal gear machinery and



(*c*) (*d*)

FIG. 25. GEAR PUMPS

(*a*) = Normal type.
(*b*) = Internal lobed rotor.
(*c*) = Internal gear.
(*d*) = Unequal meshing gears.

mesh correctly without external driving gears. The Roots blower for air or gas generally has only two teeth, sometimes three, occasionally four, and requires separate driving gears to maintain angular phrasing. Tooth profiles are cycloidal or involute.

In addition, helical gear pumps, usually herring-bone gears for oil, and simple spiral gears on the Roots blower, are sometimes used. In fact there is an excellent type of screw pump consisting of two screws intermeshing, which is obviously a particular case of the helical gear pump.

The normal Roots blower has no pre-compression of the air before delivery, and is thus noisy and inefficient. Helical Roots blowers permit pre-compression if the porting is arranged so that as the air moves along the pump, as well as round with the gears, it is compressed by the moving sealing point between the teeth and prior to the outlet port being uncovered.

High pressure gear pumps are available giving over 2 000 lb/sq in., sometimes in more than one stage to reduce leakage losses. The pressure obtainable is limited mainly by working clearance. Generally each successive stage is of slightly smaller delivery than the earlier one so that cavitation is avoided; relief valves at each stage deal with surplus fluid, by-passing it back to tank or suction. One method is to use relief valves between the stages (and not back to tank) fitted with extra loading pistons subjected to the pressure of the succeeding stage, so that the ratio of pressure between stages can be a predetermined and constant amount, depending on the areas of the seat in the relief valve and the loading piston. The last stage, of course, has a normal relief valve. This system is shown diagrammatically in Fig. 26.

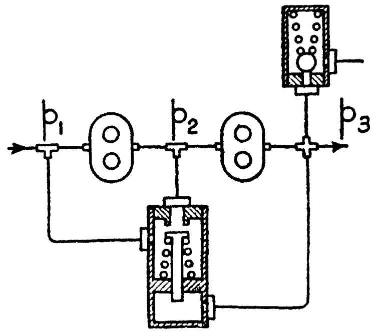


FIG. 26. MULTI-STAGE GEAR PUMP RELIEF VALVE SYSTEM

Internal gear pumps are known, both the special type (Fig. 25 (*b*)), where the rotor fills the internal gear, and the more normal type (*c*), using a crescent-shaped spacer between the gears.

It is not necessary for the tooth shape to be any well-known type provided the teeth intermesh. Many variations have been experimented with, and literally hundreds of shapes and types have been patented in the last hundred years. Fig. 25 (*d*) shows one more or less reasonable variation of the Roots blower. Another practical compressor uses a single hook-shaped tooth on each rotor. The main interest in special tooth forms is to get pre-compression, area instead of line sealing, and greater swept volume for a given bulk. The intermeshing screw pump referred to above can be considered as a type of helical Roots pump.

3.4. Vane Pumps. A vane pump is essentially of the type shown in Fig. 27 (*a*). It consists of a casing with a central rotor carrying vanes to divide up the crescent-shaped space between casing and rotor, and enable the volume between two vanes to be carried round and delivered at the outlet port. As shown in the Fig. the volume increases during the suction stroke until it

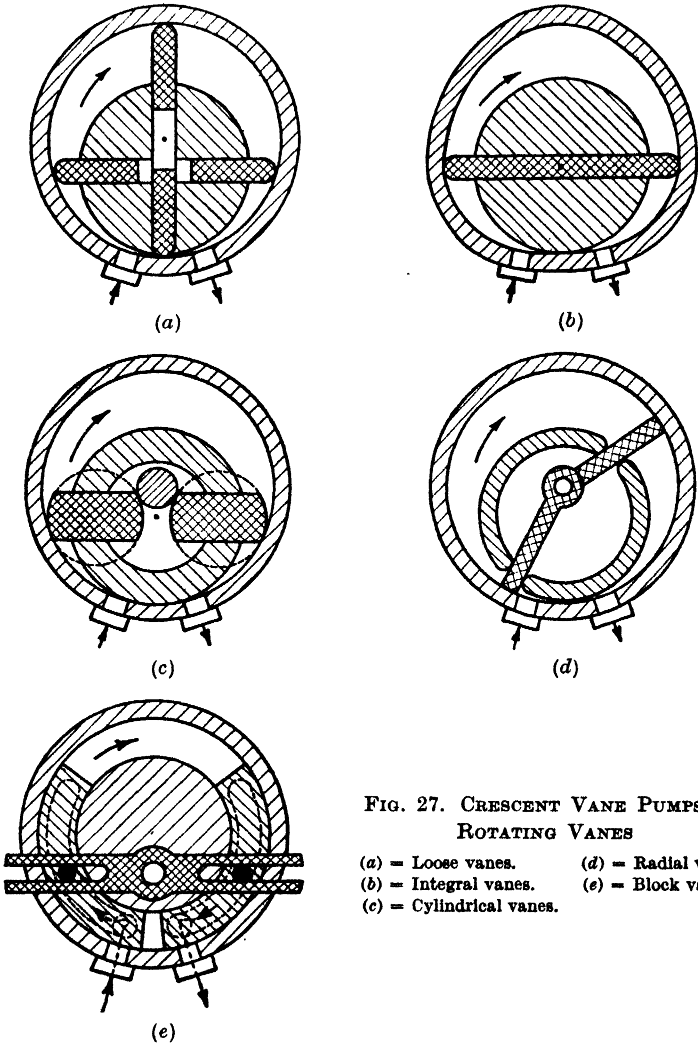


FIG. 27. CRESCENT VANE PUMPS—ROTATING VANES

(a) = Loose vanes.
(b) = Integral vanes.
(c) = Cylindrical vanes.
(d) = Radial vanes.
(e) = Block vanes.

is a maximum with two vanes equally spaced on either side of the maximum gap between casing and rotor; subsequently compression occurs until the corresponding position, at the point of minimum gap, is reached, a very small volume in fact being allowed to pass back, unless, as is sometimes done, the rotor is sunk into the casing over an arc equal to that between two vanes. With an air compressor of this type, the degree of pre-compression can be controlled by suitable positioning of the delivery port, but with an oil pump the delivery port must be enlarged circumferentially to prevent pre-compression. Similarly, there is no point in not enlarging the inlet port.

Fig. 27, type (*a*) shows loose blades, usually spring loaded for slow speed work. The thrust on the blade tips can be taken on the casing or on a rotating drum between the blades and the casing, there being only small relative slipping motion between the blades and the drum as it rotates. Type (*b*) is a variation where the blades are connected and the casing is, therefore, non-circular.* In some types the blades are guided with fixed tip clearances by circular cams on the pump casing end plates, and with flat followers on the blades and at right angles to them. In a third variation, type (*c*), the blades are sections of cylinders guided against inward motion by the axial cylinder, but still subject to centrifugal force unless spring loaded by tension springs.

Another type, existing in many variations, is shown at (*d*). In this case the vanes are mounted axially, only the rotor being eccentric. The blades pass through the rotor, usually in some sort of sliding joint. No tip problem occurs since centrifugal loads are eliminated by balancing the blades, but as they rotate with irregular angular velocity acceleration forces are set up in them.

Although most vane pumps have a circular or substantially circular casing, giving a single suction-delivery cycle per revolution, good pumps exist with approximately elliptical outer casings, giving two pressure cycles per revolution. The advantage of this is that the pressure forces on the rotor, normally on one side of the rotor only, are now in balance, thus relieving the rotor and its bearings of considerable load on a high pressure pump.

There is a further family of pumps kinematically similar to type (*d*), but using an offset slider mechanism to effect angular relative reciprocation of two or more segmental blocks, as shown in (*e*). It will be seen that as the pump rotates the space between the circular segments varies, and by suitable porting pump action takes place. An advantage of this principle is that the seals are areas of appreciable surface, and that the surfaces involved, being cylindrical, are relatively easy to produce.

Other important types of vane pump are shown in Fig. 28. Type (*a*) is the simplest type with an eccentric rotor, the cylinder

* The actual profile is known as a limaçon curve.

volume being divided by a reciprocating plate. The plate is spring loaded to maintain contact against pressure forces tending to lift it. Type (*b*) uses an eccentric "tad pole" vane mounted on the eccentric rotor, and passing through a slot in the casing. Type (*c*) uses an eccentrically moving, but not rotating, "rotor", the rotation being prevented by the central diaphragm web.

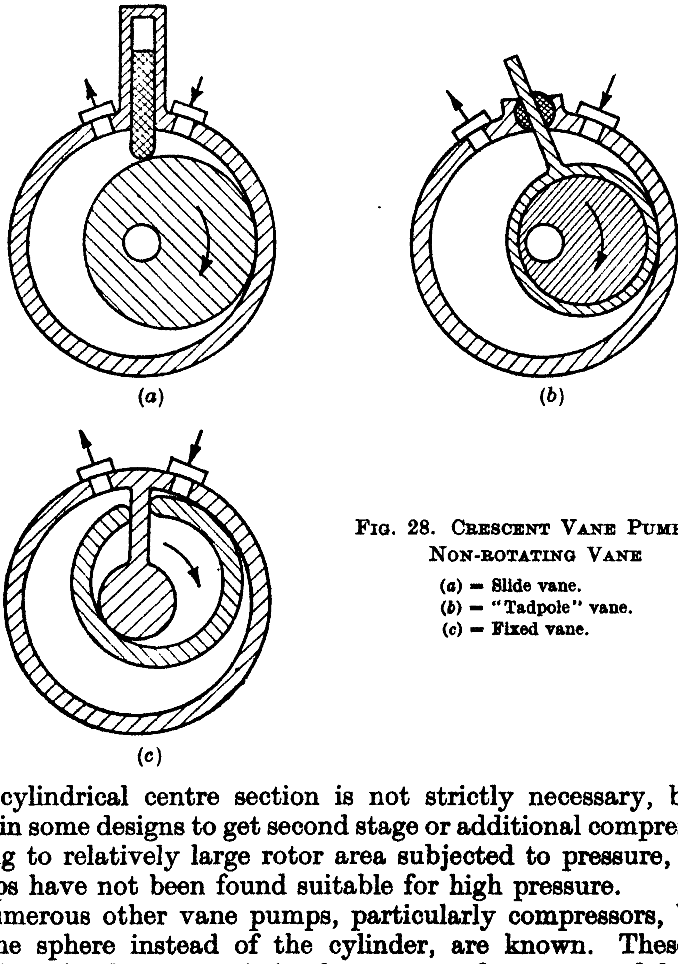


FIG. 28. CRESCENT VANE PUMPS—NON-ROTATING VANE

(*a*) = Slide vane.
(*b*) = "Tadpole" vane.
(*c*) = Fixed vane.

The cylindrical centre section is not strictly necessary, but is used in some designs to get second stage or additional compression. Owing to relatively large rotor area subjected to pressure, these pumps have not been found suitable for high pressure.

Numerous other vane pumps, particularly compressors, based on the sphere instead of the cylinder, are known. These use flapping circular or semi-circular vanes, often operated by the Hooke's joint mechanism. These have little practical value, but can be studied in textbooks on kinematics.

3.5. Variable Delivery Systems. Some hydraulic systems use pumps of variable delivery arranged so that when a desired motion has been completed, the pump can cease to deliver either automatically or manually. One well-known system, invented many years ago, uses a rotary piston pump of the types shown in Fig. 22 (*a*), the delivery being varied by moving the central distributor shaft from its point of maximum eccentricity over to the axis of the casing, thus eliminating the reciprocation of the pistons, and the pump output. This system, in conjunction with a hydraulic motor, has been used for gun turret operation on ships and aircraft.

Another method is to spring load the outer cam track with relation to the central rotor shaft, allowing the piston reactions to move the track over against spring tension so that the output of the pump is a function of spring load and thus pressure. By suitable selection of spring loads, initial tensions, etc., the pump output can remain constant up to, say, 90 per cent full pressure and then fall off progressively until 100 per cent pressure is reached. One particular pump uses a spring in the form of a Euler strut, giving a rapid collapse of pressure due to the toggle effect on the spring.

Pumps of the barrel type (Fig. 21 (*b*) or Fig. 23 (*a*) or (*c*)), can be made automatically to vary their output by allowing the cam or cams to swing against a spring into the position of zero output.

Other equivalent mechanisms could probably be devised for many other pump mechanisms to achieve the same results.

The disadvantage of the spring loaded constant pressure pumps is that at zero delivery they still absorb power due to friction, etc., and this is generally not inappreciable. The manually variable types do not suffer from this disadvantage, and in fact pump systems have been evolved using separate jacks controlled by the hydraulic system, to reduce the output, the jack being kept in position by, for example, a non-return valve, while the pumps idle at zero pressure and zero output. An equivalent system is the use of a mechanical clutch on the pump, manually or automatically operated from the hydraulic system; a system for effecting this is described in Chapter 17.

3.6. Hand Pumps. Although obviously any power pump can be operated manually, hand pumps are always of the reciprocating piston type, since an oscillating motion of a handle is easier to accomplish from the physiological point of view. Oscillating vane pumps are sometimes used for low pressures.

Experiment has shown that the optimum travel of the handle is about 12 in. when the operator is seated and about 15 in. if standing up. For continuous duty a handle load of 30 lb is a maximum, although 50 lb is not excessive for short periods.

This determines the amount of work that can be done—360 in./lb for normal seated operations, and 450 in./lb for the standing position. The output of the pump is, therefore, limited, since the work done is equal to the swept volume multiplied by the

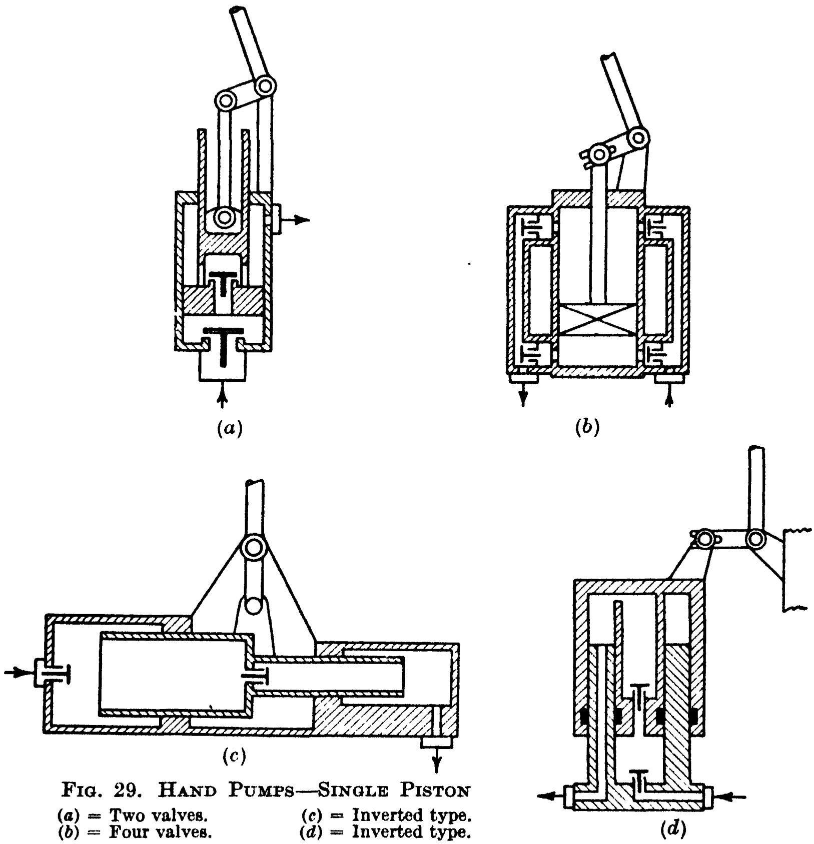
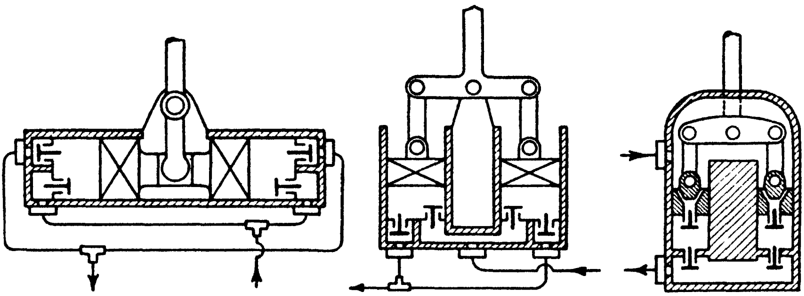


FIG. 29. HAND PUMPS—SINGLE PISTON

(*a*) = Two valves. (*b*) = Four valves. (*c*) = Inverted type. (*d*) = Inverted type.

pressure. Thus at 1 000 lb/sq in. the maximum output per single stroke should be 0·36 or 0·45 cubic inches for the two cases.

One of the oldest known pump mechanisms is that using a single piston (originally known as a "bucket" pump). Fig. 29 (*a*) illustrates this mechanism. On the up stroke of the piston the cylinder is filled with fluid; on the down stroke the suction valve closes and the transfer valve in the piston passes the fluid into the annular volume round the rod, any excess being expelled to the output connection. On the next up stroke while suction is again occurring the annular volume is discharged; thus the pump is double-acting on the pressure side, although single-acting on the suction side. The latter feature makes it more sensitive to suction losses than double-acting suction types, since suction velocity is twice that of the double-acting type. By arranging the rod area to be equal to the annular area (i.e. rod $= 1/\sqrt{2} \times$ cylinder bore), the discharge of the pump is regular. Early water pumps of this type had very small piston rods, and were thus practically single-acting. Although type (*a*) pumps will work as shown without a delivery valve in the output, a valve may be needed to prevent back pressure loading the suction



(*a*) (*b*) (*c*)

FIG. 30. HAND PUMPS—DOUBLE PISTON

(*a*) = Dumbell type. (*b*) = Linked. (*c*) = Unidirectional flow.

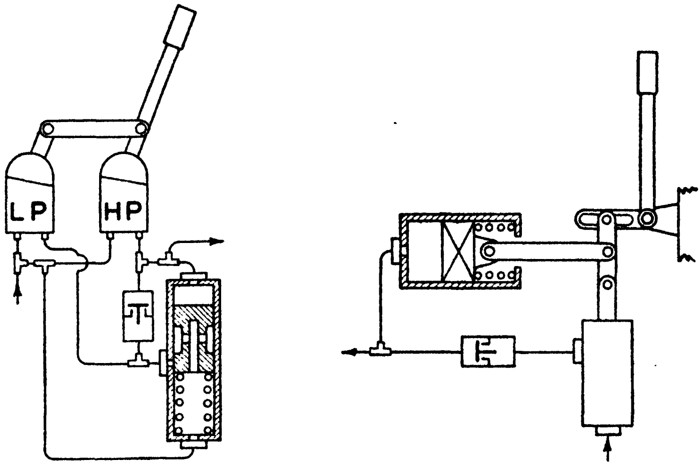
valve, or causing movement of the pump handle should either internal valve leak.

Type (*b*) shows a similar arrangement with external valves, giving alternate discharge of the full and annular volume, and will be nearly regular only if the rod is small. This pump has little advantage over type (*a*) except that the suction is more regular, being double-acting, but the pump has four valves as compared with two in the case of type (*a*).

Type (*c*) is an inversion of type (*a*), inversion being a common mechanical trick which may or may not produce advantages. The only obvious advantage of this design is that the sealing diameters are external instead of internal, and this may sometimes be an advantage. Type (*d*) is a further "synthetic" inversion. The evolution of further inversions of known pump mechanisms is an amusing mental exercise.

The other main type of pump is shown in Fig. 30. At (*a*) is a piston pump with four valves and which is double-acting on suction and delivery strokes., Type (*b*) is similar, except that the two pistons operate in parallel bores, and while the handle linkage may or may not be more complicated, the porting between suction and delivery valves is simplified. Type (*c*) is an interesting type with "uniflow" action, giving excellent lubrication of the rocker assembly.

Pumps are known which combine low and high pressure cylinders with a cut-out valve to eliminate the low pressure large volume pump when a certain pressure is reached, thus enabling the operator to do more work when the requirements of the system involve varying pressures. Hand pumps giving a fully variable delivery at constant work, irrespective of the



(*a*) (*b*)

Fig. 31. Variable Delivery Hand Pump

(*a*) = Duplex system. (*b*) = Variable stroke.

pressure, have been proposed, if not actually used. One way of doing this is to use a pump with a variable leverage, the leverage being increased by increase in pressure. Both of these pumps are shown in Fig. 31. Type (*a*) is the so-called "Duplex" system with a pressure operated by-pass valve, short-circuiting the low pressure pump at a predetermined pressure. Type (*b*) is the variable leverage mechanism. Another solution is to vary the leverage manually by, for example, rotating the handle so that any particular lever ratio can be adopted to suit the operator.

Hand pumps with in-built safety valves, and even including a fluid reservoir, are known. Another type of hand pump, particularly useful on single-acting jack systems, has a built-in discharge valve (or cock) operated by a lever on the handle; this valve short-circuits or by-passes the pump when the lever is moved so that fluid on the pressure side can by-pass back to the tank via the suction line.

There is little fundamental difference between a hand air and an oil pump, except that due regard to clearance volume and numbers of stages must be paid in the case of the compressor. The remarks already made on power pumps apply to hand pumps as well.

Foot-operated pumps for continuous duty are possible, but unusual, owing to the foot normally preferring single action. Foot-operated compressors are common, mostly spring returned; foot-operated pumps for hydraulic braking systems are well-known, but these are simple single-acting units designed to operate during a single stroke.

CHAPTER 4

RECEIVING UNITS

A RECEIVING unit can be any device which receives fluid under pressure and performs work on an external body. A jack is the most usual receiver, but motors are common enough. Sometimes a receiving unit is called a slave unit, the pump in this case being called a master or motor. In normal terminology, however, a motor is a receiving unit involving continuous rotation; a jack can be defined as a receiving unit giving rectilinear motion of a limited travel only; and a semi-motor can be considered as a receiving unit giving a limited number of turns to its output shaft only. Jacks are commonly called "cylinders" in the U.S.A.

4.1. Motors. Both hydraulic and pneumatic motors are widely used, the former on machine tools, gun turrets, and various variable speed transmissions, the latter on such applications as pneumatic drills, and special applications such as rotating the gyroscope on a gyro compass.

Hydraulic or pneumatic impulse or reaction turbines are outside the scope of this book as they are rarely, if ever, applied to pressure control systems.

Almost any pump can be made to act as a motor merely by applying pressure to the output connection. Certain reservations must, however, be made. On reciprocating pumps, positive valve timing is naturally necessary, exactly as with a steam engine. Unless a flywheel is to be used—obviously objectionable—a single cylinder will not be enough, and even with two, self-starting is not certain. Therefore three pistons are the minimum requirement for normal duty. Certain of the cam piston reciprocating mechanisms have a low efficiency in the reverse direction (e.g. the swash plate), and the excessive friction will not only cause loss of energy, but may in fact prevent self-starting. To ensure starting under adverse conditions, high mechanical efficiency is an absolute necessity.

Gear pumps make successful motors without difficulty, with the provision that plain bearings in the pump usually have sufficiently high friction to prevent self-starting, roller or needle bearings being indispensable.

Vane pumps make very successful motors, the rotor normally being carried on anti-friction bearings any way. Most commercial pneumatic motors are of the vane type, generally of the simpler loose vane type.

4.2. Jacks or Cylinders. The main essentials of actuating jacks are well known. They consist of a cylinder with a moving piston mounted on a piston rod, and arranged so that fluid under pressure moves the piston up and down the cylinder. At least two glands are necessary, one on the piston and one on the

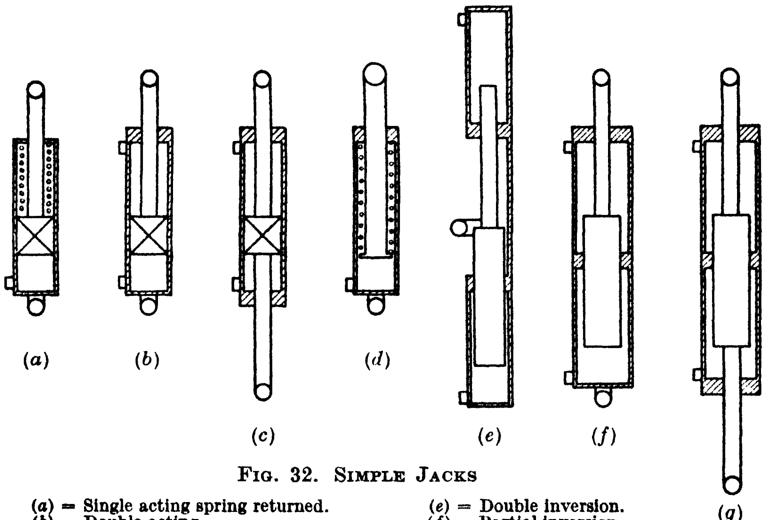


FIG. 32. SIMPLE JACKS

(a) = Single acting spring returned.
(b) = Double acting.
(c) = Through rod.
(d) = Inverted spring return ("plunger" type).
(e) = Double inversion.
(f) = Partial inversion.
(g) = Through rod inversion.

piston rod, except on a simple single-acting jack. The "dead length" of a jack is the space taken up by end fittings, piston and gland housing thicknesses, and is thus the difference between the closed length and the travel.

SIMPLE JACKS. Fig. 32 shows the principal simple types. Type (*a*) is single-acting using the full volume (with a single seal) or the annular volume (with two seals). Motion in the return direction is often by an internal spring. Type (*b*) is the normal double-acting type, the piston areas being different in each direction of motion. Type (*c*) has a through piston rod, three seals, and the same or different areas according to the diameter of the two ends of the rod. Other things being equal, type (*b*) will be longer than type (*a*) by the thickness of the piston rod gland assembly, while type (*c*) is longer than (*b*) by a similar amount (or by twice this amount, as compared with type (*a*)).

Inversions of these three basic types are shown in (*d*), (*e*) and (*f*). Type (*d*) is in fact the common "plunger" jack, the single seal being on the rod in place of the piston; the return spring, if used, can be as shown or as a tension spring inside the rod. The effective extension area is obviously the outside diameter of the rod. Type (*e*) is a double-acting inverted jack with two rod seals. This jack is of interest when, for some reason, sealing on the rod is preferred to sealing on the bore. An example of this might be a jack operating on dirty or sandy fluid, the rods being more easily hardened or chromium plated than the bores. A jack made of glass would be another example, since grinding the bore of the glass cylinders would not be easy. This jack will be seen to be identical with the hand pump of Fig. 29 (*c*).

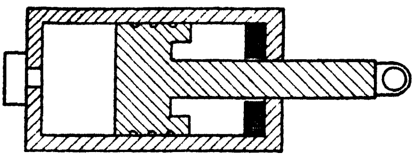


FIG. 32 (*h*). JACK WITH END SEAL

Type (*f*) is a further obvious inversion of type (*b*). Type (*g*) is a more practical inversion of the through rod type (*c*). The types (*e*), (*f*) and (*g*) are of little interest, since they are approximately once the stroke of the jack longer than normal equivalent types. The jack shown at Fig. 32 (*h*) is a single-acting one without the need for piston seals, an end seal only being used. This is a useful construction for pneumatic gear since piston ring friction is avoided by the use of a close fit only, insufficient air escaping while the jack is moving to prevent proper sealing at the end of the travel. A similar construction when the jack is required to close instead of extend is possible.

MULTI-TRAVEL JACKS. A multi-travel jack is a combination unit consisting of more than one jack combined so that the unit being operated can be moved to more than the two extreme positions of the simple type of jack. In order to accomplish the intermediate position at least three connections to the jack are required, as well as additional seals. Selectors suitable for use with these jacks are described in Chapter 5.

Fig. 33 illustrates the main types of double-acting jack giving one, or possibly two, intermediate positions of extension by dividing the total stroke accordingly.

Type (*a*) consists of two simple jacks with the cylinders mounted end to end. By means of four connections fluid can be admitted to extend one end, then both ends, giving an intermediate position corresponding to one rod extended only.

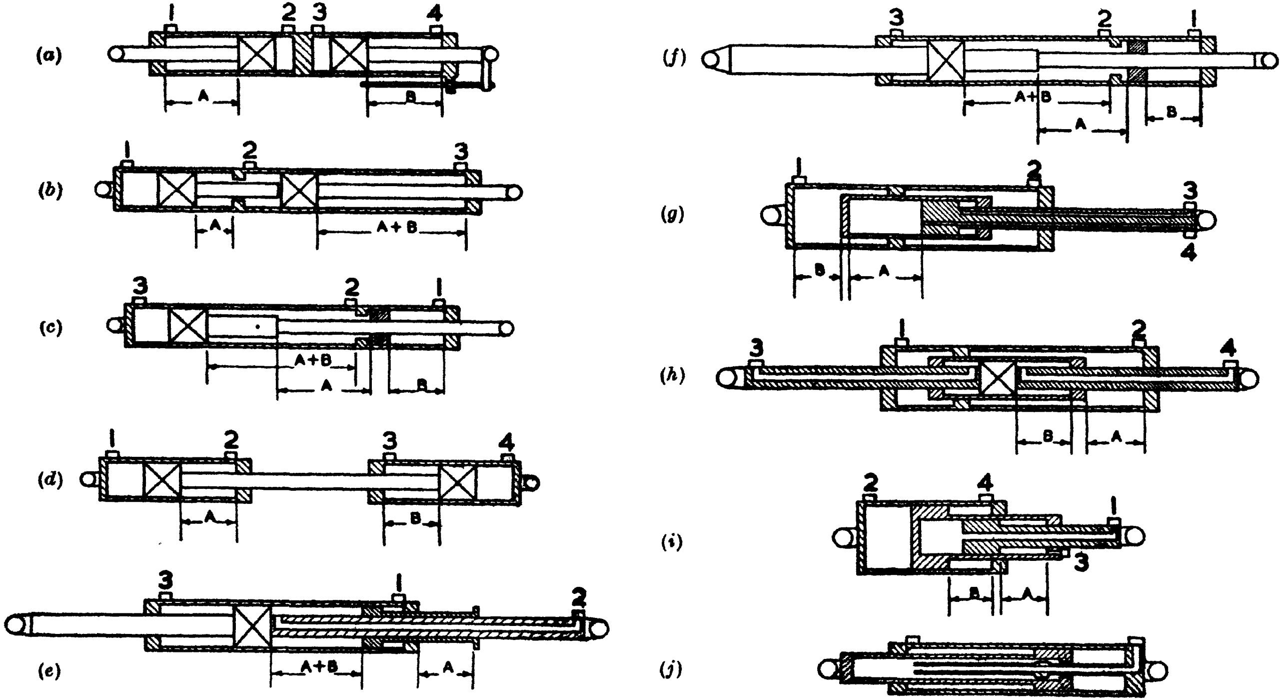


FIG. 33. MULTI-TRAVEL JACKS

(a) = Connected cylinders.
(b) = Auxiliary piston.
(c) = Auxiliary annular piston.
(d) = Connected rods.
(e) = Through rod type with auxiliary annular piston.
(f) = Through rod type with auxiliary annular piston.
(g) = Inverted.
(h) = Through rod inverted.
(i) = Telescopic.
(j) = Jack with local restriction to stop itself intermediately.

This sequence is as follows—

Pressure at: (1) and (4)
(2) and (4)
(2) and (3)
(2) and (4)
(1) and (4) again.

It is possible first to extend one rod, then retract it and extend the other one, and if the travels of the two jacks are different, this will give two intermediate or four total positions.

This sequence is as follows—

Pressure at: (1) and (4)
(2) and (4)
(1) and (3)
(2) and (3)
(1) and (3)
(2) and (4)
(1) and (4) again.

This requires more complicated selectors, and during the change from one intermediate to the other the jack may in fact extend an additional amount in the meantime.

Provision will normally have to be made to prevent the cylinder from rotating either by a guide rod, or, for example, a pair of toggle links between one rod and the cylinder. There are four seals on the jack, and, owing to the freedom of the cylinder, four flexible pipes will be needed; the basic length is the sum of the travels, $A + B +$ the dead length.

Type (*b*) consists of a normal jack with a second piston with a short extension rod forming a single-acting jack. Fluid is admitted to the secondary jack first, at connection (1), and pushes the main piston outwards by a certain amount. Fluid subsequently being applied to the intermediate connection (2) will extend the rod the whole travel. Note that there is during this second stage an empty volume to be filled between the two pistons before motion continues, and if at stage 2 pressure is relieved from connection (1) the floating piston will be pumped in again. In the reverse direction pressure must be applied at (3) and (1) simultaneously, so that as the main piston retracts it meets the stop formed by the floating piston. Finally the pressure remaining at (3) is released from (1) and the jack closes fully.

The sequence is thus—

Pressure at: (3)
(1)

Pressure at: (2) and (1), or (2) only.
(3) and (1).
(3) again.

The difficulty with this is that the sequence going out is not the same as closing and this complicates selection by a single lever considerably. To get over this the only method is to use a sequence as follows—

Pressure at: (3)
(1)
(2) or (2) and (1).
(1)
(3) again.

In this case the motion from fully extended to mid-position is not positive, the connection (2) merely being released, so that external load on the jack closes it until it meets the stop piston. This may not be objectionable on some systems.

There are three seals on this jack, but no flexible pipes are needed; the basic length is greater than type (*a*) by the stroke A, i.e. equals $2A + B +$ the dead length.

Type (*c*) is similar in some respect to type (*b*) in that the additional position is obtained by a moving stop piston, but in this case at the rod end, and thus more applicable to a jack working the other way round from type (*b*), i.e. closing instead of extending. The sequence will thus be—

Pressure at: (3)
(1)
(2), or (2) and (1).
(1)
(3) again.

This jack has four seals and needs no flexible pipes. Its basic length, however, is $2B + A +$ the dead length.

Type (*d*) is an inversion of type (*a*), the rods instead of cylinder being connected. Should an accumulator be used on the larger area (see page 53), the two pistons may be connected by a hollow rod. Connections are the same as for type (*a*), the jack having four seals, but needs only two flexible pipes; the basic length is $A + B +$ the dead lengths.

Type (*e*) is a through rod type of jack with a moving stop around the rod, connection (2) to the inner volume being through the piston rod. The sequence of operations is the same as for type (*c*), the sequence being 3, 1, 2, 1, 3, as before, with the same lack of positive motion between the fully extended and mid-positions.

As the jack extends full travel in one direction, the pressure pushes the floating piston away in the opposite direction, the annular area at connection (1) being less than at (2); therefore clearance on the piston rod for the projecting stem of the piston must be provided.

This jack has five seals, but requires only one flexible pipe; the basic length equals $2A + B +$ the dead length.

Type (*f*) is another through rod type very similar to type (*c*), but by virtue of the second rod can have the smaller piston area on the left hand side as drawn. The jack has five seals and needs no flexible pipes. Its basic length is $2B + A +$ the dead lengths.

Type (*g*) is strictly an inversion of type (*i*) described below; it consists essentially of an internal jack, both connections to which pass up the piston rod, and the cylinder of which forms the piston of another jack. The sequence of operations is similar to type (*a*) and, there being four connections, it is positive in all motions. If the cylinder wall of the inner jack is thin, the areas of the jack at both closing connections and both extension connections are practically the same. While being functionally equivalent to type (*a*), and having the same number of seals (4), it has two less flexible connections, a fixed cylinder not requiring rotational restraint, and is in fact rather shorter in length.

Type (*h*) is a similar type of jack with a through rod. It is one through rod jack mounted inside another with the cylinder acting as a piston for the outer jack. Areas are very nearly constant for all motions. Unlike type (*e*), it has positive action in all motions, being provided with four connections. It has, however, six seals and two flexible connections, but has a basic length $A + B +$ the dead lengths, and is thus the same length as type (*a*).

Type (*i*) is illustrated as a double telescopic jack, one jack inside the other, but the only one of the types so far described which can be, and is, made with several rods, giving if necessary several positions. This type of jack has been used for many years largely on account of its extreme shortness for a given stroke; the basic length is the longest of the individual travels (A or B, etc.) irrespective of the total travel, and for this reason it is widely used for lifting jacks. In these cases the full areas of all pistons are usually connected, while some times on single-acting lifting jacks, piston rod seals only are used, making the jack nothing more than a set of closely fitting telescopic tubes. The objection to using this type as a multi-position jack is that as the effective areas gets progressively smaller on the inner pistons or rods, the jack is uneconomical of space and fluid volume.

A type of jack not strictly like the other but giving substantially the same effect is shown at (*j*). This, when used with a system with an automatically centralizing open centre selector, will enable the jack to stop automatically at intermediate points by restricting the supply (or exhaust) of fluid so that the selector release is made to operate and by-pass the pump. The restriction must not be excessive or the jack will not re-start when wanted, and a hydraulic lock will be needed to lock the jack in

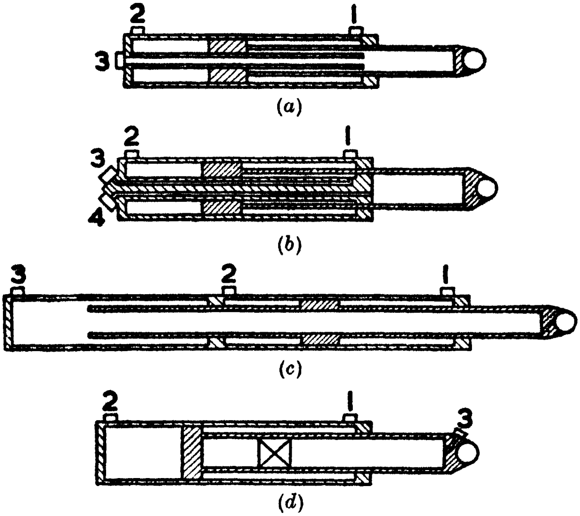


FIG. 34. MULTI-VOLUME JACKS

(*a*) = 3-volume. (*c*) = 3-volume.
(*b*) = 4-volume. (*d*) = With integral accumulator.

position. Information on the type of selector mentioned is given in Chapter 5.

MULTI-VOLUME JACKS. A multi-volume jack is one which contains for special reasons more than two volumes to which fluid is admitted. Strictly, most of the multi-travel jacks come under this heading, but in their case the intention is to obtain a division of the travel. In multi-volume jacks the travel is always simple and undivided, but the extra volumes are used to provide additional or emergency means of moving the jack, or other special features.

In Fig. 34, type (*a*) is a normal jack with an inner rod passing through a seal in the piston. This jack can be used for various purposes—

(i) To provide two independent areas for extension, one being possibly an emergency area.

(ii) To reduce the effective extension area on a jack whose main load is during closure and, therefore, requires a small pressure only to extend. In this case the inner connection alone need be used, the cylinder outer volume being unused.

(iii) To provide two-stage operation (e.g. high and low pressure; low pressure for rapid extension, high pressure for final work), using a special selector.

(iv) To provide two-speed operation, rapid extension on the smaller volume, slower extension on the larger volume.

(v) By using the inner volume as an accumulator, automatic air spring extension can be obtained.

Type (*b*) is similar, except that the inner rod carries a piston head, and thus the jack has four volumes, and is double-acting by two volumes in each direction. There is an obvious constructional limitation to the inner area of effecting closure due to the relatively small difference of diameter of the two rods. The uses of this jack are as for type (*a*), except that it can hardly be used with an integral accumulator.

Type (*c*) is a through rod type of jack with the inner volume of the rod and a clearance chamber at one end used for emergency extension or as an accumulator.

Type (*d*) is a normal jack with the piston rod inner volume used as an accumulator connected to the full volume or, as shown, to the annular volume.

JACKS WITH BUILT-IN DAMPERS. For certain purposes, especially when jacks are released and tend to move rapidly under external force, built-in damping may be useful to prevent the jack hitting its stops with excessive velocity.

Fig. 35 illustrates some types of damper. Type (*a*) is a simple plug entering a hole in the piston at the closed end of the jack. Type (*b*) is similar and has a damping piston carried in the main piston, damping again being against the closed end of the jack. Type (*c*) is applicable to the extended end of the jack, and consists of a floating piston which has fluid trapped between it and the main piston.

The above three types are applicable only to hydraulic jacks, since with a pneumatic jack insufficient air would be contained in the damping volume to be effective. Fig. 35 (*d* and *e*), however, illustrate damped pneumatic jacks with provision for an adequate volume maintained under pressure from that end of the jack under pressure. Type (*d*) is damped against extension, the air in the annular chamber formed between the inner rod and the piston rod escaping at the close fit between piston and inner rod. At this point any required degree of damping can be provided by varying the section of the inner rod. A non-return valve allows air under extension pressure to keep the damping volume at least at this pressure. Type (*e*) is similar, but damps against closure in the chamber at the end of the piston rod, the orifice being between the inner bore of the rod and the piston head of the inner damper rod.

Several of the multi-travel or multi-volume jacks can be connected to give damping in a more or less complicated fashion; fortunately, damping is rarely necessary.

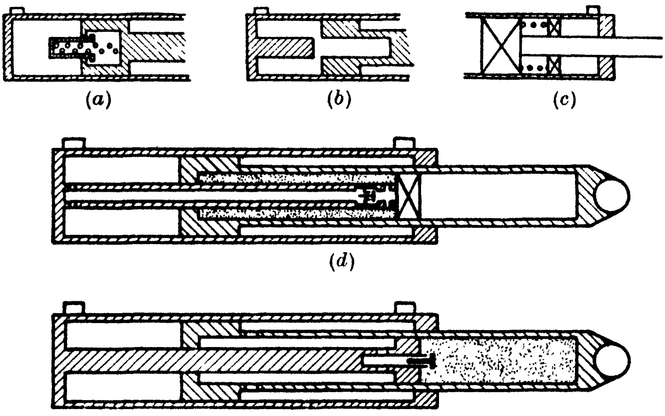


FIG. 35. JACK DAMPERS

(*a*), (*b*) = Closing direction. (*c*) = Extending direction. (*d*) = Damper system for pneumatic jack—Extending. (*e*) = Damper system for pneumatic jack—Closing.

An ingenious system of damping the motion of a pneumatic jack by injecting a cushion of air into one end to act as a damper is referred to later (see Fig. 84 (*h*)).

4.3. Semi-motors. A semi-motor is a jack arranged to give a limited number of turns to a shaft, instead of using a normal jack to move a lever over a fraction of a turn.

The most usual method of achieving this is to use a pinion and a rack on the piston rod, as shown in Fig. 36 (*a*). An interesting variation of this is to use a helical rack, using the gear thrust to engage a dog clutch to commence motion (type (*b*)). At (*c*) is shown the use of cables and pulleys to multiply the motion. Another obvious method of achieving a similar object is to use a ratchet mechanism, the jack advancing the ratchet one or more teeth with each cycle. At (*d*) is shown an example of this, using a double-acting ratchet which will rotate the output shaft intermittently in one direction as long as the jack reciprocates.

Type (*e*) is a semi-motor producing rotational motion by means of a helix or thread. By using a thread on the jack rod, this member turns, and the output shaft turns owing to the rod turning and to its own thread as well.

Diagram (*f*) is a type of rotary jack sometimes used to obtain about 300° rotation. It is not easy to seal perfectly, but satis-

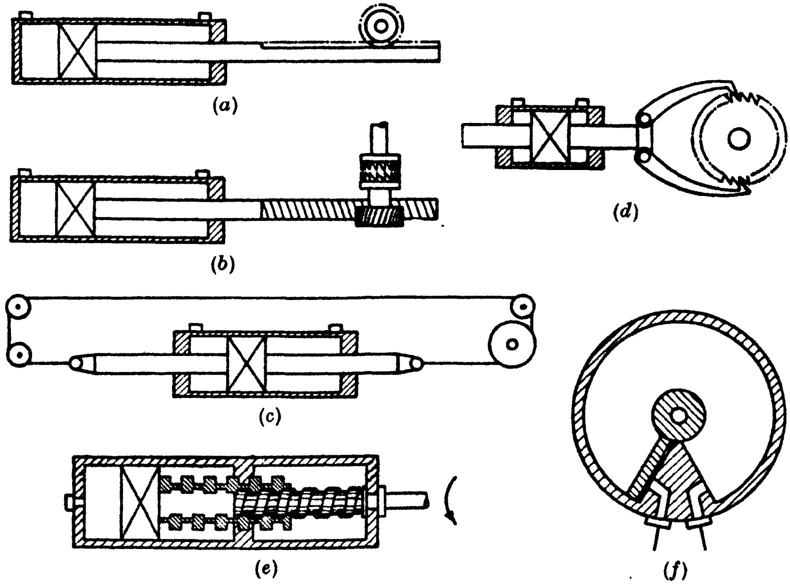


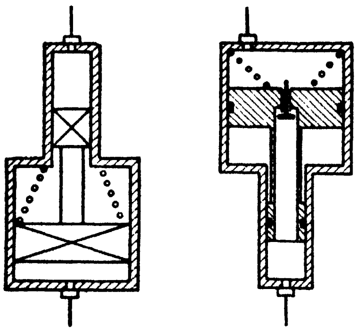
FIG. 36. SEMI-MOTORS

(*a*) = Rack and pinion.
(*b*) = Helical rack and engaging pinion.
(*c*) = Cable and pulley.
(*d*) = Two-way ratchet.
(*e*) = Double screw.
(*f*) = Rotary jack.

factory performance has been obtained; it is obviously economical as regards physical bulk.

4.4. Relays—Converters or Intensifiers. A relay is a type of jack or motor in which fluid is admitted to a cylinder with a moving piston, and the other fluid is ejected from a second cylinder, the piston of which is connected to the first piston. The usual object in doing this is to change pressure, or alternatively the fluid and often both.

Fig. 37 (*a*) shows a common type of intensifier, fluid at relatively low pressure being transformed into high pressure in the smaller chamber, in proportion to the piston areas. A common application is to obtain high hydraulic pressure from relatively low air pressure from a shop air line. Another application is to obtain oil pressure of 25 000–30 000 lb/sq in. with a normal hand-pump giving only 2 000–3 000 lb/sq in. At (*b*) is shown a system using a secondary "hydropump" to feed the output of the main unit and compensate for small losses of fluid. The second unit is provided with a feed tank and a valve opening at the end of its stroke in a manner elaborated on in "Remote Controls", Chapter 12. When the secondary relay is fully released and the valve opens, excess fluid can be expelled or any leakage made up from the tank. By operating the small unit rapidly, fluid can be pumped through to prime or bleed the main line when, for example, a unit to be tested is first connected up to it, at a much



(*a*) (*b*)

FIG. 37. RELAYS OR CONVERTERS

(*a*) = "Hydropump" or intensifier.
(*b*) = Debooster.

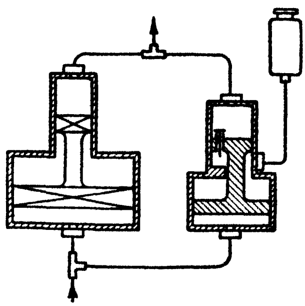


FIG. 38. INTENSIFIER SYSTEM WITH AUXILIARY BOOSTER UNIT

quicker rate than may be possible with the main unit. Depending on the physical location of the units and pipes with respect to the reservoir, a delivery valve assembly as used on car hydraulic brake units may be needed (see Chapter 12), to ensure that the secondary hydropump aspirates correctly.

Fig. 38 shows a "debooster" used on some aircraft hydraulic brake systems to reduce the pressure at the actual brake below the normal hydraulic system pressure. In this case, in order to compensate automatically for leakage in the low pressure system, should the piston reach the end of its travel, a valve is opened allowing the high pressure system to fill the low pressure circuit, at the same time applying the brakes abnormally hard; when the brakes are released everything returns to normal, any excess fluid being discharged from the low pressure circuit through the non-return valve. By suitable spring loading of the valve (see Chapter 7—"Relief Valves"), it can be arranged to cause a loss in pressure on opening, such that a safe maximum low pressure is not in fact exceeded, but this would then interfere with the free return of excess fluid or thermal expansion in the low pressure circuit.

Fig. 39 illustrates a rotary relay consisting of a motor driving a pump, giving an increase or decrease in pressure with a lesser or greater output at a comparatively high transformation efficiency. Such a mechanism, while perfectly practicable, would only be justified when the transformation involved warranted the high cost and complication of the relay.

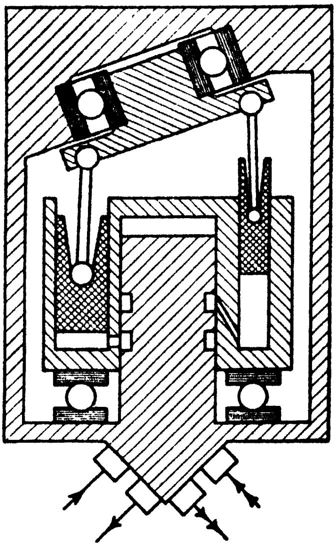


Fig. 39. Rotary or Continuous Converter

4.5. Jack Locks. Although the mechanical aspects of jack locks are outside the scope of this book, a review of the elementary mechanisms operated by fluid pressure may be of interest.

Fig. 40 shows various known mechanisms. Type (*a*) locks in any position, types (*b*), (*c*), (*d*), (*e*) and (*f*) are shown applied to locks in the extended position of the jack, but can be adapted to the other end of the jack if necessary. Type (*g*) applies only to the closed position of the jack with the rod retracted.

Type (*a*) is a rectilinear equivalent of a roller clutch. The piston head has two conical cam surfaces on which balls are trapped between the cams and the cylinder bore. Springs wedge the balls on to the cylinder. The piston head proper floats axially and carries a series of projections which contact the balls and push them down the conical cam away from the cylinder, when moved in one or other direction. Normally with the piston at rest, the piston remains central and should an external force tend to move the rod in either direction, one or other of the cams prevents motion, due to the balls being jammed on the cylinder walls. When pressure is applied to the piston, however, its initial travel pushes those balls which are preventing motion in that particular direction, away from the cam, releasing the rod. The limitation of this design is obviously the brinelling of the cylinder bore due to the ball loads, and it is only suitable for light loads.

Type (*b*) is another ball lock for one particular position of the

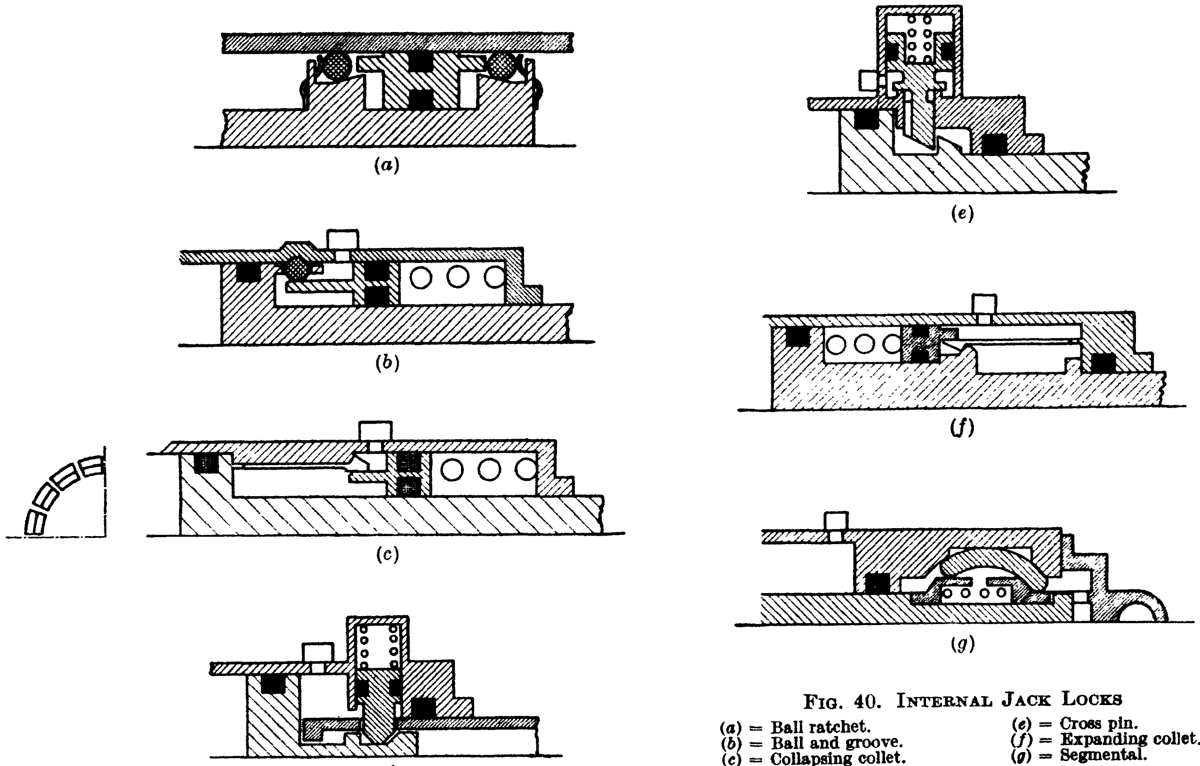


FIG. 40. INTERNAL JACK LOCKS

(a) = Ball ratchet.
(b) = Ball and groove.
(c) = Collapsing collet.
(d) = Cross pin.
(e) = Cross pin.
(f) = Expanding collet.
(g) = Segmental.

motion. In this case, instead of the balls being wedged to the cylinder walls, they fit into a groove in the cylinder. The balls fit in radial holes in the piston, and a spring-loaded lock bolt prevents their moving out of the groove. Any external compression load on the jack is thus taken through the piston to the balls, and out at a series of points to the cylinder wall. Again this limits design, but the groove can be made a separate hardened part, which is hardly possible with the cylinders of type (*a*). When fluid arrives to close the jacks, the main piston being locked will not move, but the lock bolt is first of all undone, thus allowing the balls to slide inwards, releasing the piston. During extension, the balls in their inward position, touching the cylinder lightly, eventually contact the lock bolt, and push it aside until the balls jump into the groove, whereupon the lock bolt moves home. There are several improvements to this basic mechanism, using rectangular section segments between the balls and the groove so that area instead of point contact is obtained.

Type (*c*) has a tongue member attached to the piston. The tongue is circular, but slotted to enable it to collapse slightly. It is in fact very similar in appearance to a lathe collet. The tongue has an enlarged end which abuts with a projection into, or sleeve or collar attached to, the cylinder, the area between collet and sleeve being appreciable compared with a series of balls. Compression forces on the jack rod are thus taken between tongue and sleeve, the collapse of the tongue being prevented by the spring-loaded lock bolt. When fluid arrives to close the jack, the main piston being locked will not move, but the lock bolt is first of all undone, thus allowing the tongue to collapse and pass through the sleeve. During extension the partly collapsed tongue pushes aside the bolt until it is fully inside the sleeve, when it springs out into position and the lock bolt moves home.

Type (*d*) is a cruder solution, using a transverse pin to block the rod. It will be seen that the pin fits in a hole in the rod, and can be unlocked by the initial motion of the piston.

Type (*e*) is similar except that the lock pin fits a projection on the rod, and forms an overbalanced valve. The head of the pin forms a piston which lifts the pin when fluid pressure is applied to the closure connection of the jack, at the same time lifting the non-return valve formed on the centre of the pin.

Type (*f*) is an inversion of type (*c*), the tongue being carried on the cylinder body and the lock bolt being carried by the piston. In this case the tongue expands outwards during locking as it passes over the projection on the piston rod.

Type (*g*) is strictly an external, not an internal, lock, since it is outside the fluid volume. Initial piston rod motion moves over the spring-loaded sleeve, and allows the crescent-shaped segments to rotate, freeing the jack and fitting trapped by the segments to the end housing of the lock casing. The angles of the housing and collars on the piston rod are arranged so that the crescent segments, when the collars are in the unlocked position (close together), are caused to rotate from one position to the other. An external abutment takes compression loads, the tension load passing from the end fitting, through the collar cone, and the segments to the housing.

CHAPTER 5

CONTROLLING ELEMENTS—SELECTORS

THE basic valve elements which, in combinations of various kinds, form all selectors, have already been described in Chapter 2. It was pointed out that rotary and slide valves can be considered as special cases of the "screw pair," while the other and principal element was the lift valve.

5.1. Fundamentals. Any valve element by itself can form an on-and-off cock by adding means for lifting or moving the valve, but not all will operate in both directions. A convenient symbolical representation of the cock is shown in Fig. 41 (*a*); this is defined as a *two-way selector*, since there are two possible ways for the fluid to flow. It has also two connections.

Type (*b*) is a 3-way, 3-connection selector, equivalent to an electrical single-pole change-over switch. This is the type required to operate a single-acting jack in the normal manner, although it can control a double-acting jack using the differential area system described on page 144.

Type (*c*) is a 4-way, 4-connection selector, equivalent to an electrical double-pole change-over switch. This is the type necessary to control a double-acting jack in the normal manner.

Type (*d*) is a 3-position, 6-way, 6-connection selector for operating a double travel jack of the type shown, involving 4-connections. The sequence of porting is—

(i)	Pressure	1, 2	exhaust	3, 4.
(ii)	,,	1, 4	,,	3, 2.
(iii)	,,	4, 3	,,	2, 1.

Type (*e*) is a 3-position, 5-way, 5-connection selector for operating a double travel jack of the type shown, involving 3-connections. The sequence of porting is—

(i)	Pressure	1	exhaust	2, 3.
(ii)	,,	3	,,	1, 2.
(iii)	,,	3, 2	,,	1.

Type (*f*) is a 3-position, 4-way, 6-connection (2 being in parallel with others, i.e. really a 4-connection type) selector for operating a double travel jack with accumulator or spring motion in one

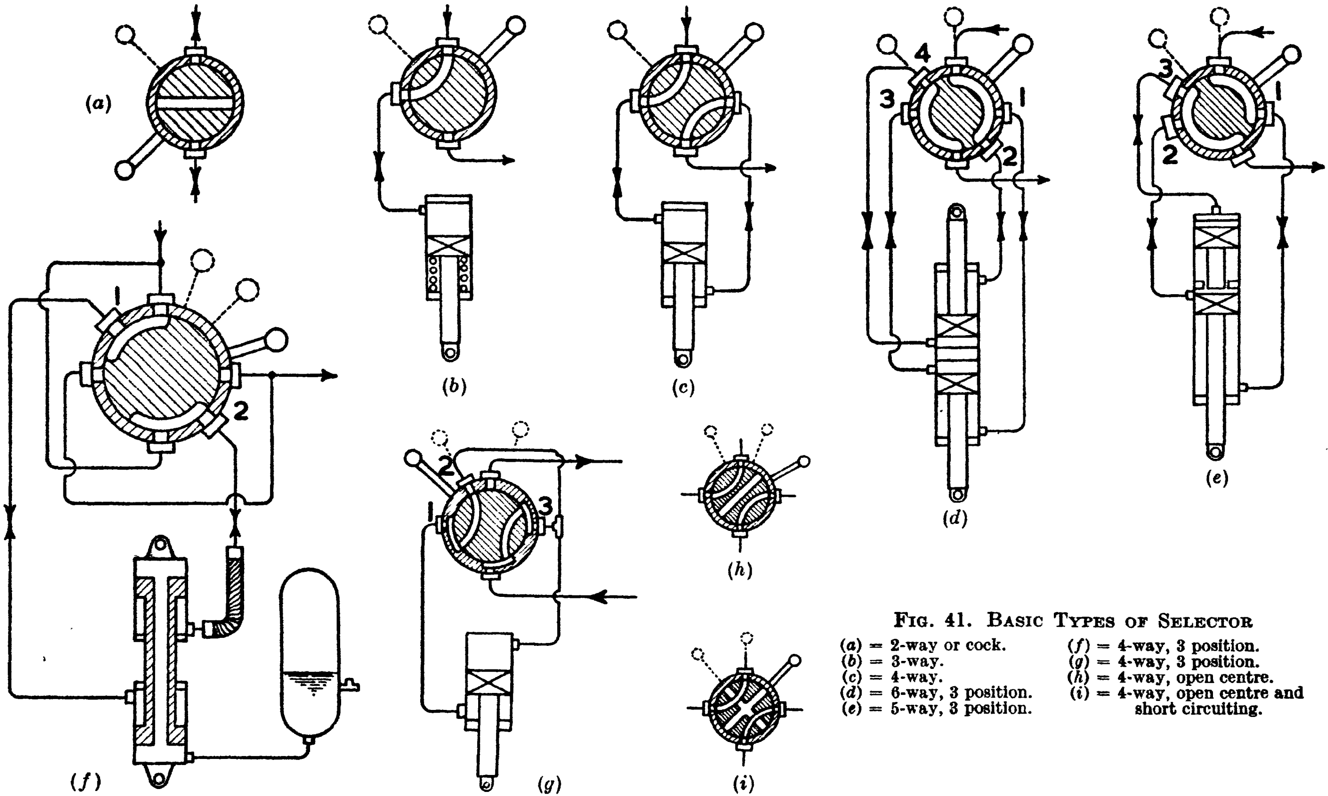


Fig. 41. Basic Types of Selector

(a) = 2-way or cock.
(b) = 3-way.
(c) = 4-way.
(d) = 6-way, 3 position.
(e) = 5-way, 3 position.
(f) = 4-way, 3 position.
(g) = 4-way, 3 position.
(h) = 4-way, open centre.
(i) = 4-way, open centre and short circuiting.

direction, i.e. a double travel single-acting jack. The sequence of porting is—

(i) Pressure	1, 2	exhaust	closed.
(ii) ,,	2	,,	1.
(iii) ,,	closed	,,	1, 2.

Type (*g*) is a 3-position, 4-way, 5 (really 4) connection selector for operating a double-acting jack with two extension speeds as described in Chapter 13.4. While the connection to the annular volume to close the jack is normal, the full volume can be connected alone or at the same time as the annular volume, corresponding to extension on the piston rod area only; this is the differential area system mentioned in Chapter 11.

The sequence of porting is—

(i) Pressure, 3 (and therefore 2 and 1); exhaust, closed; fast extension.
(ii) Pressure, 3; exhaust, 1; 2, closed; slow extension.
(iii) Pressure, 1; exhaust, 3; 2, closed.

Any of the above selectors can be provided with stop positions corresponding to all circuits blanked off, although this is most commonly done only for types (*b*) and (*c*).

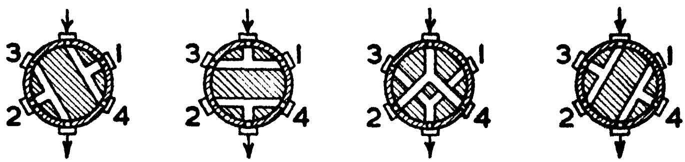


FIG. 42. PORTING ARRANGEMENTS FOR A FOUR POSITION DOUBLE TRAVEL JACK

Type (*h*) is a further common type used with "open centre" systems. In this case a normal 4-way selector (type (*c*)) is provided with a connection between pressure and exhaust when the selector is central, thus making it a 4-way, 3-position unit. Type (*b*) can be similarly adapted.

Type (*i*) is a further variation where, in the central position, all connections are connected together, thus putting the whole system under a common (and probably zero) pressure.

Fig. 42 illustrates the relatively complicated porting required to use the system mentioned on page 40, where a double travel jack is extended one way, then the other, then both. The sequence of connections required is—

(i) Pressure	1, 4	exhaust	2, 3.
(ii) ,,	1, 3	,,	2, 4.

(iii) Pressure 2, 4 exhaust 1, 3.
(iv) „ 2, 3 „ 1, 4.

This can be obtained by a true helical selector where the porting is as shown, and using axial as well as rotary motion.

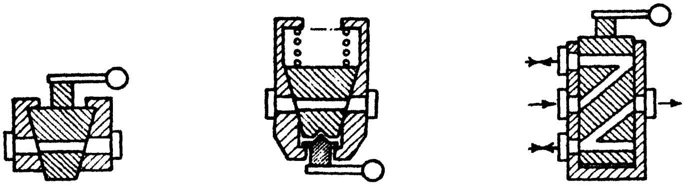


FIG. 43. ROTARY SELECTORS
(a) = Cone. (b) = Wedge lifted cone. (c) = Barrel.

5.2. Rotary Selectors. As already mentioned in Chapter 2, the general case of a rotary selector is the cone type, with the flat disc and barrel types as special cases.

The common cone plug cock is shown in Fig. 43 (a); since pressure tightness is obtained by the fine fit between the rotor and the body, the cone shape is to be preferred to flat or cylindrical types since wear is to some extent self-compensating and because the cone surfaces can be mated and bedded in together during assembly.

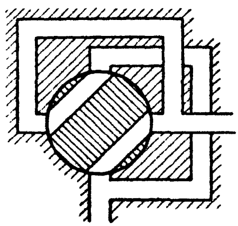


FIG. 44. METHOD OF BALANCING ROTOR LOADS ON A BARREL SELECTOR

In (b) in the same illustration is a variation of the cone cock arranged so that handle torque, acting on cam surfaces, tends to unwedge the plug during rotation, thus making turning motion easier.

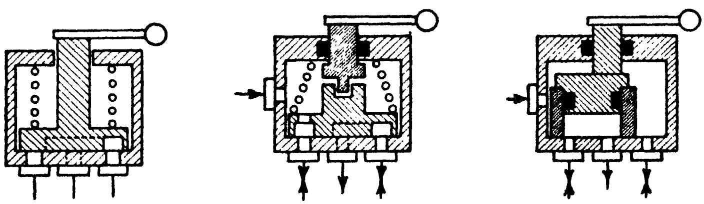
Type (c) illustrates suitable drilling for a plain barrel 4-way selector. Many other types of porting are possible, the diagrammatic types of Fig. 41 suggesting some immediately.

It will be observed that pressure applied to any port when blanked off causes a load on the rotor depending on the size of the uncovered port. Since this may give rise to excessive operating load, balanced ports of the type shown diagrammatically in Fig. 44 can be used.

The two main types of disc valve are shown in Fig. 45.

Type (a) has a flat disc rotated by means of a handle to allow arcuate grooves to cover and uncover ports. Sealing is obtained between the ports and the atmosphere by means of the perfection of the two mating surfaces. Since the disc is subjected to pressure forces tending to lift it, it must be comparatively heavily spring loaded. A further objection is that unless porting is duplicated symmetrically about the axis, the disc will tend to lift at one edge.

A preferred scheme is shown in (*b*). In this case the pressure inlet is through the body, and the handle spindle is thus sealed. The pressure now acts on the masked, or covered area of the disc and pressure loads it against leakage. The spring can be light to deal with initial contact and negative pressures. Excess back pressure in one of the ports (e.g. the exhaust port) might, however, lift the disc if the input pressure was low.



(*a*) (*b*) (*c*)

FIG. 45. ROTARY SELECTORS—DISC TYPES

(*a*) = Spring loaded. (*b*) = Pressure loaded. (*c*) = Pressure loaded.

Suitable port arrangements on the disc and body are easy to determine from the basic diagrams of Fig. 41.

Another method of obtaining the necessary disc porting is shown at (*c*), where the disc is an annular ring carried on an eccentric. It moves on an arc, which may have some advantage from the point of view of wear.

A known means of sealing the disc is shown in Fig. 46 (*a*). In this case, in place of fine finish between disc and body, the ports are sealed with bronze washers pressed on to the disc by the initial compression of a rubber ring, the fluid pressure in the port augmenting the contact pressure between washer and disc by acting on the sealing ring. Another similar construction, Fig. 46 (*b*), uses port "pistons" fitted with a diametral seal and spring loaded on to the slide or disc. The pressure of the system acting on the sealed diameter of the piston pressure loads the port surfaces to prevent leakage.

5.3. Slide Selectors. Although slide valves are usually cylindrical in section, flat valves of this type are known, and are common on steam engines for example. Sleeve valves are used on internal combustion engines, but are rare on other mechanisms. For pressure systems, generally confined to hydraulic gear, cylindrical surfaces are preferred, since they are the only ones which can be manufactured and lapped in to the requisite degree of accuracy of fit to control leakage, although the device illustrated in Fig. 46 (*b*) has successfully been applied to a high pressure hydraulic flat slide selector. Modern production methods of machine lapping and "super finishing" the plunger, and honing or lapping the bores, result in selectors which operate satisfactorily with a working clearance of about 0·0001 in., and leakage is measured at a few drops of oil per minute at 2 500 lb/sq in.

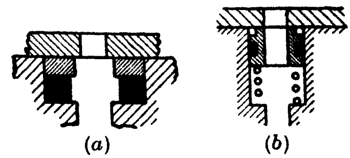


FIG. 46. SEALING ELEMENTS FOR ROTARY DISC SELECTOR

In order to avoid difficulties due to differential thermal expansion, the materials have to be chosen with similar coefficients of expansion, a high expansion steel alloy being available should the body be in aluminium alloy.

The high state of development reached in the Diesel injector pump has done much to improve slide selector technique for hydraulic applications.

In order to avoid unbalanced pressure loads on the plunger tending to increase friction, the construction shown in Fig. 47 (*a*) is not considered as satisfactory from the point of view of operating load as construction (*b*), where the annular chambers in

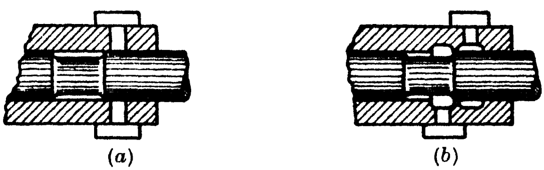


FIG. 47. PORT ARRANGEMENTS WITH SLIDE SELECTOR
(*a*) = Unbalanced. (*b*) = Balanced.

the body balance the pressure loads on the plunger. On the other hand, the leakage path between one port and another on (*a*) is much longer than between one groove and another on (*b*) and therefore the latter is relatively much more difficult to make. It is possible to achieve porting by holes drilled in the plunger, balanced in a manner similar to that shown in Fig. 44, but this is not generally considered worth while, since the increase in leakage path lengths is not sufficient to justify the extra complication of the body, and the means necessary for preventing the plunger rotating. It has been found with plungers as illustrated in Fig. 47 (*a*), that the leakage tends to take place unsymmetrically and that the pressure distribution tends to be such as to push the slide over to one wall instead of allowing it to float centrally with a very small oil film all around it. The result is that the slide tends to stick, the effect being appreciable at high pressures. The addition of small balance grooves turned, where possible, in the plunger at close pitch will assist in reducing friction, since the pressure unbalance is eliminated at each groove and the total force tending to force the slide on to one side of the bore will be much reduced. The construction of Fig. 47 (*b*) is obviously advantageous from this point of view since the unbalanced area between one port and another is necessarily small owing to the narrowness of the dividing land; the outer extremities of the slide or body should be relieved to reduce the surfaces in contact at these points also.

In the practical application of slide valves to hydraulic servos, difficulties with valve friction of this type have sometimes been overcome by rotating or oscillating the body (or a sleeve between plunger and body); this eliminates most of the "stiction".

Fig. 48 shows the main types of slide selector, although special varieties are legion and have been evolved to give the most complicated porting sequences.

Type (*a*) is the normal 3-way selector (c.f. Fig. 41 (*b*)), end leakage generally being prevented by seals. Type (*b*) is a similar unit of slightly simpler construction. Type (*c*) is the normal 4-way selector for a double-acting jack (c.f. Fig. 41 (*c*)). Type (*d*) is again another variation of the same type.

Type (*e*) is a 4-way 3-position selector of the open centre type strictly equivalent to that of Fig. 41 (*h*).

Type (*f*) is a 4-way selector of the "reaction" type so arranged that the load to operate the plunger is proportional to the pressure in the jack, etc., being controlled. This is valuable on some servo systems (see Chapter 15).

Type (*g*) is a 6-way 3-position selector exactly equivalent to that of Fig. 41 (*d*), and is used for controlling a double travel jack to one of three positions.

Type (*h*) is a 3-way unit with an enlarged central part enabling the end sealing to be poppet-valve-like.

The particular arrangement of slide selector shown in Fig. 48 (*i*), can be arranged to have valuable "overcentre" characteristics without the use of a mechanical detent; by underlapping the central spool width, i.e. allowing a slight leak either side of the inlet port slide spool when it is central, and by a similar slight overlap of the tank ports, the slide can be made to move rapidly to an extreme position when moved from the other position just

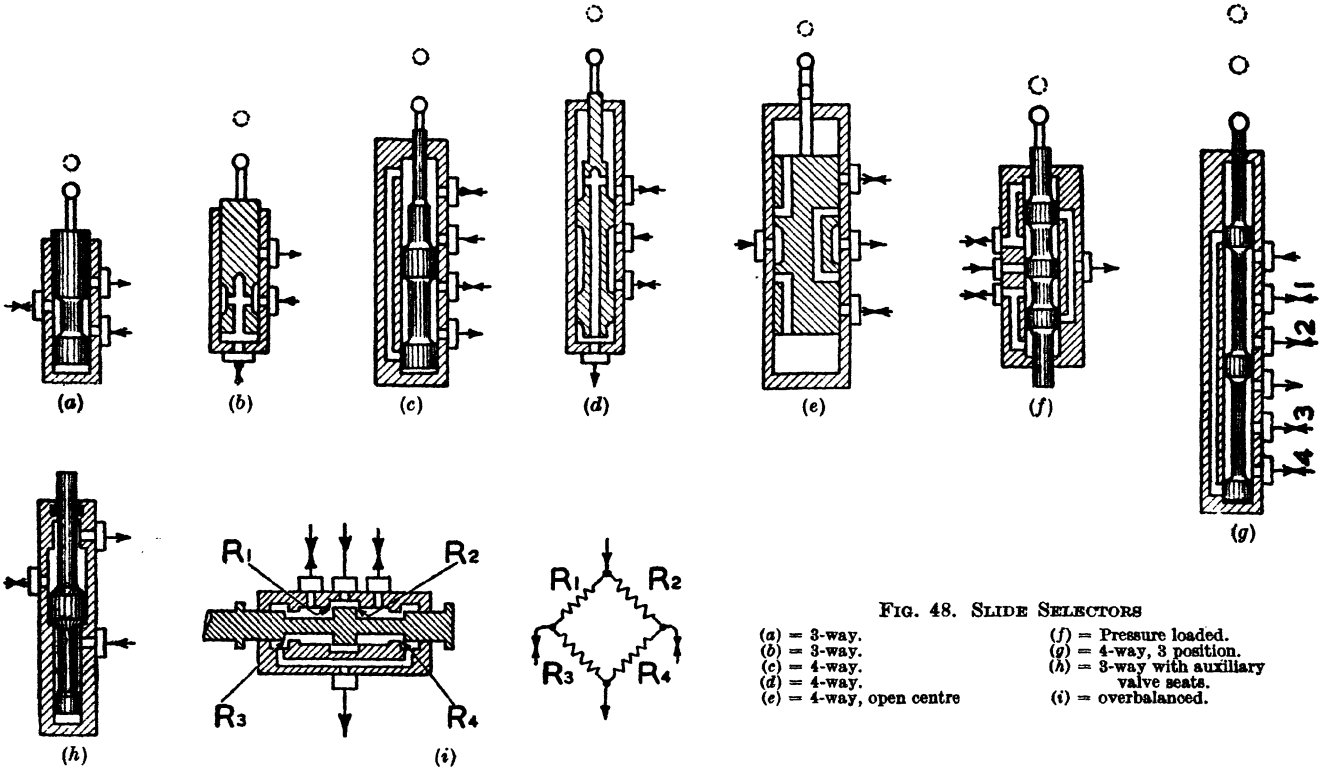


FIG. 48. SLIDE SELECTORS

(a) = 3-way.
(b) = 3-way.
(c) = 4-way.
(d) = 4-way.
(e) = 4-way, open centre
(f) = Pressure loaded.
(g) = 4-way, 3 position.
(h) = 3-way with auxiliary valve seats.
(i) = overbalanced.

past the central position which gives balanced flow and thus hydraulic balance of the piston. In practice the piston will not stay in the balanced position but will fly off centre with great accuracy of the position at which this occurs.

Owing to the cylindrical construction of the slide type of selector, use is often made of the plunger to act as a piston of a jack (c.f. the "reaction" type above) to enable the plunger to operate automatically under particular circumstances.

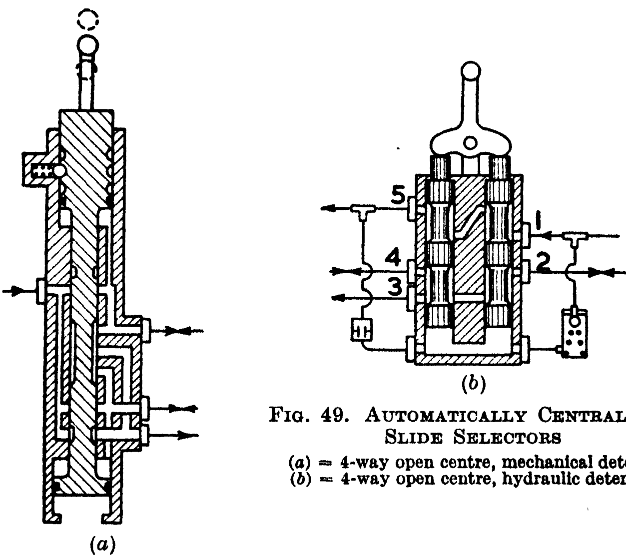


FIG. 49. AUTOMATICALLY CENTRALIZING SLIDE SELECTORS

(a) = 4-way open centre, mechanical detent.
(b) = 4-way open centre, hydraulic detent.

Perhaps the commonest example of this is the automatically centralizing selector which, when the pressure rises at the end of the travel of a jack, automatically returns to neutral, usually with an "open centre" connection for the pump. Fig. 49 (*a*) illustrates one well-known form. The plunger and body are arranged with ports giving 4-ways and 2-main positions for a normal double-acting jack system. In the central position, as drawn, the pump is "open circuited" to the tank for idling purposes; both jack connections are connected to tank as well. When the selector is moved to either extreme position, a spring-loaded detent mechanism engages and holds the plunger in this position against the reaction of the circuit pressure on a piston head connected to the plunger. When the pressure rises to a predetermined amount, as at the end of the jack travel, the detent spring load is overcome and the plunger automatically re-centres itself.

Type (*b*) is a unit performing a similar task except that it is used in a "series" system as explained later in Chapter 11. Connection 1 is from the pump, and is joined to the undersides of the plungers through a relief valve set above the maximum system operating pressure. Connections 2 and 4 are to the jack, connection 3 to tank, and connection 5 to the remainder of the selectors in the system in series and then to tank (this form of open centre system is described later as already mentioned). In the central position, as drawn, the pump idles through the connection 5 to the open circuit, both jack connections 2 and 4 being connected to tank as well through connection 3. When either plunger is depressed the open circuit is closed and the

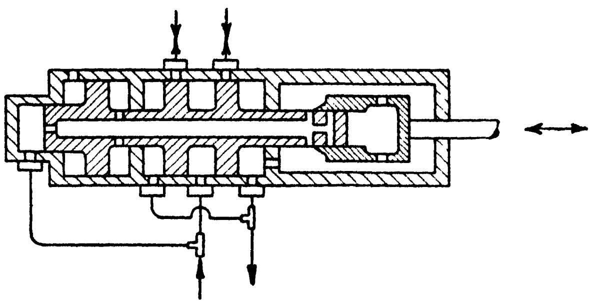


FIG. 50. PILOT-OPERATED SLIDE SELECTOR

pump connected to one jack connection in the normal manner. The plunger is kept depressed by a simple detent mechanism not shown. When the pressure rises on the completion of the jack travel, the relief valve opens and admits fluid to the undersides of the plungers, pushing the depressed one up against the detent load, and thus automatically open-circuiting the system again.

The small bleed restrictor shown, while not sufficient to prevent fluid from the relief valve operating the plungers, is adequate to allow the fluid under the plungers to leak away when the relief valve closes again.

The fundamental difference between these two types of selector is that the first has a purely mechanical detent, while the second has a hydraulic detent in the form of a relief valve. Since a relief valve can be set to open with considerably greater precision than a mechanical spring release mechanism, the reliability of the second construction will be greater than the first.

Fig. 50 shows an ingenious pilot controlled 4-way slide selector unit designed to be operated by a sensitive element such as a bellows or capsule and calling, therefore, for a frictionless and effortless pilot valve. It will be seen that the pilot valve consists of a sleeve which floats on escaping air or oil, and by moving under the action of the bellows, etc., regulates the loss of fluid, and thus the degree of balance of the slide selector proper, causing it to follow up exactly the movement of the pilot, and so effect the necessary control on the main system. The piston diameter at the left-hand end of the slide is connected always to pressure, and has a restricted orifice allowing reduced flow up the centre of the slide to the larger pilot ports; when the latter are open the slide is overbalanced to move to the right on the left-hand piston area. When the pilot ports are closed the pressure acts on the larger differential piston also and moves the slide to the left.

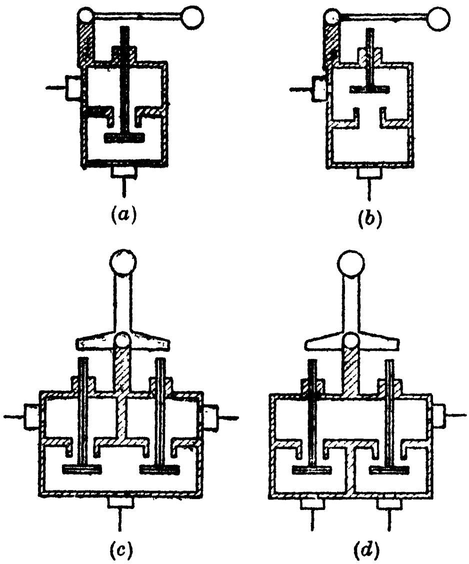


FIG. 51. POPPET VALVE SELECTORS—2-WAY
(a) = Opened. (c) = Double, common inlet.
(b) = Closed. (d) = Double, common outlet.

5.4. Poppet Valve Types.

Poppet valve selectors are formed by various mechanical combinations of the many valve elements. The types that are described in this section exist in various designs using almost all of the known poppet or ball valve elements.

Fig. 51 shows the principal 2-way selectors or cocks using poppet valves. Types (*a*) and (*b*) differ only in the action of the handle, in one case the valve being opened and in the other it being closed by a downward push on the handle. Type (*b*) may of course be lifted to open against pressure, and in this case the only difference between (*a*) and (*b*) is that whereas in (*b*) the pressure is applied continually to the valve spindle gland, in (*a*) it is not. Type (*c*) and (*d*) appear at first sight to be 3-way selectors, but this is not strictly correct. They are better considered as two cocks joined together and opened alternately by the handle rocker. In type (*c*), fluid from a single source is admitted to either of two connections; in type (*d*) fluid from either of two sources is admitted to a common connection.

Fig. 52 shows the principal 3-way poppet selectors.

Type (*a*) has 2-valves opened alternately by the rocker, to admit fluid from the pressure source to the system (e.g. the jack), or return it from the system to the tank. The non-return valve shown may be necessary to prevent back pressure in the return to the tank opening the pressure inlet valve and passing back down the pressure system. A similar non-return valve may be needed at each tank return connection to prevent back pressure opening the tank valve of an inoperative selector.

Type (*b*) is a 3-way selector, called the "shuttle" type, since the 2-valve elements are connected together rigidly and one has to be opened before the other is closed. Thus in the intermediate position all the connections are joined, i.e. "open centre".

Type (*c*) is a similar 3-way shuttle selector differing only in the disposition of the valves. Whereas inlet pressure closes the valve on type (*c*) it tends to open type (*b*).

Type (*d*) uses an "I"-shaped valve with the exhaust port formed in the hollow stem of the valve push rod. The lever action first of all closes off the exhaust valve and then opens the inlet valve.

Type (*e*) is similar to type (*d*), except that it is used with air, which can be exhausted to atmosphere instead of through an exhaust connection. Type (*f*) is similar to type (*e*), since the two valves are superimposed and become a single element.

Type (*g*) is an inversion of type (*d*), arranged to be pulled instead of pushed. The inlet valve is either held shut, or overbalanced by suitable choice of its spindle diameter, so that it remains closed automatically. Initial motion of the lever from the "valve open" position closes it, and then lifts the second valve from its seat. This valve is unusual and interesting in that if a spring is used inside the valve body tending to force the valves apart, the selector is free from play, or overlap, between opening and shutting of the valves, and is compensated for the bedding down of the valves. This may be important on a servo-jack system where the play necessary on an ordinary selector between one valve shutting and the other opening causes a degree of inaccuracy.

Type (*h*) is the same as type (*g*), except that the fluid passes through the lower valve to the jack, the return connection being the one on the side of the body.

Type (*i*) is an interesting 3-way design using a single rotary gland to seal the body in place of two glands on the valve stems of an ordinary type. The rocker opens either one or other valve, the inlet valve being spring loaded so that the rocker can open

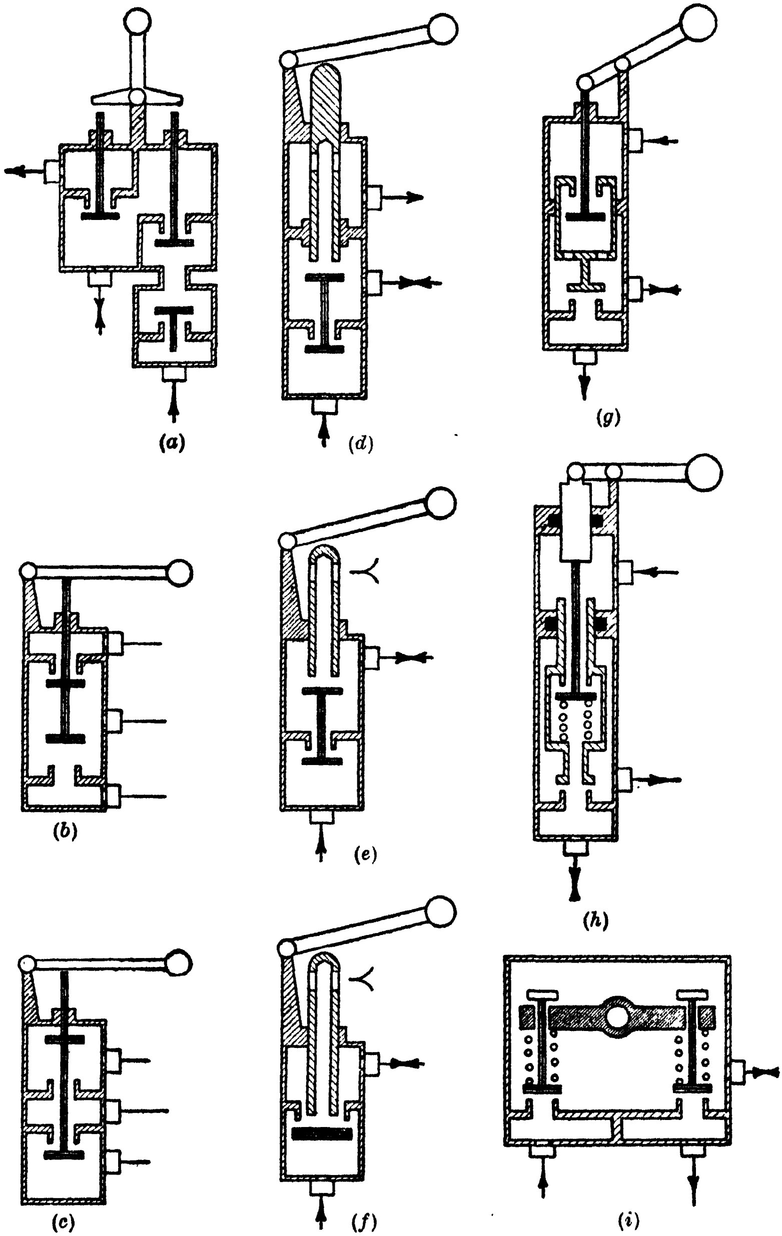
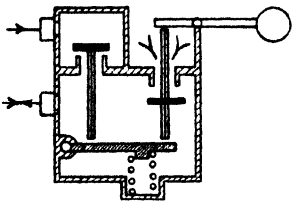
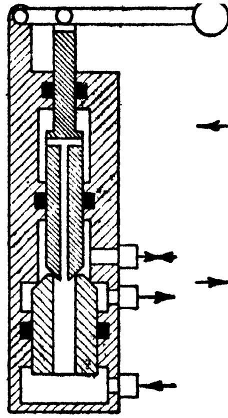


FIG. 52. POPPET VALVE SELECTORS—3-WAY

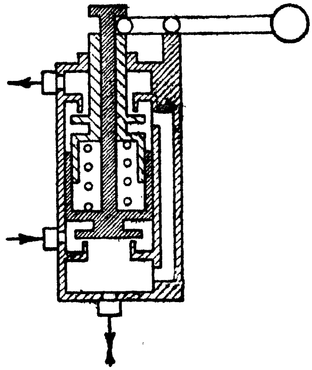
(a) = 2 valves.
(b) = Double valve.
(c) = Double valve.
(d) = I valve.
(e) = I valve, pneumatic.
(f) = Single disc.
(g) = Valves superimposed.
(h) = Valves superimposed.
(i) = Valves connected by beam.



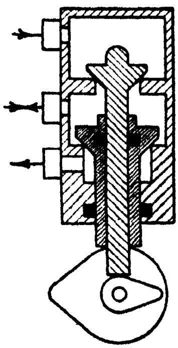
(*j*)



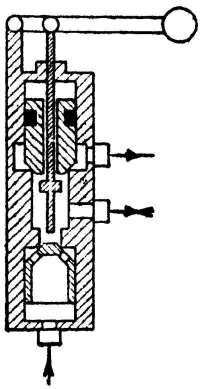
(*m*)



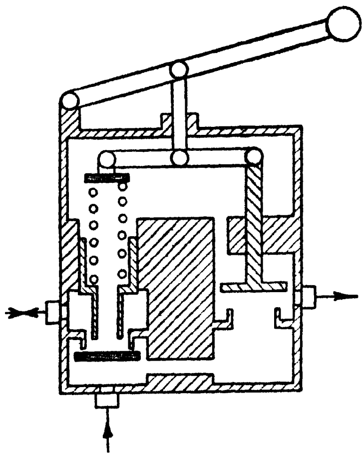
(*o*)



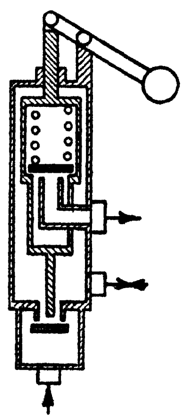
(*k*)



(*n*)



(*p*)



(*l*)

Fig. 52 (*contd.*). Poppet Valve Selectors—3-way

(*j*) = Valves operated by hinged lever.
(*k*) = Concentric valves.
(*l*) = Valves superimposed.
(*m*) = One valve seating upon the other.
(*n*) = Push-pull spindle.
(*o*) = Concentric operating stems.
(*p*) = Balance lever.

the exhaust valve leaving the inlet valve firmly pressed on to its seat.

Type (*j*) is another 3-way selector for low pressure systems, particularly pneumatic ones. This selector is interesting in that it has no moving seals, but the inlet valve is opened by a spring which limits its size and the inlet must open before the exhaust closes. Kinematically this valve is equivalent to a spring loaded shuttle selector, type (*c*).

Type (*k*) is another ingenious 3-way solution with the two valves mounted co-axially, the one inside the other. The construction is of particular value in the construction of small multi-valve selectors, although it is probably more difficult to manufacture than a normal type with two independent poppets.

Type (*l*) is similar to type (*i*) without the lever. The exhaust closes first and remains closed while the spring allows the inlet to be opened.

Type (*m*) is a "play-less" type, like type (*h*), and has one seat formed on one of the valves. Both valves are suitably balanced as will be seen.

Type (*n*) has the valves in line and the exhaust valve surrounds the operating spindle, and is balanced to seat firmly. This unit is somewhat similar to types (*d*) and (*g*).

Type (*o*) is another axial type with some similarity to type (*k*) except that the valves move in opposite instead of the same direction. A valve of this type is used on a well-known vacuum brake system where the absence of a seal on the outer valve member is probably not important.

Type (*p*) shows an interesting open centre type of 3-way poppet selector. Initial lever movement hinges a lever and closes the cock of the open centre; subsequent motion opens the selector proper. This unit will be of value in simple systems, since the open centre cock acts also as a relief valve loaded by handle pressure.

Four-way selectors usually consist of two 3-way selectors joined together, and any 3-way type could be adapted in this manner. Fig. 53 (*a*) shows the normal 4-valve 4-way selector using two linked rockers or an equivalent camshaft mechanism. Type (*b*) is an excellent radial construction. Type (*c*) is the 4-way shuttle type.

Type (*d*) is a mixture of types (*a*) and (*b*), while type (*e*) is a double I-valve type combining two 3-way I-valve selectors.

Type (*f*) is a 4-way unit with both valve heads on a common stem; obviously some means will have to be provided to ensure that both valves seat simultaneously, for example by using elastic seatings. As in other valves of the shuttle type the central position allows all valves to be open.

Five and 6-way poppet valve selectors are made from 2- and 3-way selectors combined with suitable camshaft arrangements, or equivalent mechanisms, to achieve the necessary timing sequences. It is virtually impossible to sketch these diagrammatically without perspective.

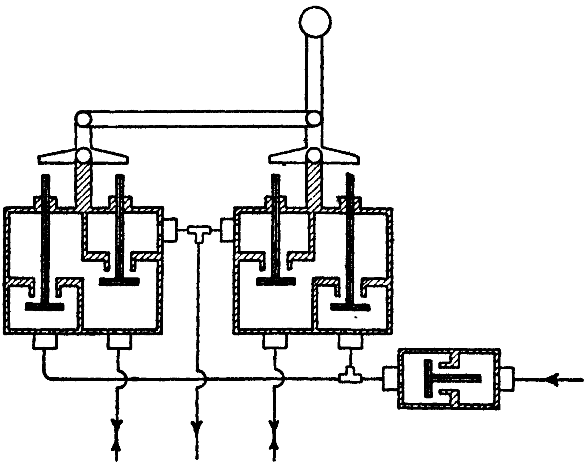
5.5. Electrically Operated Valves. Obviously any type of selector may be operated by electrical means, but there is a family of solenoid operated selectors which are of particular interest.

Fig. 54 shows this family. Type (*a*) is a poppet valve cock which is normally closed and is opened by the solenoid. Type (*b*) is a normally open cock closed by the solenoid. These two combined make up 3- and 4-way selectors. Type (*c*) consists of two type (*a*) normally closed valves, and forms a normal 3-way selector with a neutral position corresponding to both valves de-energized. Type (*d*) is a combination of one normally open and one normally closed valve, the normally open valve being at the tank return connection, so that if the current fails the jack, etc., is connected to the tank. Type (*e*) is the converse in that the pressure inlet valve is normally open, so that if the current fails the jack, etc., operates under pressure. These three valves give a choice of occurrence should the current fail.

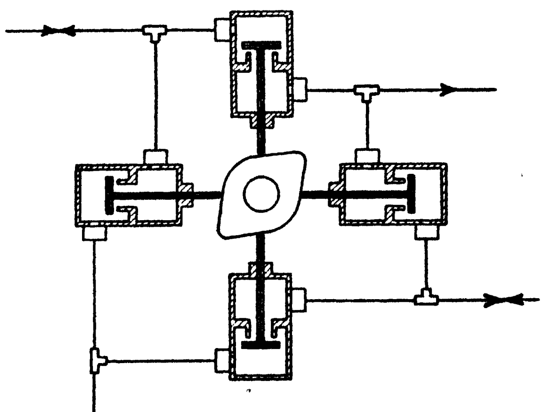
Type (*f*) is a 4-way selector formed by combining two 3-way type (*c*) selectors.

Owing to limitation in the size and pull of a solenoid, electrical selectors of these types are generally of small port size, and are often used as pilot valves to operate larger relay selectors. Sometimes a mechanical toggle device is used to improve the solenoid pull efficiency which varies considerably up to a maximum when the air gap is very small. Valves are also known where the accelerated armature inertia is used to give the valve an impact to assist in opening it, thus requiring a smaller solenoid than that dictated by the steady opening load of the valve. Often the full movement of the armature operates a switch to leave energized only a holding winding.

Fig. 54 (*g*) shows an interesting example of an electrically operated slide selector, the detail of the slide porting proper being omitted for simplicity. Normally a small bleed flow is allowed to enter the chamber at each end of the slide through a pair of orifices, shown diagrammatically at the port connections; the pressure is then dissipated in flowing through two small orifices formed by the central double valve member, and thus away to tank. A state of balance is thus achieved with the slide plunger in pressure balance, with the two orifices equal in size. When one



(a)

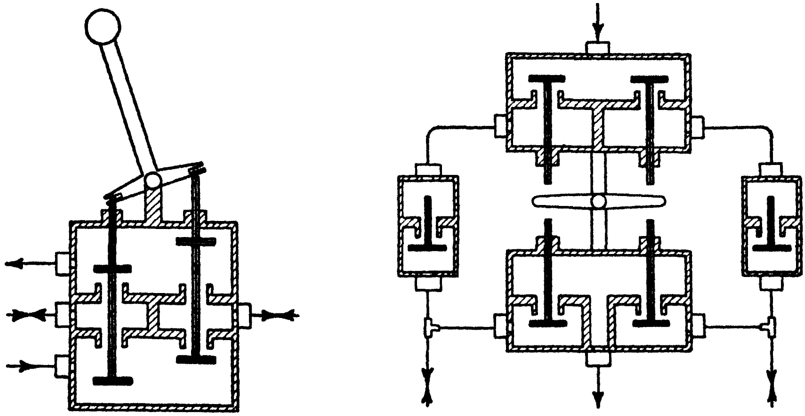


(b)

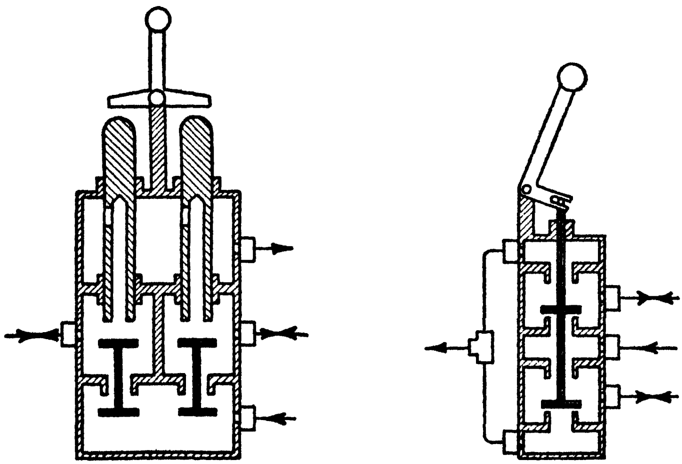
FIG. 53. POPPET VALVE SELECTORS—4-WAY
(a) = 4 valves ganged. (b) = 4 valves radially.

or other solenoid is energized, whichever valve is closed, the other opening, causes the slide to move with it so that the pressure balance is again restored. The solenoids have, therefore, the comparatively light duty of pilot valve operation, the slide moving under appreciable pressure force. Further reference to this type of system will be found on pages 113 and 180.

5.6. Special Solutions. There are certain special applications of selectors which require specific mention.



(c) (d)



(e) (f)

FIG. 53 (*contd.*). POPPET VALVE SELECTORS—4-WAY

(*c*) = 2 double valves. (*e*) = 2 I-valves.
(*d*) = 4 valves opposed. (*f*) = double I-valve.

SEQUENCE VALVES. A sequence valve is generally a simple cock or selector valve, mechanically operated by some part of the system, usually a jack, to allow another part to move after the first one has completed its motion, i.e. in sequence.

Fig. 55 illustrates the main types of sequence valve. Type (*a*) is a cock arranged to operate when the spindle is pushed, and

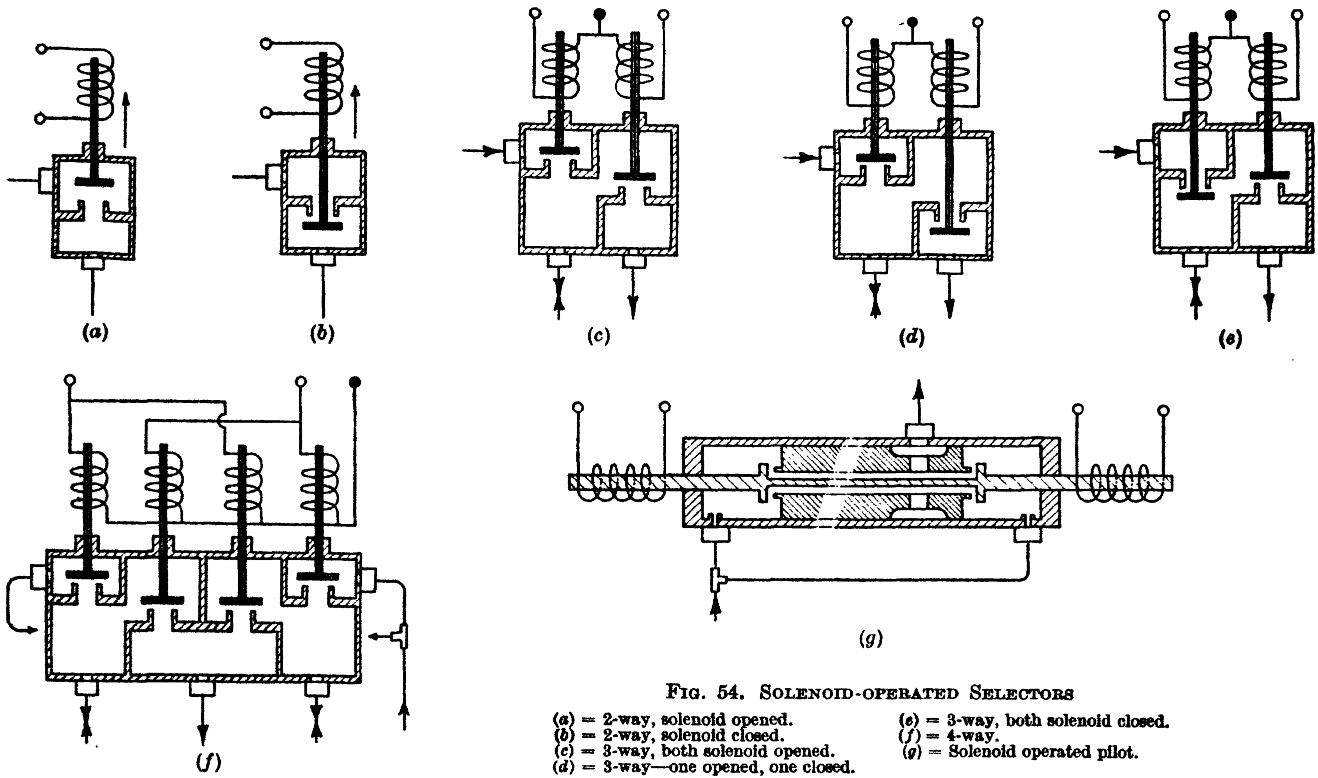


FIG. 54. SOLENOID-OPERATED SELECTORS

(a) = 2-way, solenoid opened.
(b) = 2-way, solenoid closed.
(c) = 3-way, both solenoid opened.
(d) = 3-way—one opened, one closed.
(e) = 3-way, both solenoid closed.
(f) = 4-way.
(g) = Solenoid operated pilot.

can thus be of any known type, not necessarily a poppet valve, although this is usual. Type (*b*) is a slide valve arrangement of equivalent characteristics, except that it will not permit free flow in one direction. Type (*c*) is a known variant with an additional drain connection so that the isolated connection is connected back to tank.

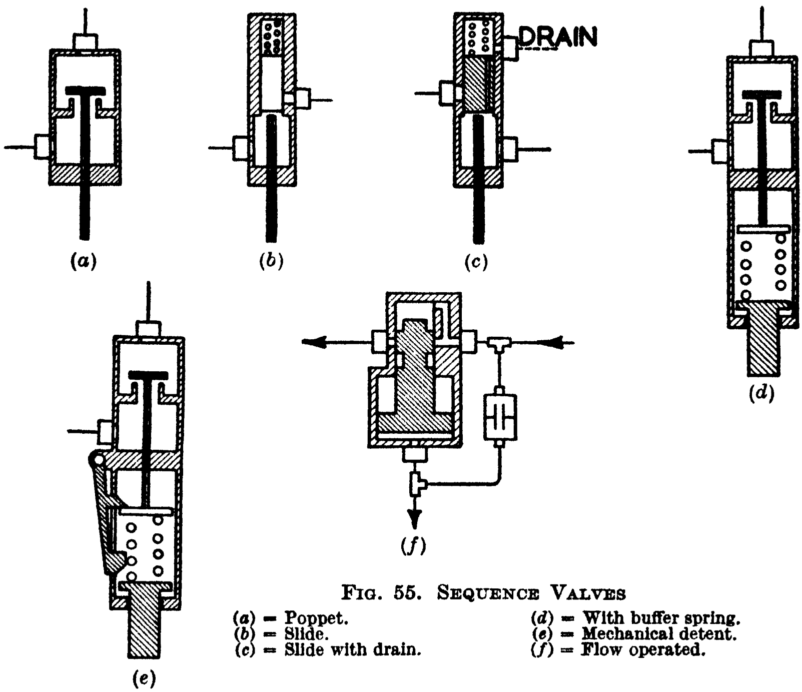


FIG. 55. SEQUENCE VALVES

(*a*) = Poppet.
(*b*) = Slide.
(*c*) = Slide with drain.
(*d*) = With buffer spring.
(*e*) = Mechanical detent.
(*f*) = Flow operated.

Since difficulties may be experienced with simple types of sequence valve, in that conditions may arise where they are only just open (see Chapter 14, page 171), some designs include means for making sure that the valve opens comparatively widely. Type (*d*) incorporates a spring which is first compressed as the spindle is pushed, until the force in the spring overcomes the back pressure on the head of the valve, allowing it to open; the spring now expands and the valve is opened widely. The opening point on this type of valve is not exactly defined, since it depends on the pressure on the valve head, but in type (*e*) a detent mechanism is used to release the spring and open the valve, the mechanism being released at a particular point in the travel of the spindle.

An interesting type of flow operated sequence valve, known as a "valve operated by stoppage of flow", is shown in Fig. 55 (*f*). Normally flow passes through a resistance to one part of the system, jack, etc. Owing to the pressure drop across the resistance the valve is kept closed. When flow stops, as when the jack in the main system reaches the end of its travel, the pressure drop no longer occurs and the valve moves automatically to allow pressure to reach the secondary system.

A useful method of mounting a sequence valve is on the piston of a jack, as shown diagrammatically in Fig. 56. This method of mounting the valve is of interest because even with the valve open the pressure in the full volume of the jack still acts on the area of the piston rod, tending to extend the jack; the jack, on the other hand, can only be single-acting.

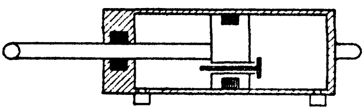


FIG. 56. MOUNTING OF A SEQUENCE VALVE IN A JACK

An automatic sequence valve system is illustrated in Fig. 119.

RELAY SELECTORS. A relay selector is one which is operated by fluid from a pilot selector, usually with the object in economizing in pipe sizes or line losses by shortening the main lines, using small lines to connect the pilot with the relay.

Obviously any type of main selector can be operated by a small jack, itself controlled by another normal type of unit. If only two positions of the main selector are required, the jack can be operated as follows—

(*a*) By a double-acting system with a 4-way pilot selector, and two connecting pipe lines.

(*b*) By a differential area jack system (see Chapter 11), with a 3-way pilot selector and two connecting pipe lines.

(*c*) By a single-acting spring return jack, a 3-way pilot selector and a single connecting pipe line, the spring, however, being large enough to operate the relay selector in one direction.

Special selector units have, however, been developed for this duty, usually with the operating jack acting directly on the relay valves.

Fig. 57 (*a*) shows a relay operated 3-way selector using a single pipe line. A 4-way selector can be obtained by joining two of these units and using two pilot pipe lines. This relay uses the basic valve mechanism illustrated in Fig. 52 (*d*), although the diaphragm between jack and tank connection is slightly differently disposed.

Fig. 57 (*b*) shows another 3-way relay selector. Use is made of the differential area system to simplify the relay. It will be seen that on moving the pilot selector, pressure acts on both ends of the selector slide, but owing to the larger area at one end, the slide moves over admitting pressure to the jack system. When the large area of the slide is opened to tank by moving the pilot selector the other way, the slide moves over blanking off the pressure port and opening the tank connection. Although the pressure slide must be a good fit, an end seal can be used to seal between jack or pump and the tank connection.

Another valve, illustrated in Fig. 57 (*c*), is similar in some respects to that of Fig. 13 (*d*) and consists of a valve normally

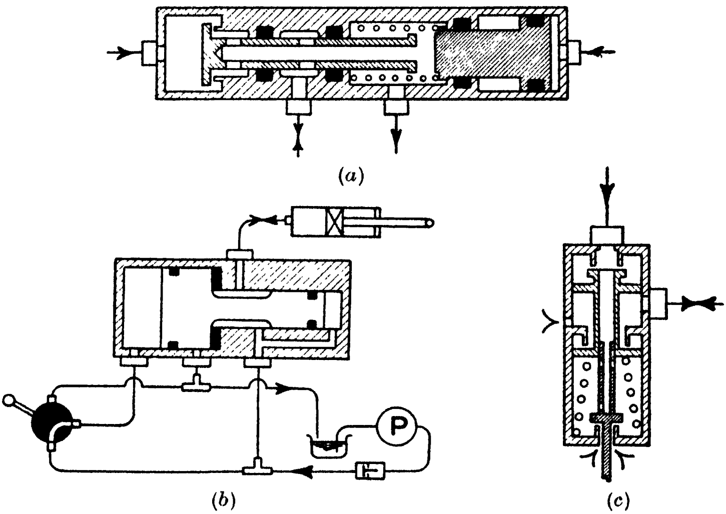


FIG. 57. RELAY OPERATED SELECTORS—3-WAY

overbalanced to "closed," but opening when a pilot valve is opened. In the position as drawn the valve is closed, the outlet being exhausted to atmosphere. When the pilot valve is opened, the valve becomes unbalanced to "open" and the upper piston diameter of the valve passes over the outlet port, admitting fluid to it; the valve remains open owing to the fit of this piston, there being a small permanent leak down the pilot valve stem and to atmosphere. When the pilot valve is closed, pressure builds up under the second piston diameter to allow the valve to be closed once more by the spring.

A type of relay selector which has three positions (e.g. A, Stop, B), corresponding to half, zero, and full pressure respectively, is shown diagrammatically in Fig. 58. The pilot selector, by acting as a simplified type of reducing valve, applies pressure to a double travel jack which operates the selector elements proper (these can be of any type); either zero, approximately half, or full pressure can be applied, the annular volume of the jack being connected to pressure continuously. By a suitable choice of areas, these pressures correspond respectively to—

Jack closed	selector shut
Small rod extended . .	selector one way
Large rod extended, small one pulled in again . . .	selector other way

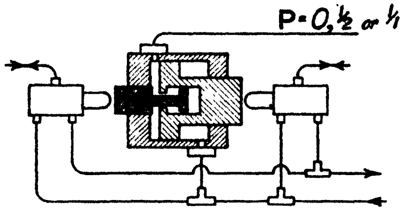


FIG. 58. RELAY SELECTOR SYSTEM (4-WAY) OPERATED BY THREE RELAY PRESSURES ONLY, IN A SINGLE PIPE

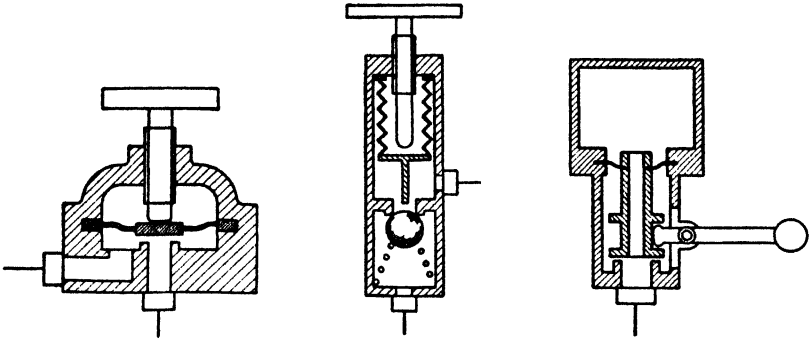
In order to obtain the required action, the areas are approximately as follows, for pressures in the ratio 0, $\frac{1}{2}p$, $1p$—

Annular area, $A_1 > A_3/2$	e.g. 6
Large piston area, $A_2 = A_1 + A_4 > A_1 + A_3$	e.g. 10
Small rod area, A_3	e.g. 3
Large piston rod area, A_4	e.g. 4

Another known method of remote control of a selector actually uses electrical solenoid operated valves, but the selector is a true relay-operated type, since the fluid from the solenoid operated pilot selector moves a pair of single-acting jacks connected to the lever of the main selector by a toggle cam device, so that once the lever is moved to one position, by an impulse of pressure, the lever remains in that position until selected the other way.

MISCELLANEOUS SELECTOR ELEMENTS. A very early design of cock, still common to-day, used a diaphragm to support the valve element, thus dispensing with a gland on the valve opening spindle. Fig. 59 (*a*) shows a typical construction diagrammatically. A flexible bellows can be used in place of the diaphragm, but in both cases the pressure which can be applied to the diaphragm or bellows is limited. In (*b*) is illustrated a bellows solution, but in this case the bellows does not guide the valve as does the diaphragm.

Fig. 59 (*c*) shows a type of release cock used on railway vacuum braking systems, so that the vacuum can be destroyed and the brakes applied (in an emergency, etc.). Normally the valve is



(*a*) (*b*) (*c*)

FIG. 59. MISCELLANEOUS SELECTOR ELEMENTS

(*a*) = Diaphragm cock. (*c*) = Quick vacuum release valve.
(*b*) = Bellows cock.

held shut by atmospheric pressure, the balancing diaphragm being rather smaller than the valve seat. When the cock is opened the vacuum in the upper chamber keeps it wide open until the vacuum is made up by air entering the chamber through the central restriction; eventually the valve will therefore close again, under the action of a light spring. This type of valve could be made to work in the reverse direction under pressure by reversing the valve seat.

CHAPTER 6

PUMP PRESSURE CONTROL VALVES

THE various systems for controlling the pressure and output of the pump in a pressure system are fully explained and illustrated in Chapter 17. The various forms of valve themselves are described in the following pages. If the principles of pump control are not understood, it is advisable to read the later section first before endeavouring to understand the workings of the valves themselves.

6.1. Automatic Cut-outs. An automatic cut-out, by-pass, or unloading valve, is a valve which, when the pressure in the system reaches a predetermined value, generally a few hundred lb/sq in. above maximum system operating pressure, by-passes the pump delivery at a reduced or negligible pressure back to tank, but maintains the system pressure at the predetermined value. The action is automatic in the sense that when the pressure in the system drops to a second predetermined pressure, the by-pass is closed and the pump re-establishes the system pressure again. Although a small pressure differential is generally desirable from the point of view of the system, a definite difference is generally necessary to obtain reliable valve performance. Nearly all cut-outs require the presence of an accumulator in the system for satisfactory operation, to prevent too frequent cutting-in due to minor leakage, temperature contractions, etc.

Fig. 60 (*a*) shows the fundamental type of automatic cut-out, basically the simplest type, but unreliable in practice. It is of the "first order" (see below). Pressure normally enters from the pump at *P*, passes through the non-return valve to the system at *S*. As the pressure in the system rises, the valve piston moves against its spring and eventually opens the valve allowing the fluid from *P* to pass back to tank at *T*, the pump thus being unloaded. The non-return valve maintains the pressure in the system. As soon as the latter drops a particular amount, the spring pushes the piston down, closing the by-pass valve and re-connecting the pump to the pressure system. The practical difficulty with a simple valve of this type is that the movement of the piston to open the valve and unload the pump is so small that the elasticity, etc., of the oil in the system allows it to move down again and close the valve. What occurs in practice in many cases, is vibration or chatter of the valve; the solution is to have a spring with a very low rate, or better still to use a special type of overbalanced valve or equivalent mechanism.

Practical designs of cut-out, which avoid the difficulty mentioned above, can be classified as follows—

(*a*) Units of the "first order," using a single valve element, the opening of this valve directly opening the by-pass, the valve overbalance being obtained by pressure action on the valve.

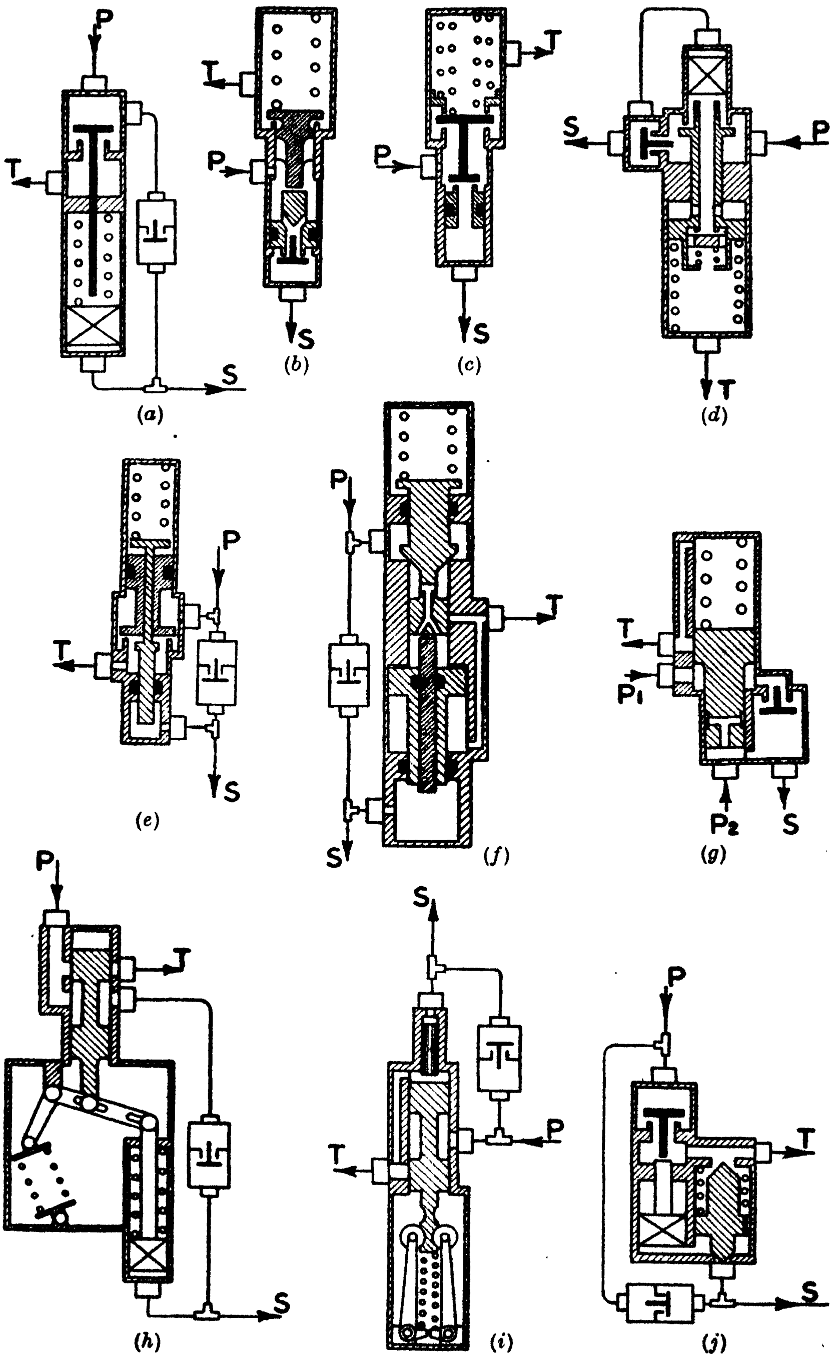
(*b*) Units similar to the above but with a mechanical device to overbalance the valve (e.g. spring and toggle).

(*c*) Units of the "second order," i.e. the indirect or relay type, using the opening of a selector element to move a piston, etc., to open the by-pass proper. These have more than one valve element.

Units of the third or even higher orders are possible, but of little practical interest. It is convenient also to classify pneumatic unloaders separately, although they operate on similar principles.

Examples of the first order or direct type are shown in Fig. 60, (*b*), (*c*), (*d*), (*e*), (*f*), (*g*). Type (*b*) has connections *P*, *S* and *T* as before, the free flow to the system being through a non-return valve in the moving piston. When the pressure from the pump reaches a predetermined value, the spring loaded valve opens. Provided the flow from the pump is appreciable, this causes a drop in pressure which, even if small, will unbalance the piston, causing it to move to open the valve even more. The piston diameter being greater than that of the valve, the valve remains open, in the desired manner, until the pressure has dropped in the ratio of valve area to piston area. Since the by-pass valve is in effect a safety or relief valve, the necessary drop in pressure at *P* to unbalance the piston is very uncertain, particularly if the rate of increase in pressure is small. In an attempt to unbalance the valve slightly, the guiding stem may be machined with masked slots (see page 16), so that the stem acts as a slide valve during initial movement; used in conjunction with a very low rated spring this may help, since the valve area proper will be somewhat larger than the stem area, and hence the slight pressure drop required will occur.

A better and more positive solution is shown at (*c*). Connections *P*, *S* and *T* are as before, the flow from pump to system being through a non-return valve formed by the piston seat. As the pressure rises the reaction on the tank valve due to the two springs is overcome by the pressure acting under the valve; the valve opens and the fluid now has to lift the secondary plate on which one spring only acts, the plate being a good fit in the


P
T
S
(a)
T
P
S
(b)
T
P
S
(c)
S
P
T
(d)
P
T
S
(e)
P
T
S
(f)
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(g)
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P
T
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(j)

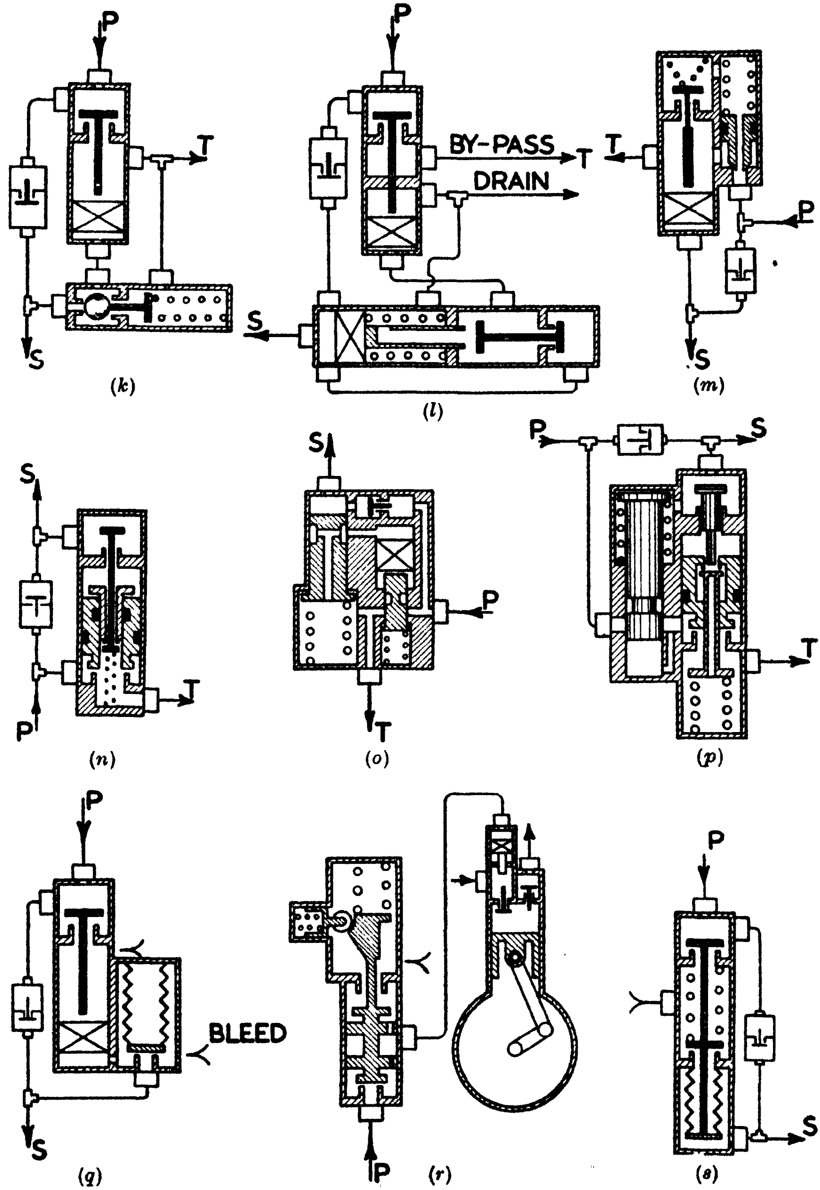


FIG. 60. AUTOMATIC CUT-OUTS

(a) = Fundamental type.
(b)–(g) = Units of the first order or direct type.
(h), (i) = Units of the first order with mechanical unbalancing mechanism.
(j)–(p) = Units of the second order or indirect type.
(q) = Pneumatic cut-out. (r) = Compressor inlet valve lifting cut-out.
(s) = Pneumatic cut-out with protective valve seat.

valve body. The pressure drop from pump connection to tank is thus in two stages, part being through each valve seat, although as the area of the plate, acting as a piston, is relatively large the intermediate pressure is relatively low, and has little, if any, back pressure effect on the opening of the main valve. Owing to the relief of the outer spring, the total force acting on the main valve is now reduced, causing a drop in pressure at the pump connection. This allows the piston acting as a non-return valve seat to close on the valve under full system pressure, and thus open the by-pass valve widely enough to unload the pump completely, except for the small loss through the secondary valve. (This is usually desirable for pump lubrication purposes.) The diameter of the piston is slightly larger than the valve area, as in the previous unit, so that the cut-in pressure (corresponding to the piston area) is less than the cut-out pressure. It should be noted that although the outer spring is relieved, the back pressure thus created reacts on the piston in opposition to the system pressure, and thus the outer spring exerts a closing load indirectly, so that the total force on the by-pass valve is more or less the same at cut-out and cut-in, the pressure differential being entirely due to the difference in diameter of valve and piston.

Type (*d*) again has the same connections *P*, *S*, and *T*, and has a built-in non-return valve. When the pressure in the system rises to the pre-determined point, the pressure acting on the small upper piston overcomes the pump pressure force closing the by-pass valve (which, it will be noted, is partially balanced) and opens it against the spring force, unloading the pump. In order to overbalance the valve and open it widely, the passage down the valve stem is blocked with a relief valve set at a low pressure. This pressure acts on the upper surface of the large piston on which the main spring abuts and thus forces it down against the spring, opening the valve an additional amount. The by-pass valve closes again when the pressure has dropped an amount approximately equivalent to the relief valve pressure.

Type (*e*) has connections as before, the non-return valve being separate. The pump by-pass valve is slightly overbalanced to remain closed; it is opened by system pressure acting on a spring loaded plunger which eventually lifts the valve. Once the pump is unloaded any back pressure in the tank line will tend to keep it open by acting on its balancing diameter, but the differential pressure drop prior to cut-in is obtained by the free travel between the two collars on the spring loaded central plunger. This valve probably works better with slight back pressure in the tank line so that the valve is forced wide open at cut-out.

Type (*f*) is similar to the previous valve in some respects, but incorporates additional means intended to open the valve widely. System pressure on the inner plunger opens the balanced by-pass valve as before. Since the exit to tank is masked by the extended end of the valve, the fluid first of all moves the large piston, forcing the annular end of it into the system volume. This is intended to cause an increase in pressure here so that the inner plunger is forced out even more, thereby opening the by-pass valve wider; the tank port is now unmasked and the pump fully unloaded except for a slight restriction in flowing through the small passages in the valve stem, which causes sufficient back pressure in the pump line to create an additional opening force on the valve; this is similar but inferior to the method used in type (*d*). The large piston meanwhile returns to its original position under the influence of system pressure.

Type (*g*) is a special unit used with a duplex pump, the main pump delivery being applied at the connection P_1, and passing to the system at *S* through the usual non-return valve. When the pressure reaches a predetermined value it acts on the exposed end of the slide plunger and moves it over against the spring pressure, opening the tank port and unloading the main pump. A secondary pump of small delivery delivers in parallel to the main pump into the system side of the unit at P_2 (or by a tee-piece at *S*). This continues to raise the pressure in the system even more until a further port in the slide plunger is uncovered and relieves excess pressure into the tank return line. The action of the secondary pump, therefore, is to open the main valve widely and to deal with slight fluid losses in the system. The pressure of the secondary pump is relieved, not unloaded, but since the output is small the comparatively large valve can deal with it reliably. When the system pressure drops, provided the rate of drop is greater than the output of the secondary pump, the slide plunger moves over and reconnects the pump. This cut-out is unusual in that no accumulator is needed.

Units of the "first order", but with a mechanical device to improve the performance, are shown in Fig. 60 (*h*) and (*i*), the valve unbalance being obtained "mechanically" instead of "hydraulically". In type (*h*), fluid enters under pressure at *P* and passes through the non-return valve to the system at *S*. Increase of pressure causes the spring loaded piston to rise and eventually to move the slide plunger (of a 3-way selector) over to the by-pass position unloading the pump. A spring loaded toggle or cam device is used to overbalance the slide in the by-pass direction to make sure that it remains widely open. A cam is preferred to a toggle as shown, as the necessary sharpness of action can more easily be obtained. As the pressure in the system drops, the spring pulls the slide down again with a slight lag at first, and then rapidly as the toggle or cam acts, reconnecting the pump to the system.

Type (*i*) is similar in some respects to type (*h*) except that a differential action is obtained by a spring loaded roller acting on the main plunger, the shape of the cam surfaces giving the necessary differential action. The detent is fitted with roller bearings to eliminate friction errors, and the plunger spring is arranged to re-act on the detent levers to avoid a second spring. The valve element is a slide plunger.

Units of the so-called "second order" are shown in Fig. 60, (*j*), (*k*), (*l*), (*m*), (*n*), (*o*), (*p*). In all these the opening of a valve at a predetermined pressure admits fluid to some sort of piston to open a second and usually larger valve; in other words, a pilot valve operates a relay, hence the use of the expression "second order". Valves of the third order are known, but seem pointless.

Type (*j*) is of the indirect type and uses the valve element of Fig. 17 (*b*). The pump delivery normally passes through the non-return valve from *P* to *S* as before. When the pressure rises to a predetermined point the valve opens against its spring. The pressure is now applied to the larger piston area of the valve opening it fully, at the same time passing to the piston which opens the unloading or by-pass valve, relieving the pump to *T* as before. The spring loaded valve has a second seat rather larger than the inlet seat, and its piston diameter has a small bleed hole through it. As the valve opens, it becomes heavily overbalanced owing to the large area of the piston, notwithstanding the small hole, and it moves rapidly over to the other seat, which, as mentioned, is larger than the inlet seat. If the travel of the valve is small, the spring force can be considered as equal to the opening pressure force on the inlet valve, and thus the net force, keeping the outlet valve on its seat once the fluid has had time to pass through the bleed hole in the piston, is the force due to the pressure on outlet seat less the spring force; or, in other words, outlet valve force less inlet valve force. When the pressure in the system drops to an amount proportional to

$$\frac{\text{outlet area} - \text{inlet area}}{\text{inlet area}}$$

the spring moves the piston valve away from the outlet seat and recloses the inlet valve; fluid in the small jack, used to open the by-pass valve, can now return to tank through the bleed hole, allowing the by-pass to close and the pump to re-establish the system pressure.

Type (*k*) is similar to type (*j*) except that a ball replaces the piston valve; the inlet seat is smaller than the outlet seat to obtain the necessary overbalance as before. A ball can, however, hardly be as effective as a shuttle piston, although the unit appears to function satisfactorily. Another similar design uses two separate balls connected by a short push rod.

Type (*l*) is another variation with a drain connection as well as the *P*, *S* and *T* connections as before. Although the unit is built in a single body it consists of two main units, a pressure selector and a "hydraulic lock" (see page 110) connected together. The fluid at *P* passes to the system through the usual non-return valve. The pressure acting on the spring-loaded piston of the selector unit eventually opens the inlet valve, and admits pressure to the small piston which lifts the by-pass valve unloading the pump. The opening pressure of the selector element is determined by the spring force, plus the pressure reaction on the valve head. The latter is relieved when the valve opens and the pressure now acts on the spindle area projecting into the chamber under the inlet valve. If this is smaller than the inlet valve, the required pressure differential effect is obtained. The opening pressure of the by-pass valve is a fixed value depending on the travel of its piston. When the pressure drops the required amount the selector inlet valve closes, and the spring moves the piston over, opening the selector exhaust valve, allowing the by-pass valve piston to drop and causing the by-pass to close. During the initial movement of the selector piston the exhaust valve is closed before the inlet opens, the selector being of the familiar type illustrated in Fig. 52 (*d*).

Type (*m*) is rather simpler. The main by-pass valve is normally balanced to close; when the system pressure rises to a predetermined value the piston lifts the small pilot valve against its spring and the pressure force on it. This allows a flow through the central passage of the main valve, which in turn unbalances it so that it opens, unloading the pump. This continues as long as the pilot valve is open and flow takes place through the main valve. (It would appear necessary to have some slight restriction in the communicating passage between the main valve chamber and the tank side of the pilot valve to maintain this flow.) The pilot valve, and thus the main valve, shut at a pressure corresponding to the spring load alone, the cut-in differential being due to the pressure load on the pilot valve.

Type (*n*) has a large valve normally balanced to remain closed. When the pressure rises to the predetermined point, it opens the small internal valve against spring force, and the loss in pressure down the small clearance between the valve stem and its hole causes the pilot valve to reseat on its second valve face. Since the upper surface of the main valve is now open to tank, it becomes unbalanced and opens. As the pressure drops, the small inner valve closes, the by-pass valve once more becoming balanced to close. The pressure differential between cut-out and cut-in is proportional to the relative areas of the two inner valve seats.

Type (*o*) uses a spring loaded slide type of pilot valve, the opening of which admits fluid to a piston which in turn opens a second slide forming the main by-pass valve. The pilot slide valve is really a three-way selector, since when the pressure drops it allows the small cylinder to be exhausted and the by-pass to close. The pressure differential is thus a function of the slide travel, the spring rate, and the overlap between the slide opening and exhausting.

Type (*p*) is somewhat similar to type (*n*), except that an additional slide valve is used; it might in fact be considered as of the "third order", since opening involves three distinct steps. The action is as in the earlier type, except that the auxiliary spring loaded slide only uncovers the port to the by-pass valve above a certain pressure, presumably to protect or isolate it during normal operation, and thus the action of the slide plunger is auxiliary to, and not necessary for, the functioning of the main by-pass valve; this unit is therefore strictly of the second order.

Pneumatic cut-outs or unloaders do not differ in principle from hydraulic types, and many of the valves described could work equally well with gases or with liquids.

Fig. 60 (*q*) is a typical pneumatic "unloader" for a compressor. By using a small bleed hole giving a permanent leak, the volume under the piston and around the spring-loaded bellows can be exhausted to allow the valve to seat properly when cutting in again. The bleed is so small that the loss of air from the system is negligible, except when the compressor is stopped, when presumably the valve is isolated.

Another known system uses a cut-out to open the inlet valve of the compressor instead of a separate by-pass valve. This is shown in Fig. 60 (*r*). The valve device of this unit is the same as type (*j*), but has a mechanical cam unbalancer as well.

A further mechanism uses an automatic cut-out device to restrict the compressor suction instead of unloading its output, i.e. the valve is closed by the lifting of the cut-out piston or bellows instead of being opened.

An ingenious means for providing valve overbalance which is known is to allow a magnet to repel and attract the unloading valve member in an equivalent manner to a spring toggle. An actual construction uses a hinged lever connected to the valve and swinging between two permanent magnets, the effect being that of a powerful spring toggle.

An interesting device applicable particularly to pneumatic cut-outs is shown in Fig. 60 (*s*). Should the bellows fail the whole system would normally drop to atmospheric pressure. By providing a second valve and seat as shown, should the bellows leak, this second valve acts as a relief valve to limit the system pressure, but prevents the gas from leaking away. By appropriate choice of valve area, the emergency relief valve pressure can easily be set a little higher than the normal cut-out pressure.

6.2. Automatic Clutch Control Valves. Pumps provided with mechanical clutches can be operated automatically to give the same characteristics as an automatic cut-out or a variable delivery pump, by disengaging the pump at a predetermined pressure and re-engaging it again as the pressure drops. Since no clutch will work adequately when disengaged a small amount only, the same problem occurs as with the cut-out, whose valve must be opened widely. Some system must, therefore, be devised to move the clutch by a predetermined amount automatically with a positive or snap action.

The pump clutch will be one of three types—

(i) Spring engaged and pressure disengaged.
(ii) Pressure engaged and spring disengaged.
(iii) Spring engaged and disengaged through an over-centre toggle.

Fig. 61 illustrates two types of known valve for disengaging the clutch against spring tension. In (*a*) is a type using the same valve element as the cut-out illustrated in Fig. 60 (*j*). The unit is, in fact, identical except that the piston of the small jack operates the clutch disengagement instead of opening a by-pass valve. The operation of this unit will be understood from the earlier description on page 82. The cut-out illustrated in Fig. 60 (*l*) could probably be adapted for clutch operation in a similar fashion.

Type (*b*) in Fig. 61 consists of 3-way selector unit connected to the clutch jack. As the pressure rises it eventually overcomes the reaction of the pressure of the small accumulator on one side of the valve piston, and moves it over, closing the tank valve and opening the pressure valve, operating the small clutch jack. As the pressure drops the accumulator pressure eventually moves the valve piston in the opposite direction, closing the pump valve, opening the tank valve and allowing the pump to re-engage. If it were not for the very low rate of the accumulator this system would have all the faults of the simple type of cut-out, but in practice it works satisfactorily with the accumulator.

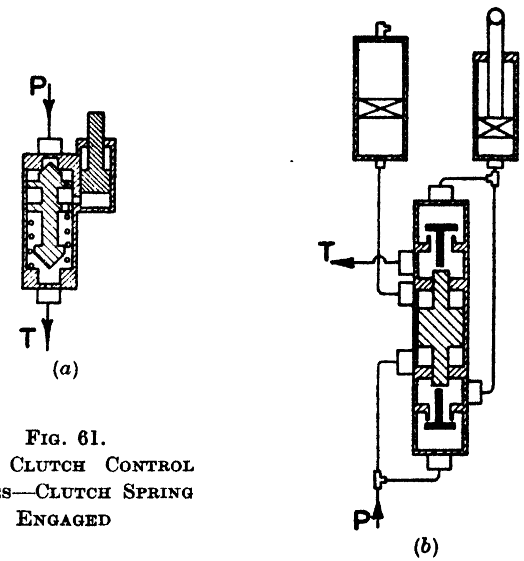
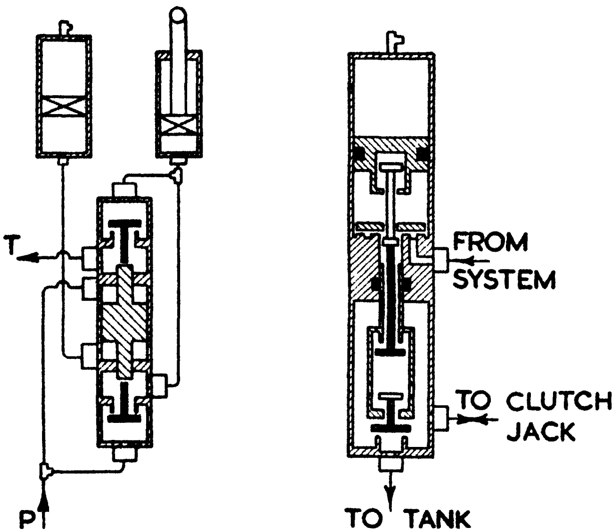


FIG. 61.
PUMP CLUTCH CONTROL VALVES—CLUTCH SPRING ENGAGED

Fig. 62 (*a*) illustrates the same valve as in Fig. 61 (*b*), connected differently to enable pressure to engage the clutch. It is supposed that some means (e.g. a further accumulator) is used to maintain the system under the initial pressure required to start the pump in the first place. Pressure from the small accumulator associated with the control valve normally keeps the inlet valve open, connecting the system pressure to the pump engagement jack. When the system pressure rises sufficiently it overcomes the accumulator pressure and moves the valve piston over, closing the pressure inlet valve, opening the tank valve, and allowing the pump clutch jack to close, disengaging the clutch. As the pressure drops the reverse occurs.

In (*b*) is illustrated another type of valve mounted inside the clutch-operating or system accumulator (not to be confused with the accumulator of the previous valve, which is merely a substitute for a spring). The valve is strictly a 3-way selector of the type illustrated in Fig. 52 (g) connected to the piston of an accumulator. The action is as follows: it is supposed that the system is under sufficient initial presssure to start the pump delivering. Fluid enters the accumulator through a plate (or any other) type of non-return valve, and lifts the piston. The pressure also opens the inner I-valve, seats the tank valve and reaches the clutch jack. As the piston rises it reaches a point (and thus a pressure) where the slackness of the inlet valve element is taken up and



(a) (b)

FIG. 62. PUMP CLUTCH CONTROL VALVES—CLUTCH PRESSURE ENGAGED

the inlet valve is pulled closed, the pressure forcing the moveable inlet valve seat on to the valve proper. As soon as the piston has risen a little more, it opens the tank valve, emptying the clutch jack and stopping the pump. As the system pressure drops the tank valve is eventually closed, then after a further drop the float of the inlet valve seat is absorbed, it strikes its abutment on the tank valve and the inlet valve opens, again re-engaging the pump. The main criticism of this type of valve, and in fact any automatic valve for pump control whose setting depends on accumulator pressure, is that the setting will vary as the accumulator leaks (although this may be slight) and with atmospheric temperature change. A temperature variation of 30° C. is not unlikely from day to day, and since the pressure varies as the absolute temperature, this would cause a 10 per cent variation in cut-out pressure.

Pumps engaged and disengaged by the action of a spring toggle are known, but rare. All that the automatic valve must do in this case is to move the toggle from one position over centre, where it will automatically remain until the valve moves it back again. The basic mechanism similar to that of the cut-out in Fig. 60 (*h*) is shown clearly in Fig. 63. Free play between the piston and toggle lever can be arranged to give the necessary snap action, although in company with all toggle mechanisms of this type the disengaging and re-engagement points will be ill-defined.

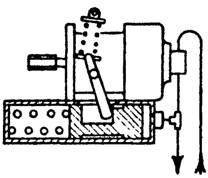
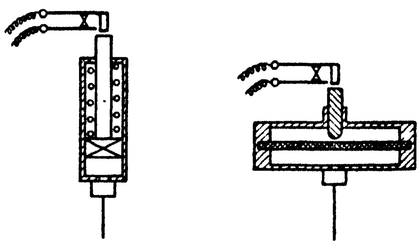


FIG. 63. PUMP CLUTCH CONTROL VALVE—SPRING TOGGLE TYPE



(*a*) (*b*)

FIG. 64. ELECTRIC PRESSURE SWITCH
(*a*) = Piston and spring. (*b*) = Diaphragm.

6.3. Electric Pressure Switches. A pressure switch is a device for opening the points of a switch when the pressure in the unit rises to a pre-determined amount, to enable the motor driving the pump to be switched off; obviously pressure switches can be used for other duties, such as the common oil-pressure warning light on a car, but the pump cut-out duty is the most important.

The elementary mechanism of a pressure switch is shown in Fig. 64 (*a*), and consists of a piston which rises against a spring until the switch contacts are opened. When used to control an electric motor the inertia of the motor will overcome the inherent likelihood of the opening of the contacts being insufficient, as occurs in an equivalent fashion with the crudest type of automatic cut-out.

A more interesting solution is to use a diaphragm, (*b*), which does not deflect linearily as does a spring, and an excellent pressure differential can be obtained.

Obviously, many types of cut-out could be adapted as pressure switches by replacing the by-pass valve by a switch.

CHAPTER 7

PRESSURE CONTROL VALVES

7.1. Relief Valves. The mechanism of relief valves is well known, as valves of this type have been used since the earliest days of pressure systems on steam engines. Although early valves used a weight to load the valve, modern pressure systems use spring loaded valves almost exclusively.

Essentially a relief valve consists of a valve held on to its seat and opened by pressure. The whole of the pressure energy of the fluid passing through the valve is converted to kinetic energy, forcing the fluid through the narrow opening of the valve at high velocity.

A simple calculation will serve to indicate the high order of the velocity and the small openings involved. For example, the lift of a valve of 0·25 in. diameter, set at 2 500 lb/sq in., is about 0·007 in. for a flow of 5 gallons of light oil per minute.

Fig. 65 shows the main types of relief valves. Type (*a*) is the classic solution using some form of spring loaded poppet valve, usually a ball on simpler designs, although preferably some form of guided poppet for high pressure work.

Type (*b*) is a similar valve using a tension spring, which improves the guiding of the valve.

With these types of valve, since the pressure tends to balance the spring load, the net force of the valve on its seat decreases steadily as the pressure rises. In practice this means that as the pressure approaches the opening point, the valve is very susceptible to mechanical vibration or pulsation of pressure, and "dribbles" or "weeps". To get over this, such valves have always to be set appreciably above the maximum system pressure (12–15 per cent). At high pressures where the velocity through the valve is high, this type of valve is usually noisy owing to valve instability—the flow tending to lift the valve.

Type (*c*) is fundamentally different in that the seat on which the ball sits moves as the pressure rises, compressing the spring until the valve hits the central abutment rod. The pressure has acted on the full area of the piston up to this point; the subsequent rise of pressure transfers the load on the ball proper from the seat to the central rod, and when the pressure has risen sufficiently, acting on the area of the piston less the ball area, the

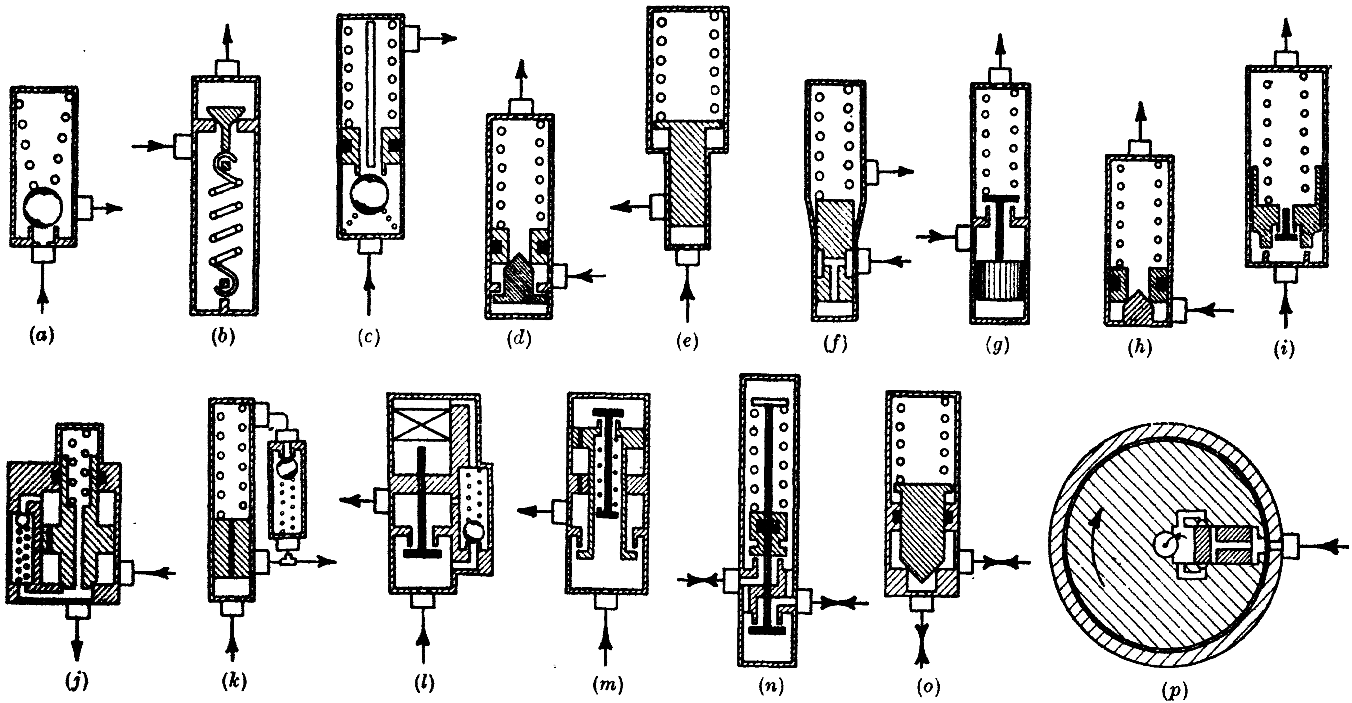


FIG. 65. RELIEF VALVES

(a) = Classic ball type.
(b) = Tension spring.
(c) = Inverted with ball.
(d) = Inverted with poppet.
(e) = Slide.
(f) = Slide with tapered orifice.
(g) = Damped poppet.
(h) = Moving seat.
(i) = One-way pressure dropping.
(j) = Pilot valve operated main poppet.
(k) = Pilot valve operated main slide.
(l) = Pilot operated "hydraulic lock".
(m) = Pilot operated "hydraulic lock".
(n) = 2-way acting.
(o) = 2-way acting.
(p) = Centrifugally loaded.

seat leaves the ball and the fluid flows past the ball as before. The pressure at which the ball just touches the rod is a fraction of the opening pressure given by the ratio—

$$\frac{\text{piston area} - \text{valve area}}{\text{piston area}}$$

In practical designs where the piston is, say, three times the diameter of the valve throat, this ratio is 8 : 9. This means that for all practical purposes, over the normal working pressure range, the higher the pressure the greater the force pressing the valve on to its seat, which is exactly the characteristic required to prevent dribble or weeping.

This type of valve can deal with high flows without noise, since all parts are stable and the piston is damped by seal friction. The small central rod, however, restricts the central passage, and there is a limit to the flow which can be passed, as otherwise the passage will restrict the flow and upset the valve performance due to choking.

Type (*d*) has equivalent characteristics to type (*c*), and owing to the shifting of the valve abutment, the central passage is unrestricted. It can thus handle higher flows than type (*c*), although at the expense of the practical advantage of type (*c*) that pressure setting can be by means of alterations in the length of the abutment rod.

Type (*e*) uses a lapped slide plunger instead of a poppet valve, and while not being perfectly pressure-tight, can be substantially leak-proof up to opening point; it is certainly more expensive than other normal types.

Type (*f*) is similar, with a conical section of the body at the end of the slide bore to allow gradual and "streamlined" increase in orifice area as the slide lifts.

Type (*g*) is a known variation of the plain relief type, but with a damping piston coupled to the valve. This addition can and does assist in preventing noise and valve instability.

Type (*h*) is known, but although it appears at first sight to be the same type as type (*c*), it is not similar since the valve is fixed. It is a simple inversion of a normal blow-off valve and functions exactly similarly, except for the inherent damping action of the seat gland which is probably beneficial.

Type (*i*) is a particular type of valve incorporating an internal non-return valve for free return flow. Although any type of relief valve can be by-passed in the return direction in an equivalent manner, the particular construction is convenient for dropping the pressure in a hydraulic line by a fixed amount, but in one direction only. This action should not be confused with that of a reducing valve which maintains the output pressure at a constant value, instead of subtracting a constant amount from it.

Type (j) is an interesting type known on aircraft and machine tools. The pressure first of all opens a pilot ball relief valve, but since the oil flowing through this pilot valve has to pass through a small hole in the piston diameter of the main valve, it unbalances the latter and opens it, allowing the energy to be dissipated through the main valve seat. When the pressure drops, allowing the pilot valve to seat, flow stops through the small restrictor hole, and the main valve, no longer unbalanced, closes under the action of its spring; this spring must be sufficiently strong to overcome the pressure acting on the sealed upper spindle diameter of the valve. Once the main valve is closed, the pressure acts on the masked throat area, less the upper spindle area, keeping the valve shut, in addition to the spring force already mentioned. The stability of the main valve depends entirely on the flow through the restricting orifice in it, and the fluid under pressure between the restriction and the pilot relief valve acts as the resilient spring tending to keep the valve closed, throttling the flow through the valve throat.

Type (k) is substantially similar to the previous valve, except that the main valve is of the slide type and therefore not completely leak-proof. A pilot flow through the small relief valve is used to unbalance the piston and move it against a spring, the piston being provided with a bleed hole to pass the pilot supply of fluid (see Chapter 8 for similar valves acting as flow dividers). A valve of this type would probably only be worth while for large and continuous flows as on machine tools.

Type (l) is somewhat similar to type (j), but consists essentially of a safety valve controlling a hydraulic lock. The action is not very satisfactory, as the natural tendency of the large valve would be to drop the pressure excessively, and the design probably works better as a cut-out (see Chapter 6). To enable the piston to seat, there is a permanent leak from the upper part of it which is compensated by a continuous flow through the pilot valve.

Type (m) is almost identical except that the pilot relief valve is carried in the main valve stem.

Types (n) and (o) are 2-way relief valves, operating equally well in either direction; they can be used, for example, in a double-acting jack system connected across the jack. Type (n) uses a single spring connected between the two valves by a tension rod. Type (o) uses a valve lifting normally in one direction of flow and balanced to lift at the same pressure for reverse flow, if the spindle area is twice the valve throat area.

Type (*p*) shows a relief valve, the spring reaction on which is replaced by the centrifugal force acting on the piston as it rotates. The valve is of the slide type, and obviously the relief pressure is proportional to the square of the rotational speed. This mechanism is used on governor systems.

Fig. 66 illustrates a special relief valve system where the reaction on a main valve piston is a function of the flow through another part of the system; in the case illustrated the setting of the main relief valve is proportional to the flow through a restriction valve up to a rate of flow which produces sufficient restriction to lift a secondary relief valve. Such a system is applicable to systems where some measure of viscosity control is to be exerted; for example, should the main flow be for bearing lubrication purposes, when starting up with cold bearings with excessive clearances, the main relief valve should be set higher to increase the main flow until the system warms up and bearing clearances decrease. By pumping a sample of the cold oil through the secondary circuit, the main relief valve can be set in sympathy with the fixed restrictor (which is not likely to dilate appreciably as it warms up), so that the flow is self-compensating. This is an interesting principle, obviously applicable in various ways to other special systems.

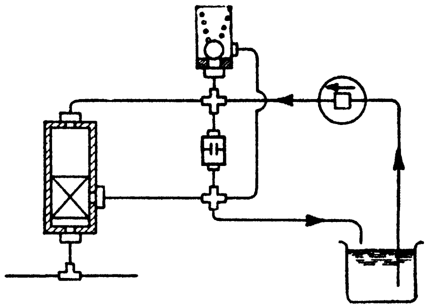


FIG. 66. FLOW COMPENSATED RELIEF VALVE

Valve Characteristics. The characteristics of the basic types of valve can be represented graphically as follows—

Fig. 67 illustrates the forces between valve and seat as the normal type of valve, (*A–B*), and on the inverted type, (*O–C–B*). As already explained, the force decreases progressively on the standard type, but on the inverted type the force increases along the line *O–C* until the valve touches the abutment at *C*, thereafter decreasing progressively along *C–B*. The shaded area shows the important advantage of this type, since this is a measure of the extra seating force on the valve. The extra slope of the line *C–B* is particularly important. Below half-pressure the force on the plain valve is actually greater, but this is unimportant. The point C, as already pointed out, corresponds to a pressure of $\frac{D^2 - d^2}{D^2} \times$ opening pressure.

Fig. 68 illustrates more or less accurately the way the spring force is distributed between the seat and fluid pressure on an

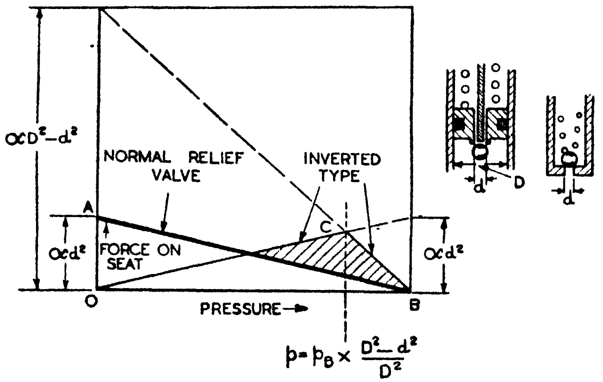


FIG. 67. RELIEF VALVE SEAT FORCE/PRESSURE CURVE

ordinary relief valve. The exact point of opening and the pressure differential are somewhat complex, but the action is approximately as shown.

Fig. 69 illustrates the various forces on the inverted type of valve (Fig. 65, types (*c*) and (*d*)). *O–C* is the curve of pressure load

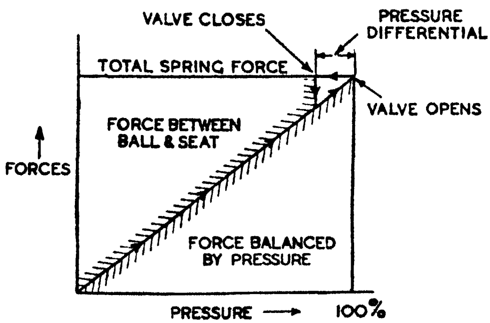


FIG. 68. CHARACTERISTIC CURVE—NORMAL RELIEF VALVE

on the piston proper, less the valve area. *O–B–C* is the spring load, *B* being the abutment rod contact point. Ordinates between *B–D* and *B–C* represent the force on the push rod, dropping to zero along *D–C* as the valve opens. The vertical ordinate between *O–B–C* and *O–C* represents the force between valve and seat. The pressure differential is too complicated, owing to friction, to represent simply. Owing to friction the opening point is certainly above the nominal point as determined by spring load.

Relief valves mounted in series are known; this device is sometimes used to limit the energy dissipation of a single valve to avoid wear or abrasion of the seat. The pressure drop at each stage is substantially the same as if the valve were dropping the same amount to zero; in other words, the functioning of a relief valve is generally independent of the back pressure—a valve set at 1 000 lb/sq in. will usually work equally well between 2 000 and 1 000 lb/sq in.

7.2. Reducing Valves. Although a relief valve can be used as a reducing valve, since it can drop the pressure in a system, the

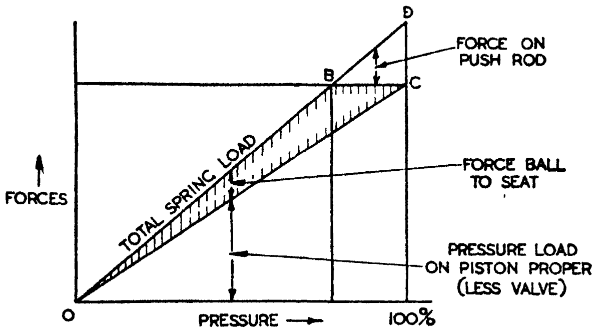


Fig. 69. Characteristic Curve—Inverted Relief Valve

reduction is necessarily a fixed arithmetic amount and independent of the inlet pressure, assuming the latter is above the opening pressure of the valve. A reducing valve proper maintains a fixed outlet pressure irrespective of the inlet pressure, provided only that the latter is above the outlet pressure required. A reducing valve can usually be identified as one where a spring opens the valve element and the output pressure tends to close it; it is thus the exact opposite of an isolation valve—see para. 7.6.

Fig. 70 (*a*) shows the fundamental mechanism of a reducing valve. Fluid passes through the inlet to the outlet, past a valve held open by a spring. As the pressure in the outlet builds up, it reacts on the diaphragm, piston, bellows, etc., dividing the body, and gradually overcomes the spring force, closing the valve until the latter is open just enough to maintain the necessary pressure balance. Thus the outlet pressure remains constant irrespective of the inlet pressure (if the pressure on the head of the valve, relatively unimportant, is ignored), and is a function only of the spring tension. In practice, reducing valves have

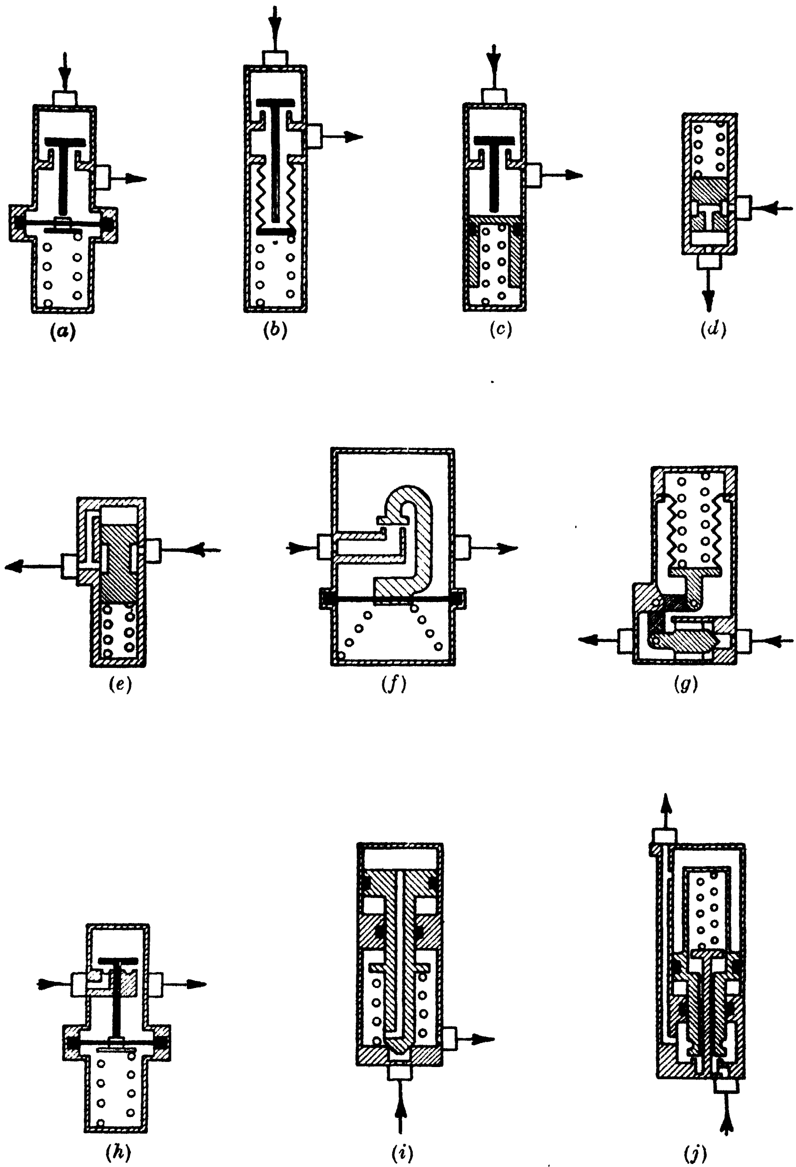


Fig. 70. Reducing Valves

(a) = Diaphragm.
(b) = Bellows.
(c) = Piston.
(d) = Slide.
(e) = Slide.
(f) = Pressure opening valve.
(g) = Streamlined.
(h) = Annular valve seat.
(i) = Differential piston.
(j) = Differential piston.

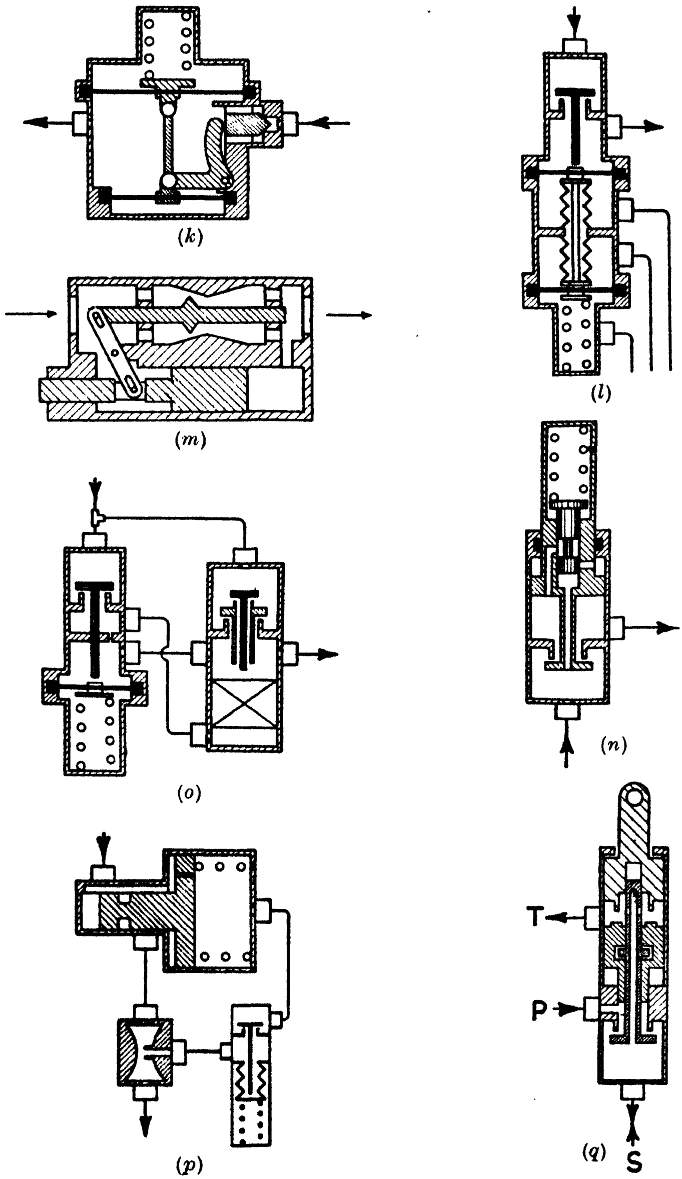


FIG. 70 (*contd.*). REDUCING VALVES

(*k*) = Differential diaphragms.
(*l*) = With auxiliary chambers.
(*m*) = Flow restricting.
(*n*) = Pilot operated.
(*o*) = Relay operating.
(*p*) = Relay operating.
(*q*) = 3 pressure type.

remarkably constant performance, and can be accurate to about $\pm$ 1–2 per cent. They are sometimes used in series to improve accuracy.

The flow through a reducing valve is at a maximum when the valve is wide open on its stops, but obviously decreases until, when the valve is almost balanced, the flow is very small. Thus, for rapid operation the pressure of utilization of a system should be well below the output pressure of the valve; also the rate of the spring should be as low as possible, so that the valve movements should be as rapid as possible during the last part of the pressure rise; in fact constructions with pneumatic springs are known.

Type (*b*) in Fig. 70 shows the bellows construction referred to, while type (*c*) uses a piston. The influence of gland friction on this type has some small effect on performance and accuracy, but makes a high pressure reducing valve possible, since neither a bellows nor a diaphragm can be used at high pressure.

Type (*d*) is a slide type more suitable for hydraulic applications. Type (*e*) is virtually identical except that the communication from outlet to reaction area is external. The similarity of these constructions with the isolation valve of Fig. 76 (*b*) will be noted.

Type (*f*) is a variation with an unrestricted valve throat.

Type (*g*) is a further variation giving straight through flow to the fluid, and avoiding the losses due to turbulence.

Type (*h*) uses a valve pulled down on to the inlet seat, but tending to be opened by the pressure.

Type (*i*) is a piston variation using a differential area piston but two glands. Type (*j*) is almost identical. Two diaphragms of different size could be used as illustrated in type (*k*).

Type (*l*) is a known type embodying several chambers which can be subjected to positive, negative, or differential pressure to influence the valve setting in various ways from external sources of pressure.

Type (*m*) is a streamlined type with a piston and lever, and suitable for high rates of flow; it is in some sense a flow control valve, not a reducing valve, since it will not close off the flow completely.

Type (*n*) is a pilot operated reducing valve; normally the flow lifts the valve and passes to the system, the internal spring holding the slide selector in a position connecting both sides of the piston and thus balancing it. When the output pressure rises sufficiently, the slide plunger lifts and shuts off the upper surface of the piston, thus unbalancing it and closing the valve.

Type (*o*) uses a reducing valve to control a "hydraulic lock".

Since the latter has a pilot valve, small changes in flow are dealt with by it, the main valve only opening for large flows. A valve of this type will deal with large flows as it approaches the balance pressure, since the main valve does not close progressively with pressure. Pilot operated reducing valves are known in other more or less obvious forms, and are very satisfactory for handling large volumes.

Type (*p*) is a special valve used in railway brake systems. In order to overcome the inherent defect of a reducing valve, already mentioned, of restricting the flow when the ratio of input to output pressure is small, and to increase its flow capacity, this type uses a venturi to "overbalance" the reducing valve. Pressure enters the valve through a slide shut-off valve, normally spring loaded to the closed position, and fully pressure balanced. When the pressure in the output line drops, the diaphragm in the reducing valve no longer balances the spring and the valve opens, allowing pressure behind the slide piston to drop. As the piston has only a small bleed hole in it, it becomes unbalanced and opens the slide cock; fluid now flows through the venturi to the system. The flow in the venturi causes a drop in pressure which over-balances the diaphragm and opens the slide cock even more. Thus the opening of the cock is proportional to the flow as well as to the back pressure; this means that not only is the flow capacity increased by the use of a relay, but the assembly is much less sensitive to back pressure in the output line, and will maintain high rates of flow up to pressures closer to the cut-off point. The degree of benefit derived from the venturi will be proportional to the pressure reduction in it as compared with the system pressure.

Type (*q*) is a special reducing valve with only two pressure settings (other than zero), and is used in a special relay hydraulic system (see Chapter 12). The connections *P*, *S* and *T* refer respectively to Pump, System and return to Tank. When the moveable end fitting is released and abuts on the end of the valve body, the system pressure is released to tank and drops to zero. The pump is isolated because the intermediate floating piston moves upwards to balance the main valve, which then remains closed under the action of the spring. When the end fitting is moved in half-way, it seals off the tank connections and admits pump pressure to the system. Since the floating piston is now centralized the pump pressure tends to open the main valve on the annular area between head and stem, while the system pressure acts on the whole valve area. The pressure is thus reduced in this ratio. When the end fitting is moved further, the inner collar on the valve stem abuts on the floating piston and the pressure in the system is raised to a maximum.

7.3. Pressure Regulators (Brake Control Valves). There is a family of valves very similar to the reducing valve and widely used on all power brake systems (steam, air, vacuum and hydraulic) on trains, vehicles, aircraft, etc., enabling the driver or pilot to regulate the brake pressure by a corresponding variation in effort on or movement of a hand or foot lever. Such valves are used for other purposes as well.

If a reducing valve is fitted with a lever or equivalent mechanism, allowing the spring compression to be varied by the driver, the pressure in the brake connected to the outlet of the valve will increase proportionally to each increase in deflection of the spring (and thus each increase in effort exerted by the driver). In early steam brake systems a simple reducing valve was used, but to enable the pressure to be released (since a reducing valve can obviously not work backwards), a release valve was opened when the brake was fully off. A much improved valve was then evolved, using a release valve seat as well as an inlet valve, so that as the pedal or lever was released, the pressure in the brake was released a corresponding amount. Fig. 71 (*a*) makes this clear. With the brake off, the spring is free and the exhaust valve open to tank or atmosphere. Initial movement of the pedal or lever closes this valve, and then opens the inlet valve, normally held on to its seat by the inlet pressure. Subsequently the pressure builds up until it reacts against the piston and spring, and closes the inlet valve. Should the brake pressure rise slightly more, or should the pedal be released a little, the exhaust valve would open long enough to release just sufficient fluid to drop the pressure to that corresponding to the spring tension. In this sense the exhaust valve acts exactly like a relief valve. Thus each position of the pedal, and each effort exerted by the driver, corresponds to a single value of brake pressure, which is the characteristic feature of a brake valve. As in the case of the reducing valve, the flow velocity is reduced progressively as the balance point is reached and means of overcoming this may be desirable. One method is to use a permanent magnet between the piston seat or diaphragm and the body, attracting them so that the inlet valve is urged to remain open. This upsets the natural flow characteristics of the valve beneficially.

With many brake systems, particularly pneumatic ones, the exhaust of the fluid requires large valve openings to achieve adequate speed of release (since the pressure available is less). This may involve appreciable pedal or lever movement when applying the brakes to close the exhaust valve. One proposal to reduce this travel uses a development of the early type of valve referred to above. In place of an exhaust valve opened by complete

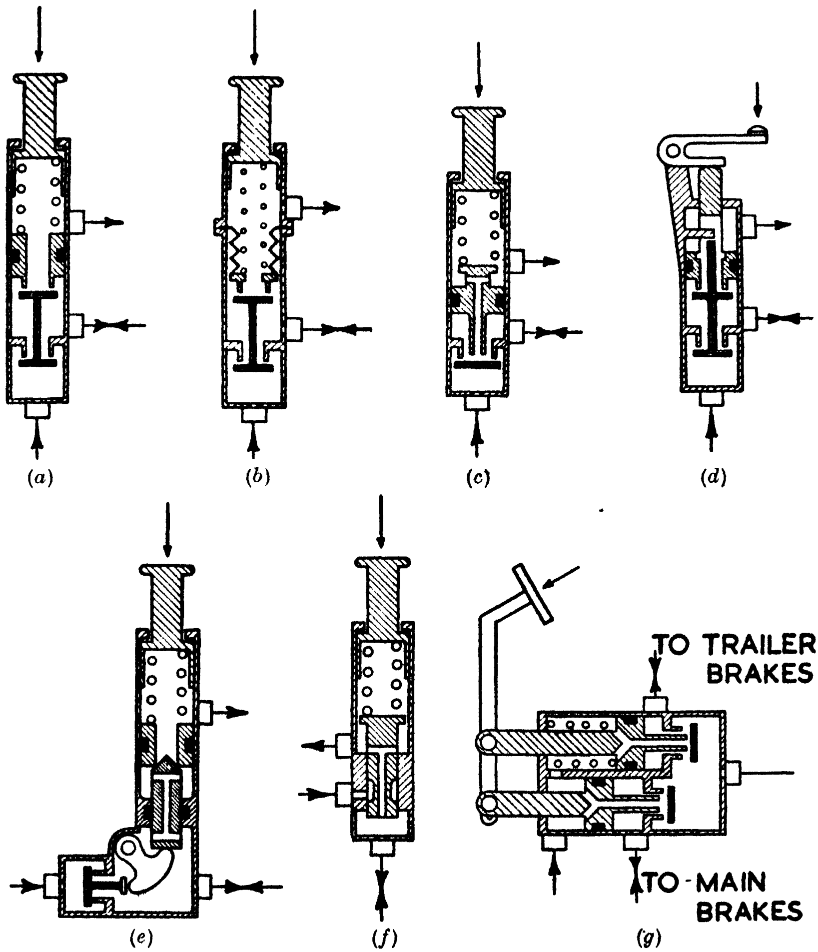
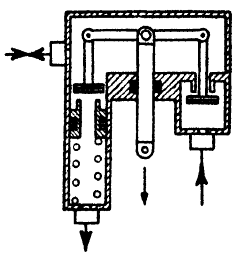


FIG. 71. PRESSURE REGULATING VALVES
(a)–(f) = Force balance types. (g) = Special duplex unit.

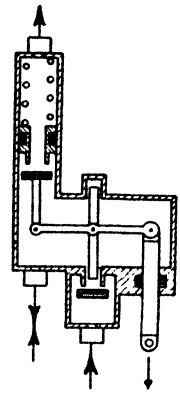
release of the pedal or lever, is a complete 3-way selector arranged to exhaust directly as before when the pedal is released, but also admitting pressure to the brake valve proper when the pedal is moved initially. The inlet valve of the brake valve is adjusted to be slightly open initially so that a small pedal travel only is necessary to open it widely for rapid brake application. Gradual release of fluid takes place in the normal manner but complete release is also through the selector exhaust valve which opens when the pedal is fully released, as already mentioned.

Type (*b*) is an obvious solution, using a bellows in place of a piston with glands. Almost any 3-way selector valve element could be used in a brake valve, but type (*c*) shows a useful solution with a single valve, using the selector illustrated in Fig. 52 (*f*).

Types (*d*), (*e*) and (*f*) show actual solutions diagrammatically. Type (*d*) is interesting because a tuning fork spring is used. Type (*e*) uses a bell crank to operate the inlet valve, for a reason



(*a*)



(*b*)

Fig. 72. Pressure Regulating Valves—Lever Balance Type

not apparent. Type (*f*) uses a slide valve in place of poppet valves, the slide plunger itself forming the reaction piston.

Type (*g*) is an ingenious valve for heavy vehicle brakes. The valve consists of two brake valves coupled together on a balance lever, the upper valve normally being open by reason of its spring. The lower valve supplies the main vehicle brakes in the normal manner, but the trailer brakes are applied by *reverse* pressure, i.e. are spring engaged and pumped off. Thus the normal position of the upper valve is open. When the pedal is applied the first motion applies the main brakes through the lower valve, the upper valve remaining open and acting as a pivot for the lever. Subsequent action on the pedal closes the upper valve and applies the trailer brakes by releasing the air in them. The relative braking amounts, etc., depend on the spring tension and the lever. The use of reverse braking in conjunction with normal pressure braking economizes in the consumption of air to apply the brakes and thus makes the performance better.

Fig. 72 illustrates a type of brake valve rather different from the previous types, and using a differential lever principle (see Chapter 15). Instead of the abutment of the spring being moveable under the action of the driver or pilot, thus varying its deflection, a lever is introduced to move the valve itself. The illustration of type (*a*) will make this clear; the control connection shifts the pivot of the internal balance lever, and thus changes the position in which the inlet and exhaust valves strike a balance with the spring tension. Otherwise the action is the same as before. There is, however, no (or almost no) reaction on the driver's control, and the pressure balance is a function only of the movement of the control (i.e. the pivot). In the normal type the pressure balance is a function of the deflection of the spring, it is true, and in this sense also a function of linear movement, but reaction is necessarily present to an appreciable degree.

There are several methods of achieving kinematically equivalent mechanisms to type (*a*), for example, a floating wedge separating the two valves. Type (*b*) is a well-known aircraft brake valve.

Fig. 73 (*a*) shows a slide type of valve completely devoid of reaction to the driver. A 3-way slide selector admits fluid to the system or exhausts it, and since it is surrounded with a floating sleeve with an enlarged diameter forming a reaction cylinder connected to the brake, the sleeve follows up until the pressure is cut off. The sleeve is spring loaded, and thus the pressure in the brake is a function of the spring deflection, since the sleeve piston must hold the sleeve against the spring; braking is therefore fully progressive as the driver's rod is pulled. Type (*b*) shows a similar unit, except that a reaction piston is added to give the driver a direct sensation of braking effort. The brake pressure acting on the piston connected to the driver's control acts in this manner, the other piston connected to the inner slide acting as the balance piston to overcome spring pressure and thus determine the brake pressure. If the reaction piston is omitted, this unit becomes identical with the previous one, with the slides reversed.

Fig. 73 (*c*) shows what amounts to an open centre power brake valve, although it is from some points of view a servo system of the type described in Chapter 15. Normally the valve connected to the pedal is open and allows the pump to idle at zero pressure. When the pedal is depressed, the by-pass is restricted and the pressure builds up, moving the jack and applying the brake. At the same time the differential lever moves the valve body to open up the by-pass again, until a state of pressure and force balance is reached. It will be noted that the valve head connected to the pedal carries a piston diameter which is subjected to the

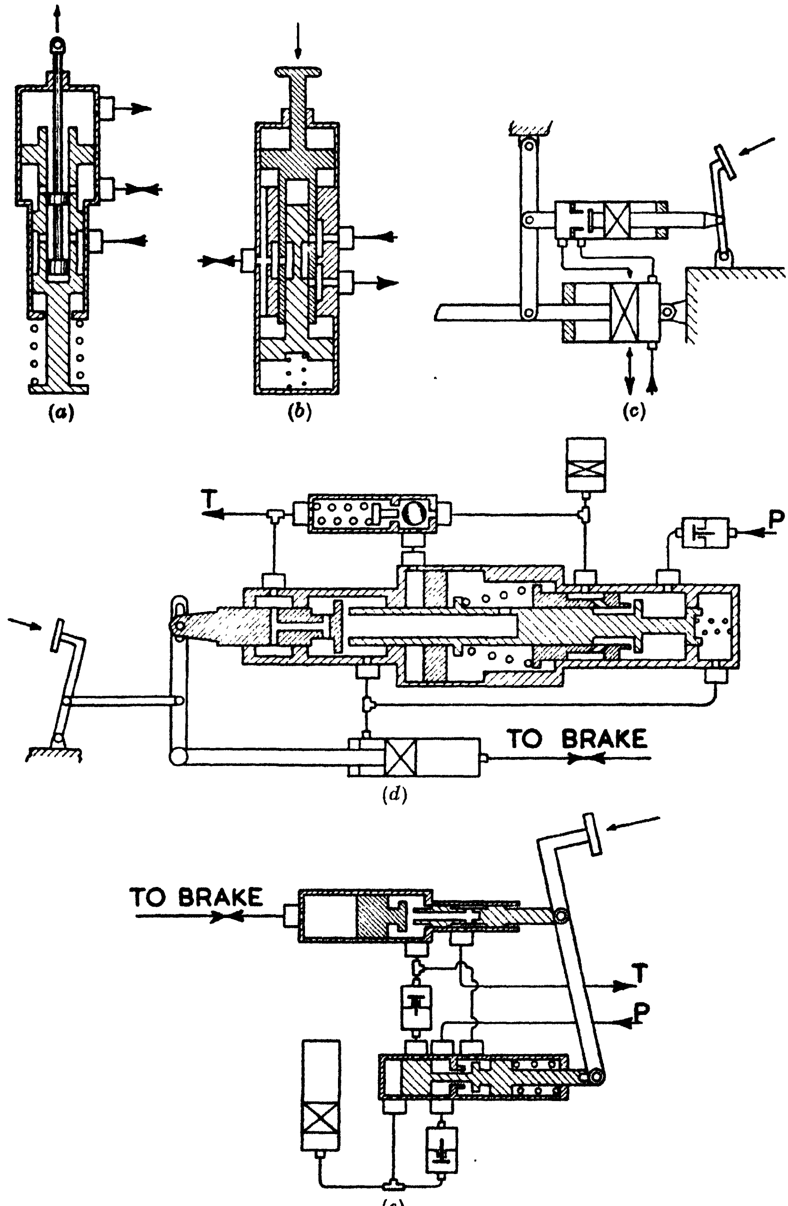


FIG. 73. MISCELLANEOUS PRESSURE REGULATING VALVES
(*a*), (*b*) = Slide type. (*c*)–(*e*) = Servo system.

system pressure and therefore gives the operator a degree of feel (see Chapter 15). Instead of the valve being mounted between the pedal and the differential lever, the pedal can operate on the middle of the lever and the valve can be connected between the other end and a fixed point.

Fig. 73 (*d*) illustrates a rather complicated, although somewhat similar, valve which acts on the brakes through a piston, i.e. by means of a simple hydraulic remote control. The diagram has been simplified to omit the feeding arrangements which will, in practice, be necessary (see Chapter 12). Fluid under pressure from a pump passes through the lower valve unit and fills a power accumulator through a non-return valve. The valve in the unit is kept closed by a spring. When the pedal is depressed, initial movement closes the tank valve on to the floating piston of the upper assembly, and then exerts a direct force through this piston on to the brakes; this initial motion is intended to "take up the slack" by filling the brake lines and advance the brake shoes into contact with the brake drums. When the reaction in the brakes rises to a given amount, the spring of the lower valve assembly is compressed, opening the valve to admit pump pressure to the upper cylinder to boost the pedal effort. Reaction feel is obtained as in the valve previously described. When the pedal is depressed fully, or rapidly, the slide piston on the end of the lower valve moves to the right to admit accumulator pressure to the brake operating piston, thus providing a substantial flow for rapid application of the brakes; at the same time it blocks the connection between pump and accumulator, because if it remained open when the accumulator was deflated, the pump would not be able to operate the brakes and would inflate the accumulator instead. When the pedal is released, the brake return springs force the separator piston back and the fluid in the cylinder can escape to tank.

A further and even more complicated unit is shown in Fig. 73 (*e*). Initial pedal movement closes the tank valve in the upper assembly, and acts directly on the brakes as before. Pump pressure cannot reach the brake cylinder owing to a cylindrical restriction (a small slide valve) on the end of the central upper valve. Pressure can, however, reach the accumulator to fill it. The unit acts as an automatic cut-out by virtue of the ball valve element which will open at a predetermined pressure and admit pressure to an annular piston; this in turn moves to the right, compressing the spring, closing the moveable valve seat on to one valve, and opening the slide valve. The pump can now idle to tank.

When the pedal is depressed, the tank valve restricts the free return of fluid and builds up brake pressure exactly as in the case of the unit illustrated in Fig. 73 (*c*). The ball valve is provided with two seats, the outlet seat being larger than the inlet seat; the operation of this valve is identical with that of the cut-out illustrated in Fig. 60 (*k*).

7.4. Differential Braking Devices. An important feature of many aircraft brake control valves is a device enabling port and starboard brakes to be applied differentially to assist in steering the aircraft.

If the two brake valves are applied through a lever, the centre of the lever being moved and the extremities connected to the two valves, each valve is deflected an equal amount since the load applied to each is half the total force exerted. Should the pilot's force not be exerted on the centre of the beam, the force on each valve is inversely proportional to the relative leverages, and thus the brake pressures are similarly different. Within limits, the sum of the two brake pressures remains constant, but each value changes differentially as the beam pivot is changed.

Fig. 74 illustrates some of the means of achieving this object. Type (*a*) uses a differential lever, pivoting about a roller on the end of an arm. To apply pressure the arm is moved bodily towards the valves; to brake differentially the lever is unbalanced by rotating the arm about its axis, thus moving the roller to one side. The lever then rotates until the forces in each valve balance the leverages; since the force in the valve is due to spring deflection, in turn proportional to the brake pressure, the two brakes are applied in proportion to the leverage.

Type (*b*) uses a swash plate or equivalent cam to apply one brake while the other is relieved, if the cam is rotated, both being applied when the cam slides along its own axis. Type (*c*) uses a bevel differential; rotation of the whole assembly about the common axis of the two bevels, opens or closes the brake valves through cams on the spindles. Rotation of the central bevel applies the brakes differentially.

Type (*d*) uses a guiding slot for two levers which operate the brake valves. If the slot is central the two brakes are applied equally as the central pin slides down the slot. If the slot is rotated braking is differential, although the action is geometrically imperfect.

Type (*e*) is a curious mechanism used with the pull type of valves illustrated in Fig. 72 (*b*). Pulling on the control lifts both levers and the brake valves equally, the two inclined and interconnected levers determining the motion by resisting the pure rotation of the main links. By disturbing the symmetry of the two inclined links the brake valves are pulled by a different amount.

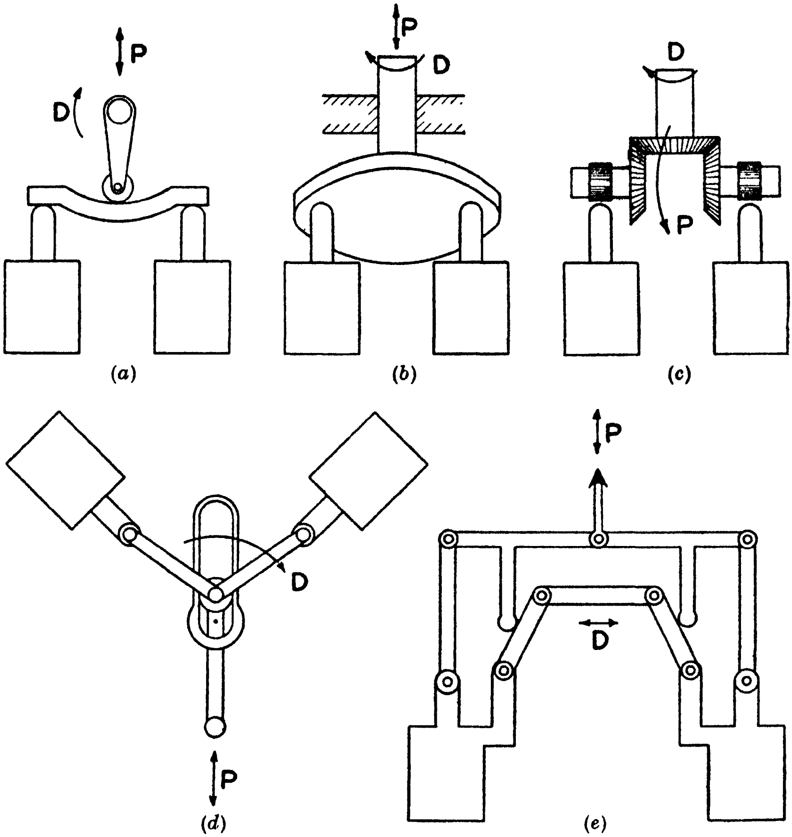


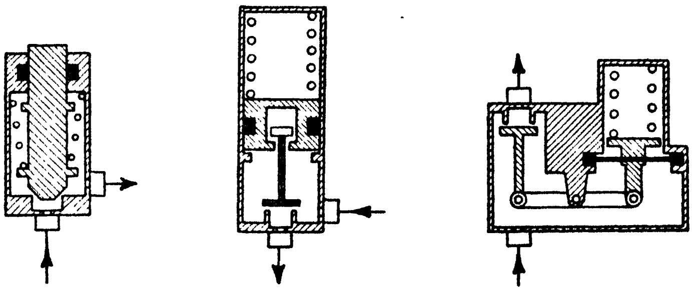
FIG. 74. DIFFERENTIAL BRAKE MECHANISMS

(a) = Hinged lever.
(b) = Swash plate.
(c) = Differential gear.
(d) = Inclined slot.
(e) = Variable linkage.

7.5. Pressure Maintaining Valves. A pressure maintaining valve is one which maintains a particular pressure in part of the system for some special purpose. It exists in a variety of forms for particular purposes, but an obvious use is to enable an air bottle, for example, to be inflated from a compressor at the same time as others, but will not allow deflation of this bottle at the same time as the others beyond a certain (i.e. "maintained") point; this might, for example, leave the special bottle for emergency use. Connected another way it can prevent the inflation of one bottle until a first has reached at least a given pressure.

In other cases a pressure maintaining valve might be used to maintain the pressure in the system before the valve, at least at a certain pressure (e.g. for brake operation), even if the system beyond the valve was operating at a lower pressure; or with several units, to prevent several brakes operating independently and unsynchronized.

This type of valve is similar to a relief valve until it opens, when it becomes unbalanced and remains open, without offering the restriction and the loss of power in the system beyond it, which a plain relief valve would cause.



(*a*) (*b*) (*c*)

FIG. 75. PRESSURE MAINTAINING VALVES

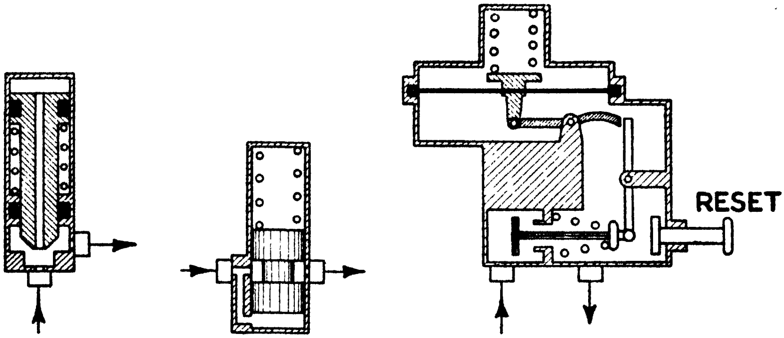
Fig. 75 (*a*) shows the mechanism of a pressure maintaining valve. It will at once be seen to be an over-balanced relief valve. It opens at a pressure determined by the spring force plus any back pressure, and closes again when the pressure has dropped down to that equivalent to the spring force on the unbalanced spindle area.

Type (*b*) is similar, although the non-return valve, which would have to be added to type (*a*) to allow back-flow, is eliminated by allowing a degree of free travel to the valve.

Type (*c*) is a known solution used on pneumatic systems, in this case a lever being used and a diaphragm replacing the piston.

7.6. Isolation or Pressure Cut-off Valves. The converse of a pressure maintaining valve is a valve which isolates part of the system when the pressure rises to a certain amount. This may be useful, for example, to prevent damage to some part of the system, or to limit the pressure applied to rear brakes on cars; as applied to trailer brakes, for example, the cut-off point may conveniently be adjustable.

Fig. 76 (*a*) shows the fundamental mechanism. The valve is normally held open by the spring and allows free flow in both directions. As the pressure rises the valve moves downwards owing to the pressure acting on the difference in areas of the two seals, until the valve is closed, preventing further rise in pressure at the outlet. Increase in pressure on the inlet merely seals it



(a) (b) (c)

FIG. 76. ISOLATION OR PRESSURE CUT-OFF VALVES

all the more. The valve opens again normally if the pressure falls. Type (*b*) is a slide valve equivalent.

Type (*c*) is similar, except that the valve is released by a detent operated by the diaphragm; the valve has to be re-cocked by pushing a knob. As drawn, the valve closes when the outlet pressure *drops* to a particular value. The operation would be similar to other isolation valves if the connections are reversed as well as the action of the detent release lever.

CHAPTER 8

FLOW CONTROLLING VALVES

8.1. Hydraulic Locks. A hydraulic lock is a valve which prevents flow in a particular line until fluid from another line arrives to permit it (i.e. "unlocking" it). Fig. 77 (*a*) shows the fundamental type. When the piston is down, flow of fluid is prevented from passing from *A* to *B*; when fluid pressure arrives at *C* the valve opens. Normally the valve is a loose poppet, so that free flow from *B* to *A* is possible, but balanced slide valves preventing flow in either direction are possible, although rarely necessary. The principal use of a hydraulic lock is to prevent motion of a jack due to external force (*A* being connected to the jack) until fluid arrives at the other end of the jack (and to *C* which is connected to it) to permit the motion. Kinematically it is equivalent to an ordinary spring loaded latch lock.

Type (*b*) is similar, but uses a diaphragm to isolate the top of the piston. In type (*a*) any pressure on top of the piston, due to pressure in the line to which it is connected, may upset the opening of the valve by partially balancing the piston.

The effect of back pressure in fact can be serious when the lock is used to prevent the motion of a jack against appreciable load. Unless the ratio of operating piston to valve is large, the unlocking pressure may be high; when the valve does open, the piston of the jack will move more or less rapidly, causing on the one hand a drop in pressure in the unlocking line tending to shut the lock again, and on the other hand a high instantaneous flow which may cause a back pressure on the piston in type (*a*), or the valve spindle in type (*b*); the latter construction is obviously better from this point of view. The result is that the unlocking action is jerky and may cause chatter and vibration.

Type (*c*) is a double unit, *B* and *D* being connected to the jack and *A* and *C* to the 4-way selector. Flow from *A* to *B* moves the jack, return flow from *D* to *C* being possible since the piston opens this particular valve. Motion in a reverse direction occurs similarly. When flow at *A* and *C* stops, the jack cannot move in either direction owing to the two valves being closed, and it can thus be locked in any position desired.

Types (*d*) and (*e*) function in a similar manner, except that two pistons are used—an unnecessary complication. In both cases A and C are connected to the selector, and B and D to the jack. In the case of (d) the fluid passes through one valve before unlocking the other, while in (e) the unlocking valve could be

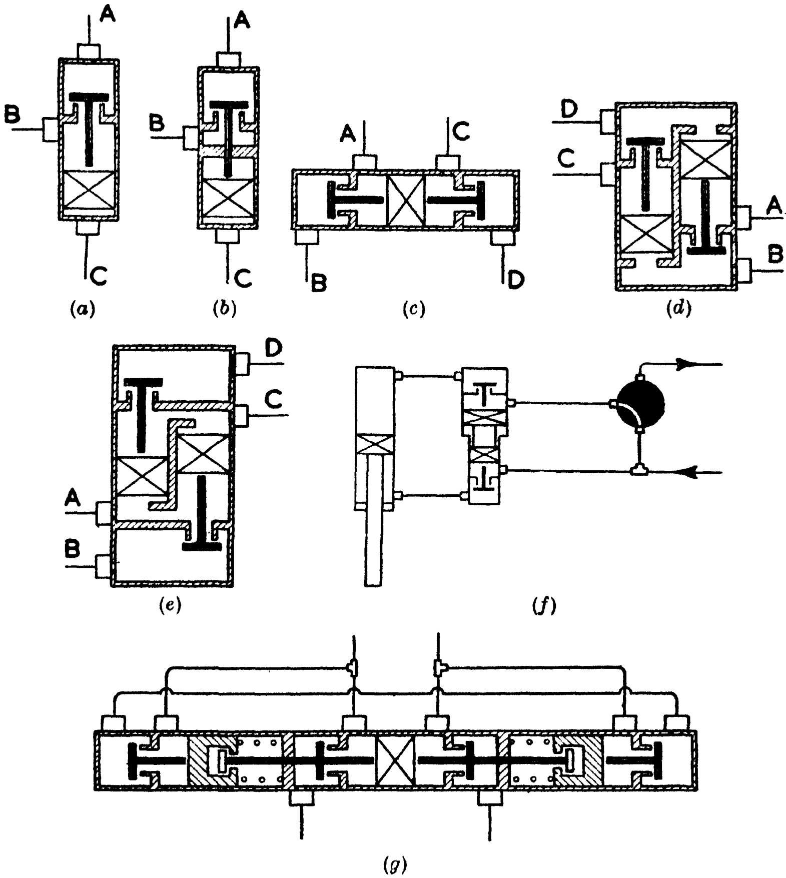


FIG. 77. HYDRAULIC LOCKS

(a) — Simple type.
(b) — With diaphragm.
(c) — Two-way, single piston.
(d) — Two-way, double piston.
(e) — Two-way, double piston.
(f) — Differential piston.
(g) — Special type.

opened before the fluid passed to the jack. The difference is a slight one—thermal expansion of the fluid in the jack, for example, would in one case cause a rise in pressure, and in the other would move the piston, although without much effect.

Type (f) is a double-acting lock for use with a differential jack system (see Chapter 11); in this the piston areas in the lock correspond to those of the jack since there is a continuous pressure applied to the annular volume of the jack. If the two lines to the selector are broken the two valves lock the jack. When the pump pressure is on, however, the closing of the jack can only be prevented if the selector is shut off to allow the pressure above the jack piston to build up and balance the lock piston.

Type (*g*) is a special unit used with some two pipe remote control systems which are under continual pressure from a compensator-accumulator (see Chapter 12). When the whole system is under pressure, the two spring loaded pistons are forced inwards against their springs, and the action of the lock is normal. If, however, the compensator-accumulator should become deflated the springs extend and open all four valves, short-circuiting the whole system and allowing the jack involved to centralize or otherwise move under external force.

8.2. Flow Dividers. A flow divider can be considered as a valve or device for dividing the output of a pump, etc., in a particular ratio between two pipe lines, with the particular case of by-passing a fraction of the total flow to tank or atmosphere. It is possible to divide flow between more than two lines, but this has probably not been attempted in a single unit.

It may be simpler to consider the by-pass type of valve first, as illustrated in Fig. 78 (*a*). Fluid under pressure enters at *A* and passes through the restricting orifice in the piston to the system or jack at *B*. If the piston were fixed, as in a plain restrictor, the amount of fluid passing through it would be a function of the pressure difference between *A* and *B*. As the piston is free, however, it moves against the spring as the pressure at *A* rises and eventually uncovers the by-pass port *C*. It acts in this way as a safety valve so that the pressure difference across the orifice remains constant, and thus the flow through to *B*, surplus being relieved through *C*. When used in a normal system a non-return valve will be used to allow free return flow, the piston sealing off the by-pass connection *C*.

Type (*b*) is similar, except that it acts in either direction. Type (*c*) is again similar to type (*a*) except that a poppet acts as the relief valve member, the connection *A* and *C* being reversed. The advantage of this construction is that the port *C* can be sealed more perfectly than with a sliding piston.

Fig. 78 (*d*) shows a by-pass flow divider which operates rather differently, using a venturi. The flow enters the valve and passes through a venturi tube; the reduction in pressure in the latter is made to suck a spring loaded piston against the spring, thus opening a by-pass connection. Thus a state of balance is reached where part of the flow is by-passed, and any increase through the venturi will move the slide to open the by-pass further, thus compensating for the increase in flow. This type of valve is probably less reliable and accurate than the relief valve type described previously. Another similar venturi type of divider is shown in Fig. 78 (*e*). In this case the main flow from A to C is regulated by the drop in pressure in the venturi throat as before, but in addition the by-pass flow from A to B is controlled by a port at the end of the regulating plunger, the flow being completely blocked when the input flow is small.

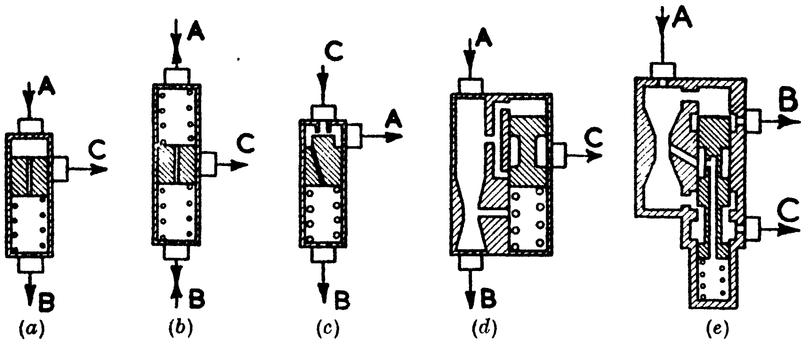


FIG. 78. FLOW DIVIDERS—BY-PASS TYPE

It will be realized that energy is dissipated in relieving part of the flow in these dividers which are thus relatively inefficient.

The basic principle of the flow divider proper can best be understood by reference to Fig. 79 (*a*). Flow from the pump passes through two pairs of restrictors, R_1 and R_3, R_2 and R_4, to the two jacks, systems, etc. Normally if the loads on the jacks are equal and the restrictors symmetrical in magnitude, the flow through each branch of the system will be equal and the pressure on either side of the floating piston at A and B equal. This floating piston is connected to the restrictors R_3 and R_4 so that as the piston moves it varies them. Assume now that the back pressure on one branch drops owing to the jack load dropping; the flow through this branch will tend to increase momentarily since the drop across, say, R_3 increases; the drop across R_1 immediately increases and thus the pressure at A is reduced. The floating piston is now unbalanced and it moves over towards A to restrict R_3 (and open up R_4) to maintain equality of flow in both branches. It is not strictly necessary to have both R_3 and R_4 differentially variable, variation of either being sufficient but less effective. Equal "addition" of reverse flow can be obtained by connecting R_3

and R_4 up to the opposite ends of the piston so that the motion of the piston varies them in the opposite sense. Since the resistances are in circuit continuously, energy is dissipated and this type of flow divider is thus comparatively inefficient.

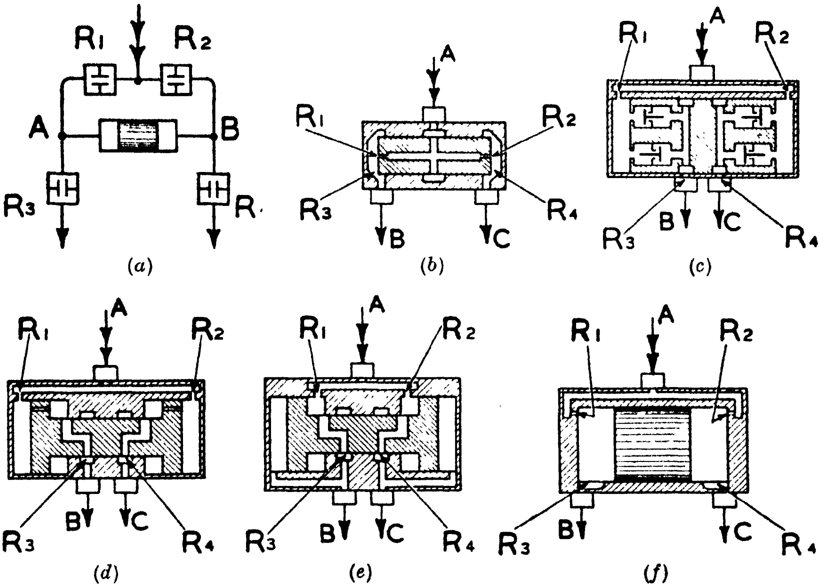
A typical practical construction will be seen in Fig. 79 (*b*). Fluid from the pump or system enters at *A*, and passes to the two branches of the system (jacks, etc.), at *B* and *C*. As it passes through the two fixed resistances (R_1 and R_2) in the central floating piston each part of the flow has to pass through a restricting orifice (R_3 or R_4) formed between the ends of the piston and the body (or by needles or other types of valve). Thus the piston tends to centralize so that the pressures on each end are in equilibrium; thus the flow at *B* and *C* remains equal in spite of back pressure variations in the lines due to unequal loading.

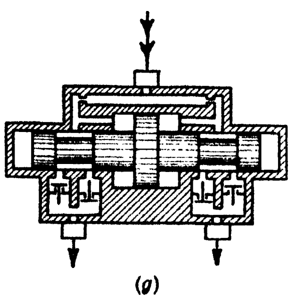
Type (*c*) is somewhat similar to type (*a*). Flow entering it passes through fixed orifices and then through valves in a floating piston to the two outlets *B* and *C*. If the flow is symmetrical the piston is balanced and it remains central with the ports at *B* and *C* equally open. Should the flow at either outlet increase or decrease, the piston moves in a direction to throttle the flow to maintain equality. As the piston is fitted with a second set of valves it can operate equally well in either direction to allow two jacks, for example, to be released at equal speed.

Types (*d*), (*e*) and (*f*) are similar to those just described. In (*d*) the flow through orifices in the piston heads of the dumb-bell slide causes equal pressure on the two ends; pressure in the annular volumes behind the heads of the pistons is built up owing to the restricting orifices where the fluid passes into the central slide cylinder; these back pressures balance also, and the flow through each secondary, variable orifice must be equal, otherwise the back pressure will be unequal. Thus the slide moves until the necessary balance is obtained.

Type (*e*) is very similar, but works the other way round to balance the return flow from two jacks.

Type (*f*) is again similar, except that a single piston is used, the chambers at each end having restricted inlets and outlets with a narrow slot restriction which can be partly masked by the piston; in this case, however, the piston is deliberately allowed some movement before masking occurs so that its displacement itself compensates for instantaneous changes of flow. The flow divider shown in Fig. 79 (*g*) is arranged to divide flow in both directions and can thus synchronize both directions of motion of a pair of jacks. In one direction the action is as before but for reverse flow, since it is the inlet orifices which are variable, the moving piston is connected up differently so that it enlarges an orifice instead of throttling it as is necessary for normal flow. The changeover of orifices is effected automatically by the four non-return valves.





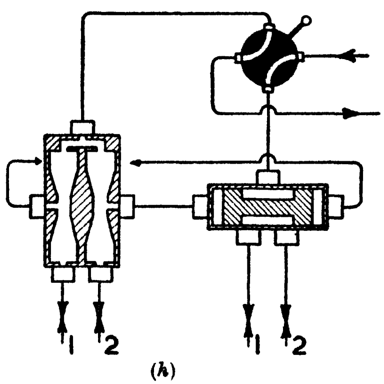
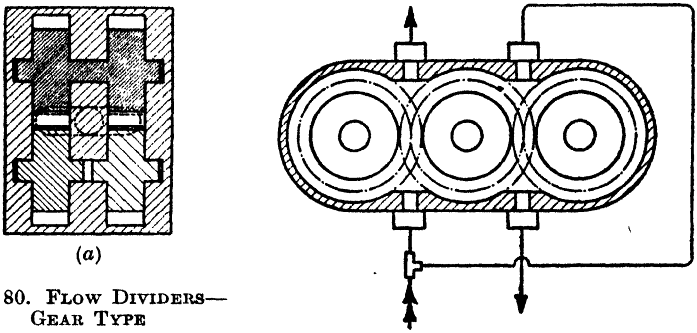


FIG. 79. FLOW DIVIDERS—DIVIDING TYPE

A type of flow divider working on a rather different principle is shown in Fig. 79 (*h*). This is similar to the by-pass type of Fig. 78 (*d*), and uses two venturi tubes, the reductions in pressure in which are proportional to the return flow through them, and this reduction in pressure is used to move a slide valve which divides the pump delivery equally between the two pressure lines of the jack. Since, however, the slide is in hydraulic balance, except for any difference in pressure on its ends, it may have a



(a)

(b)

FIG. 80. FLOW DIVIDERS—GEAR TYPE

tendency to oscillate as the venturi pressure varies. Centralizing springs on it may help to reduce this tendency. Obviously any ratio of flow can be obtained by adjusting the venturis.

There is a further type of flow divider using two mechanical pumps or motors joined together. The input of oil to the two pumps is common, and since the pumps are of equal size and rotate together on a common shaft, the delivery is bound to be the same within the limits of leakage or slip.

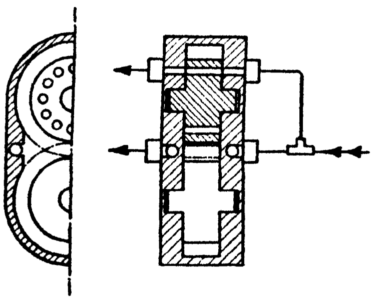


FIG. 81. FLOW DIVIDER—INTERRUPTER

Fig. 80 shows two known types, although obviously many other variations could be devised. Type (a) consists of two hydraulic gear pumps (strictly motors) coupled on a common shaft. Unequal division could be by means of different gears, and division into more than two parts could be effected by adding additional pumps. Type (b) is a simpler construction using only three gears; in this case the gears deliver partly to each outlet, but necessarily an equal amount owing to the symmetry of the unit.

Fig. 81 represents another type of interrupter which by-passes a percentage of the total flow of a system mechanically. It can be compared with a perforated disc which rotates in a beam of light; the total amount of light which passes through is determined by the area of the holes as a fraction of the annulus containing them. If the flow of fluid from a system is passed through a rotating disc containing a number of holes, the flow can be by-passed in proportion to the area of the holes, over the area not perforated. Although the disc could be driven by a separate hydraulic or pneumatic motor, a convenient construction is that shown in the diagram, which consists of a gear motor with the holes in one of the gears. The gears rotate idly in the fluid line; the inlet is connected to a further connection on the side of one gear which has a number of holes drilled across it. When the holes and ports coincide, the flow is by-passed to tank, etc. By using a large number of small holes the flow can be reasonably pulsation-free.

A further special mechanism for balancing the action of two jacks is shown in Fig. 82. The jacks are connected by a rack and pinion mechanism (or any equivalent device, such as a lever); unequal action will cause lateral motion of the pinion; this operates a balancing selector or dividing valve through a lever, arranged to distribute the supply of fluid so as to keep the pinion central.

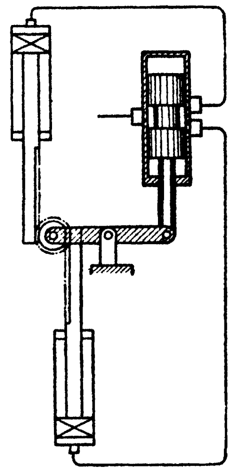


FIG. 82. FLOW BALANCER

8.3. Flow Controllers or Regulators. A flow controller is a valve or device to maintain a particular rate of flow or speed of operation or at least limit the maximum rate, and is the hydraulic equivalent of a governor.

Flow control can be obtained by passing the fluid through a venturi tube, the reduction of pressure in which is proportional to the fluid velocity and thus to the flow. The pressure is used to operate a spring loaded shut-off cock, which restricts the flow proportionately to the reduction of pressure and thus to the flow; a constant rate of flow within certain limits is thus maintained. Fig. 83 (*a*) illustrates the mechanism.

The principal type of flow regulator, however, consists of two restrictors, one fixed and the other variable, in such a manner as automatically to maintain a constant pressure across the fixed orifice, and thus a constancy of flow. The best known type consists of a reducing valve with its piston or diaphragm acted upon differentially by the two pressures on either side of an orifice in the output line from the reducing valve. Fig. 83 (*b*) shows the system diagrammatically. Since the pressures acting on the piston are the same as those across the restrictor and since the flow is proportional to the pressure drop, the reducing valve will open and shut to maintain a particular pressure drop and thus rate of flow. In the particular case of the output being at zero pressure (i.e., the regulator being fitted in the tank return valve), the connection to the underside of the piston can, of course, be omitted.

Fig. 83 (*c*) will be recognized as a slide type of reducing valve arranged similarly with a restricting orifice and the piston or slide subjected to differential pressure.

Fig. 83 (*d*) is a further and similar type where the first restrictor is fixed and the second one is variable, i.e., the converse of the previous type. The pressure drop across the orifice in the piston reacts against the spring, and opens and closes the outlet orifice to maintain the pressure drop, and thus the flow, constant irrespective of the outlet pressure.

By suitable arrangement of spring rates, these regulators can be used to limit maximum rates of flow only, remaining open at lower flows.

A somewhat similar type of unit is shown at (*e*); the flow from the pump to the selector passes through a shut-off cock, the tank return line passing through a restrictor and connected so as to shut off the cock as the flow increases.

Another flow regulating system is shown in Fig. 83 (*f*). In this the speed of action of a jack is adjusted by means of a restricting cock, whose degree of opening is determined by a spring deflected by a hydraulic dashpot or damper attached to the moving piston rod of the jack itself. Since the force produced in the dashpot is proportional to the speed of movement, this force can be used to balance the spring deflection and thus maintain a constant speed of operation of the jack. This system is of particular interest for a pneumatically operated jack, since the speed of motion with a compressible fluid depends on the resistance to motion as well as the supply of air and thus a normal flow controller would be ineffective.

A flow balancer or regulator which works on a balanced circuit similar to that shown in Fig. 79 (*a*) is illustrated in Fig. 83 (*g*). This diagram shows a system where the flow through a pilot circuit is used to act as a reference for various other circuits, maintaining the flows in each of the latter equal to that in the pilot. The particular case of two flow branches only will be observed to be similar to that of Fig. 79 (*a*) with the two outlet branches connected together and to the return line instead of to two jacks, etc. The similarity of this circuit with an electrical Wheatstone bridge will immediately be noted. In the general case shown in Fig. 83 (*g*), the three (or more) inlet restrictions are equal and therefore the pressure drops across them are a function of the flow in each branch. The variable outlet orifices are controlled by the pressure differences on the two sides of the pistons; the pistons will move

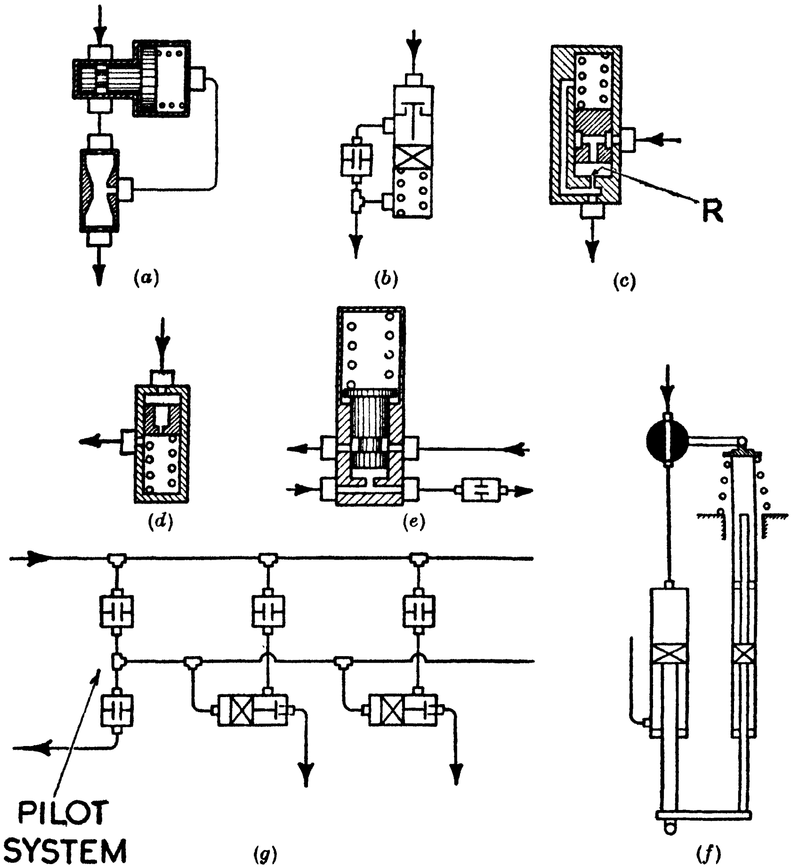


FIG. 83. FLOW CONTROL DEVICES

(a)–(e) Valves. (f) Control system. (g) Multi-branch system.

to vary the orifices until each piston is in balance and the flow down any branch is identical with that in the pilot system.

This system can be considered either as a multiple flow divider or as a flow controller or regulator since it has features of either type.

8.4. Transfer Valves. A transfer valve is a particular type of valve which transfers fluid from one side of a jack to the other under certain circumstances. In many double-acting jack systems the main load is in one direction, and on releasing the jack it "free-wheels" relatively rapidly to its other position, at least to some extent, and gets ahead of the flow of fluid. A well-known example of this is in aeroplane undercarriage retraction, where the main load is to retract, the unit extending most of the way by gravity. To save time under these circumstances a transfer valve is introduced which, when a suction is created in one side of a jack, opens and allows fluid from the other side of the jack,

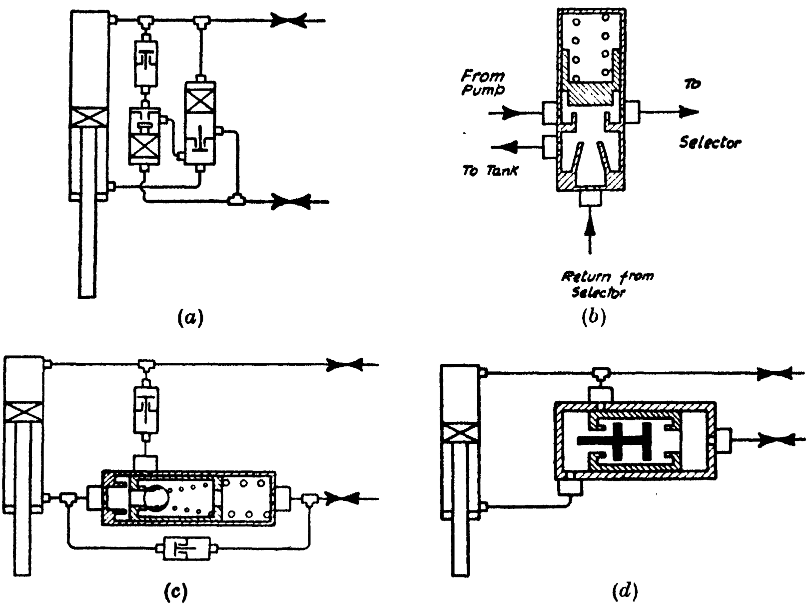
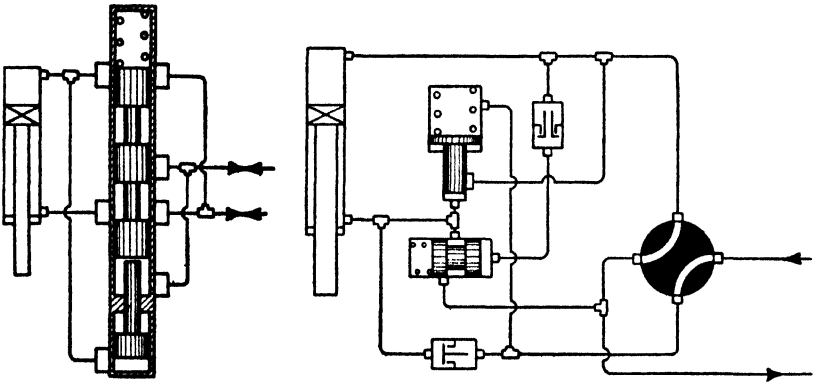


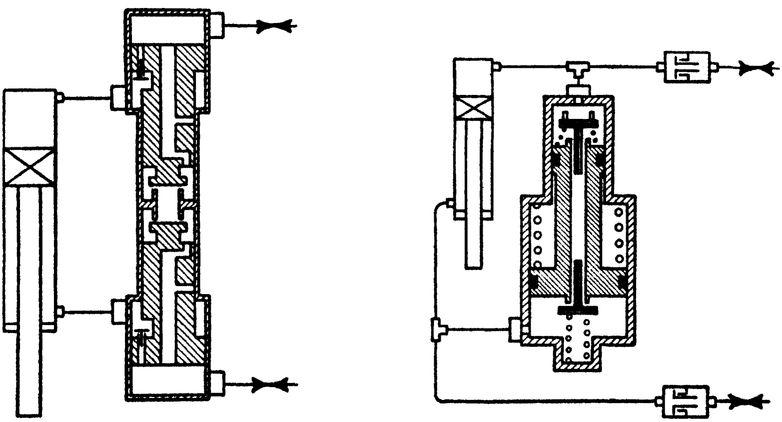
FIG. 84. TRANSFER VALVES

which is returning to tank, to by-pass to the low pressure end. When the resistance to motion increases, the valve closes again, isolating the connection between each end of the jack.

The conditions under which this type of valve must work are difficult, and there are many more bad valve solutions than good ones. Known solutions, not necessarily good ones, are shown in Fig. 84. Type (*a*) consists of two hydraulic locks and a non-return valve in a common body. Flow to close the jack can readily be followed from the selector through the valve of one lock to the jack, the second lock, connected to the upper end of the jack, being closed by the pressure. When the jack is released and extends rapidly under gravity, etc., the fluid in the annular volume cannot return to tank at first because the hydraulic lock valve is closed, but passes through the second lock and non-return valve to the upper end of the jack where a partial vacuum exists. When the initial extension of the jack stops, pressure builds up in the upper end of the jack to continue extension, and



(e) (f)



(g) (h)

FIG. 84 (*contd.*). TRANSFER VALVES

the valve in the hydraulic lock opens connecting the lower end of the jack to tank.

Type (*b*) differs in that it is connected in the pump and tank side of the selector. It depends on an abnormally high rate of return flow, due to the jack extending rapidly. The pressure energy of the returning fluid is converted to kinetic energy at the jet. The jet acts on the valve and lifts it, allowing considerable additional flow to be added to the pump output and to be passed to the other end of the jack. When the return flow slows down, the valve closes and is held shut by pump pressure.

Type (*c*) is a further valve installed close to the jack. In this case normal flow to the two ends of the jack is unrestricted, but return flow from the lower end of the jack must pass through a type of shuttle valve which uncovers a port connected to the upper end of the jack, thus effecting the necessary short circuit. The movement of the shuttle piston, however, is controlled by an internal blow-off valve set at a relatively low pressure; this means that the force tending to move the shuttle piston against its spring depends on the flow through the blow-off valve and the restriction offered by its throat. Thus the shuttle will move only when the flow from the jack is abnormally high, but will normally isolate the connection to the top of the jack.

Type (*d*) is somewhat similar to the previous one, except that the valve must be used with a differential area jack system (see Chapter 11). Normal flow to close the jack shuttles the piston over to the end of the cylinder, at the same time opening the valve in a sequence fashion to allow fluid to pass to the jack. The close fit of the shuttle piston isolates the other end of the jack. When the jack is released the internal valve moves across and closes the port in the piston, forcing the piston across to uncover the connection to the other end of the jack. The jack then operates on the differential area principle, and extends on the volume of the rod. It may be argued that this is not a true transfer valve, since there is no gain in performance over a normal differential jack system, except that the fluid has a shorter path.

Type (*e*) uses a lapped slide plunger as the transfer element. Normal flow to close the jack lifts the slide against the spring pressure and isolates the short-circuiting connection. When the jack extends, the spring holds the plunger in the original free position which connects both sides of the jack. When, however, pressure builds up in the jack, acting for the moment on the area of the rod, the small loose piston lifts and forces the slide over to the position closing the short circuit and allowing the jack to extend on the full area. This is a simple and straightforward unit which, since the slide is balanced, should work satisfactorily.

Type (*f*) is another slide type, if rather complicated; normal closure flow takes place in the usual way, the two diameter slide valve being closed by the greater force of the pressure on the larger area. When the jack is released, fluid cannot pass directly back from the annular volume to tank through the selector, due to the non-return valve in the line, and must therefore open the two diameter slide valve and pass to the top of the jack. When pressure builds up, the spring loaded slide moves and allows a direct return of fluid from the annular volume to the selector and tank. The one-way restriction valve used prevents the spring loaded slide from closing under sudden and unsustained reduction in extension pressure due to any reason.

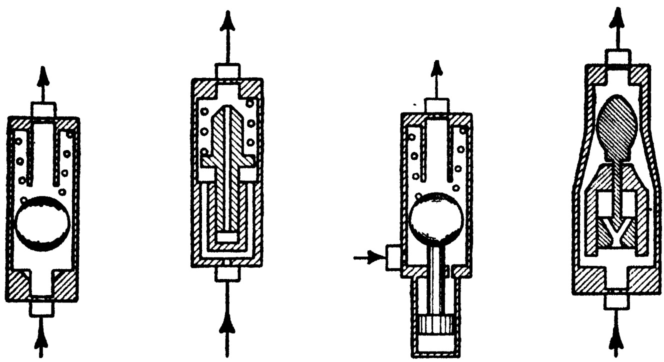
Type (*g*) is unusual in that it acts in both directions; it could be used on a jack with an "over-centre" action, tending to move ahead of the fluid supply for part of the travel in either direction. Normal flow passes through an internal non-return valve to one end of the jack, the return flow first of all having to lift the piston at its particular end of the valve until a port is exposed allowing free return. Should, however, the jack move rapidly ahead of the fluid supply, the lifting of the piston, already referred to, opens a valve in the centre of the body; fluid pressure will now open the second valve connected to the second piston and allow the fluid to by-pass. Valves of this type are fraught with difficulties in practical application, owing to the unbalance of the valves and pistons, and the fact that a more or less direct channel exists between the two sides of the jack should a valve leak.

The valve system shown in Fig. 84 (*h*) might be termed a "temporary" transfer valve since it does permit short-circuiting of the jack under certain circumstances although the object is to slow the motion, not speed it up. The system is designed to damp the motion (extension as drawn) of a pneumatic jack should one end of the jack not be under at least partial pressure when pressure is applied to the other end to move the jack. Under the influence of an external load the motion of the jack might be excessively rapid and the valve allows a quantity of air to be injected into the empty end to act as a cushion. The system operates as follows: normal closure of the jack takes place as usual, the lower valve in the larger piston of the valve device isolating the upper end of the jack. The same occurs should the jack be extending with pressure in the annular volume of the jack. If this back pressure is sufficient the piston assembly moves up owing to the difference in areas and positively closes the upper valve on to its seat. If the jack is extending without any back pressure in the annular volume, the double piston moves down, the upper valve is opened by its spring and a quantity of air is injected into the annular volume of the jack. This leaks away down the line past the restrictor but builds up until the piston assembly is again forced up against its spring and shuts off the upper valve mechanically. This valve now remains closed under the pressure acting at this end of the jack, the latter unit extending with an adequate cushion effect.

8.5. Hydraulic Fuses. A hydraulic fuse is a convenient term for a valve which closes off or isolates a line—

(i) when the instantaneous flow exceeds a particular velocity,
(ii) after a predetermined quantity of fluid has passed, or
(iii) after a predetermined time interval.

In some respects the isolation valve, referred to in Section 7.5, can be likened to a fuse since it closes when the pressure rises



(*a*) (*b*) (*c*) (*d*)

FIG. 85. HYDRAULIC FUSES
(*a*)–(*c*) = Flow operated. (*d*) = Quantity measuring.

to a predetermined amount, but the hydraulic fuse proper functions irrespective of pressure.

The earliest known valves of this type relied on the flow past a valve to close it if the flow became excessive. Fig. 85 (*a*) shows the elementary type. Normal flow passes the ball, held off its seat by a spring, to the system. When the flow increases, the loss between the ball and the body causes a pressure rise tending to move the ball against the spring, eventually allowing it to seat. The action is obviously very uncertain and will only be safe against flows well above normal. In addition, instantaneous variations of flow may cause premature closure.

Fig. 85 (*b*) shows a modern equivalent of this simple type, and differs only from the elementary type in that the valve is balanced and thus not sensitive to back pressure, etc.

The next step is to add a dashpot (type (*c*)), so that either the action of the ball is damped slightly and it is not so sensitive, or an actual time lag is introduced while the dashpot empties. In this case the time factor will largely depend on the forces exerted on the ball. The old-fashioned "waste-not" water taps use this mechanism, the valve being opened manually and closed by a spring against a dashpot (see page 134).

Type (*d*) is a version of type (*c*) actually applied to some aircraft systems. The restriction in the main line is streamlined, and the orifice in the dashpot device is carefully chosen to be "streamlined" and to have the same characteristics as the main restriction round the piston, i.e. having laminar flow characteristics. In this way the action of the device can be made to be independent of viscosity and rate of flow, since the greater the flow the more rapidly does the dashpot empty; the unit, therefore,

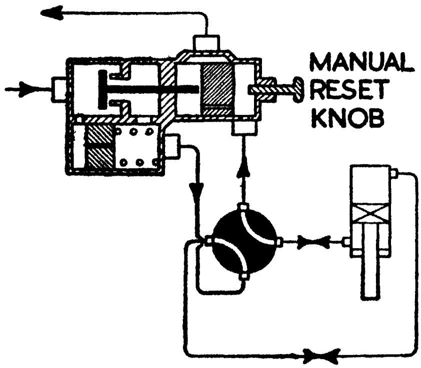


FIG. 86. HYDRAULIC FUSE—RETURN FLUID OPERATED

closes after a particular quantity of fluid has passed, more or less independently of the prevailing pressure. In the actual unit a light return spring is used, and sometimes a small non-return valve to facilitate the return filling of the dashpot.

A different type of fuse is shown in Fig. 86. This operates if either line to the jack is broken, and is installed before the selector. Fluid entering the fuse cannot reach the jack through the closed valve of a hydraulic lock. It, therefore, acts on a small piston which transmits partial motion to the jack, i.e. by a "sample" of fluid. This causes a corresponding action of the piston of the hydraulic lock, opening the pressure valve, and, in sequence, uncovering the tank return port. The hydraulic lock valve being open, normal flow can now continue. Had the tank return line or the pressure line been faulty, no unlocking signal would have been transmitted, and the main valve would have remained closed. So that the fuse will close if the return line should fail while the main valve is open, a small leakage past the lock piston allows it to close promptly. This means that it is insensitive to small rates of return flow, which in practice is satisfactory, and in fact desirable. Similarly, a small leak past the pressure signal piston will allow just sufficient fluid to pass to compensate for small leaks at the selector, etc. A manual opening or resetting device is provided.

Another type of fuse protects a pressure or feed line to a circuit when return flow from it, passing through the unit, ceases, as when a pipe breaks. This system, applicable only to continuously circulating circuits, consists simply of a hydraulic lock to block the pressure line, and kept open by the back pressure generated in passing the return flow through a restrictor or low pressure relief valve.

CHAPTER 9

MISCELLANEOUS VALVES AND COMPONENTS

9.1. Non-return Valves. It is hardly necessary to describe a non-return valve, since it obviously consists of any of the known lift valve elements already described, fitted in a body, and generally provided with a return spring. Fig. 87 illustrates the obvious solutions diagrammatically. Damped and other special types of non-return valves are known. A non-return valve is really a relief valve of very low pressure.

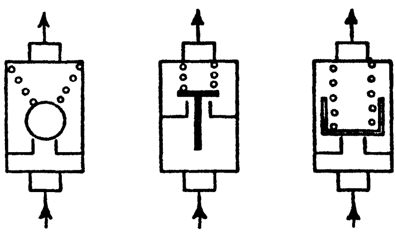


FIG. 87. NON-RETURN VALVES

9.2. Restriction Valves. A restriction valve is either fixed (i.e. a "restrictor") or else a non-return valve with a hole in it so that it restricts in one direction only.

In Fig. 88, (*a*) and (*b*) illustrate ordinary restriction valves symbolically. In actual systems, small drilled holes, adjustable needles, or sometimes the clearance between a screw and tapped hole, are used. A plain hole normally causes a pressure restriction proportional to the square of the flow—i.e. if the flow be doubled the loss increases four times. A restriction giving true linear connection between flow and pressure can be formed, for example, by passing the fluid back and forth between numerous flat plates mounted close together as in a radio condenser, but this is rather impracticable. Linear or "viscous" loss can be achieved by a tapered valve as shown in Fig. 88 (*c*), the spring tension on the valve being initially zero, or by a flat plate valve (*d*) which bends as it opens, so that the orifice increases progressively as the valve lifts or plate bends. Such valves are used in hydraulic dampers. The hydraulic equivalent of friction damping is obviously a relief valve.

A device which may be classified under the heading of a restrictor is that used on the Solex pneumatic measuring gauge. This uses a sensitive pressure gauge—actually a column of fluid—to measure the loss in pressure when air flows from a hole or holes in a measuring head (see Fig. 88 (*e*)) and between the narrow gap between the measuring head and the object being measured;

although the diagram illustrates a plug gauge measuring a bore, female gauges are obviously equally possible. With small clearances between the gauge and component the pressure is directly proportional to the linear gap between the parts, and thus the pressure measures the size of the component directly. In order

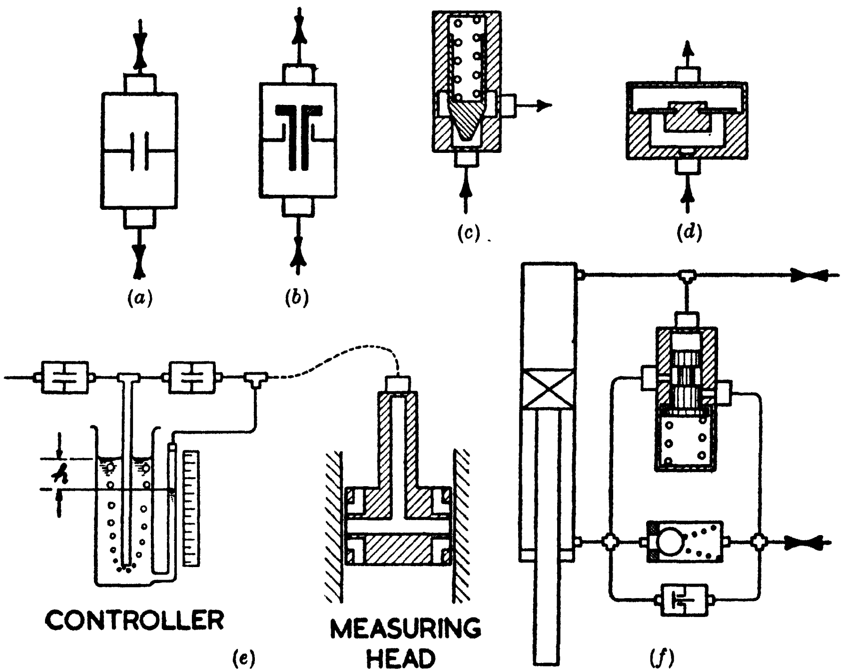


FIG. 88. RESTRICTION VALVES

(a) = Fixed orifice.
(b) = One way.
(c) = Linear type.
(d) = Linear type.
(e) = Device used on measuring intrument.
(f) = System to prevent rapid extension of jack.

to maintain a constant input pressure to the air jets, the air supply to these is tapped off from a supply which passes through a relief valve in the form of a column of water, so that provided the fluid level is maintained the pressure is constant. By using the same fluid level for the relief valve and water manometer, the effect of small fluid losses is ingeniously compensated for.

Fig. 88 (*f*) shows another restriction device diagrammatically; this is intended to prevent excessively rapid movement of a jack (e.g. under gravity) by imposing a relief valve in the return line to create a damping back pressure. A non-return valve by-passes this relief valve in the other direction; in order to eliminate the restriction when the pressure builds up in the other end of the jack, a slide by-pass cock is arranged to be opened against a spring when this pressure has built up to the required value.

9.3. Pressure Storage Devices. The following are designed to store fluid under pressure—

ACCUMULATORS. An accumulator is a device for storing hydraulic fluid under pressure ready for subsequent use.

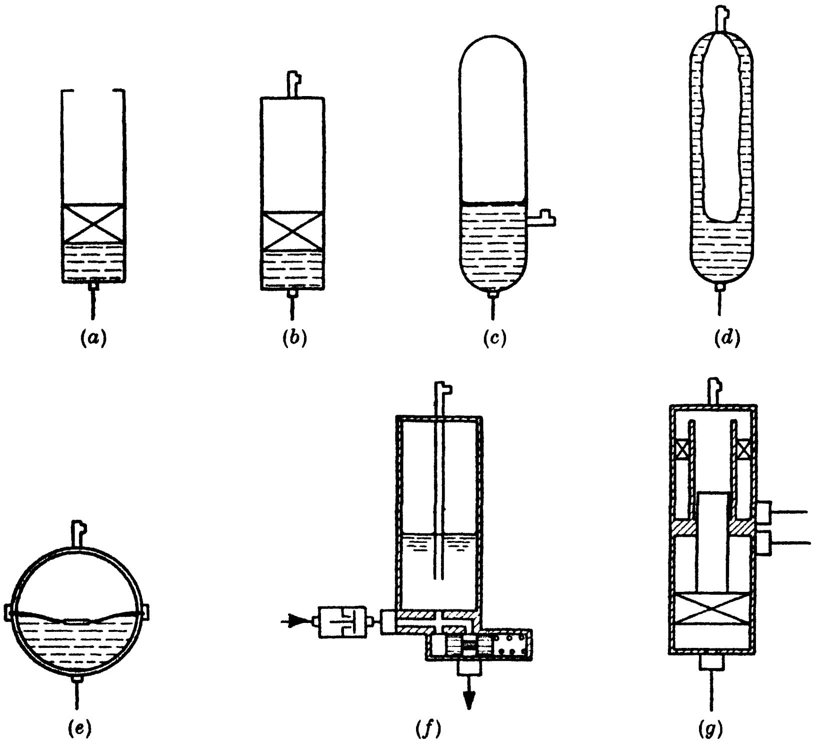


FIG. 89. ACCUMULATORS

(*a*) = Gravity type.
(*b*) = Separator piston type.
(*c*) = Pistonless type.
(*d*) = Flexible bag type.
(*e*) = Diaphragm type.
(*f*) = With pressure cut off.
(*g*) = Incorporating relay unit.

The earliest types used for many years on large hydraulic installations merely used the potential energy of a heavy mass in a cylinder, falling under the action of gravity (Fig. 89 (*a*)). Modern systems use compressed air or springs to replace the action of gravity (type (*b*)); a separator piston is used between air and oil. Plain bottles are used in some systems without the separator piston, when the system is a closed one. Since air dissolves in oil the separator piston is necessary to prevent continual loss of air if fresh oil is fed regularly into the cylinder. If the circuit between the accumulator and jack, for example, is closed, the oil becomes saturated and further air is not lost, making the use of a plain bottle possible (type (*c*)). Type (*d*) shows a unit where a flexible rubber bag is used to separate air and oil, and type (*e*) a spherical accumulator with a rubber diaphragm between air and oil. In either case the pressure load on the rubber is balanced, except when the accumulator is empty, when the rubber covering the oil port may require reinforcement.

Fig. 89 (*f*) is a power accumulator without a separator piston. Although still liable to suffer from loss of air by dissolution, it prevents the loss of air into the system by cutting off the outlet connection when the pressure drops to the minimum acceptable value; a spring-loaded slide valve does this, and provided the accumulator remains vertical the action is equivalent to the usual separator piston bottoming.

Fig. 89 (*g*) is a special accumulator incorporating a relay unit or "hydropump". The separator piston of the accumulator proper is annular and enables energy to be stored in the normal manner. The other piston, of the relay unit, enables oil under pressure from a selector to bè relayed, with a slight step up in pressure, to other oil on top of this piston; in the actual system for which this unit has been used the second lot of oil is of different quality and thus must be isolated. The piston rod of the relay projects into the air chamber of the accumulator and therefore the air pressure acts as a return spring for the relay piston.

Other examples of accumulators incorporating special valves, etc., are described in Chapter 7.

AIR BOTTLES. These hardly require description. Other things being equal, a spherical bottle is lighter than a cylindrical bottle, since the stress in a sphere is half that in a cylinder subjected to the same pressure. Cylindrical steel bottles are made by welding or by manipulating tube; light alloy bottles, both cylindrical and spherical, can be made by manipulation without seams. Wire tension winding is used to reinforce cylindrical bottles; the "autofrettage" process of hooping the bottle at regular intervals, and then tightening the hoops in place by over-expanding the cylinder, is interesting.

To mitigate energy dissipation when a bottle is burst, aircraft bottles have been made consisting of several small spheres connected together by narrow necks. If one sphere bursts the air expanding from the others is throttled by the narrow neck.

9.4. Shuttle Valves. A shuttle valve is one which permits either, but not both, of two pressure pipe lines to be used to actuate a jack, motor, etc. The commonest use is for the emergency operation of a jack, etc., if the main pipe line is broken; typical applications are on vehicle brake systems, aircraft powered controls, such as landing gear, and on dual control systems for braking, etc., on aircraft, trains, etc., when either of two driving positions may be in use.

The simplest form is shown in Fig. 90 (*a*), and consists of a ball which can seat on and block one or other of the arrival ports of the valve. Normal flow takes place from *A* to *B* and *B* to *A*, the ball sealing *C* as long as there is pressure at *A* or *B*. If the line at *A* breaks, pressure at *C* will first of all shuttle the ball over to seal *A*, and then the action from *C* to *B* takes place as before from *A* to *B*. Owing to the relatively uncertain action

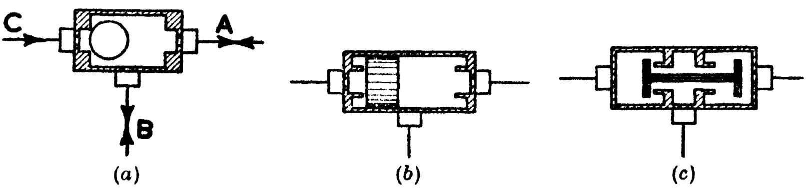


FIG. 90. SHUTTLE VALVES
(*a*) = Simple ball type. (*b*) = Piston type. (*c*) = Vacuum type.

of the ball in passing from one seat to the other, and the loss of fluid which would normally take place, simple valves of this type are usually only used with air.

Type (*b*) uses a fitted and relatively leak-proof piston, but acts exactly as before. The end ports may or may not be sealed by various known valves, but usually simple rubber or fibre end seals are used. If the action through one port predominates, a spring is usually added to keep the piston over one way, especially under suction loads, and to compensate for small leakage of fluid from the emergency line which might tend to move the piston over slowly and block the connection to the jack.

Type (*c*) is a similar type of unit used in a vacuum system where the action of the "piston" is reversed.

Fig. 91 (*a*) shows a feeding device which is strictly a type of shuttle valve. It is used in simple single-acting hydraulic systems, such as brakes, to keep the main line full of fluid. Normally the system is in communication with the tank so that any losses or excesses due to thermal expansion are made good. When the pump is operated the flow of fluid from it shuttles the piston over and seals off the tank connection; the pumping action then continues through the restriction in the piston, which must thus be carefully chosen to be small enough to allow the piston to shuttle, but not so small as to delay pumping. The return stroke is unrestricted owing to the internal valve. The action of such a valve can hardly be said to be positive, but appears to work in some cases.

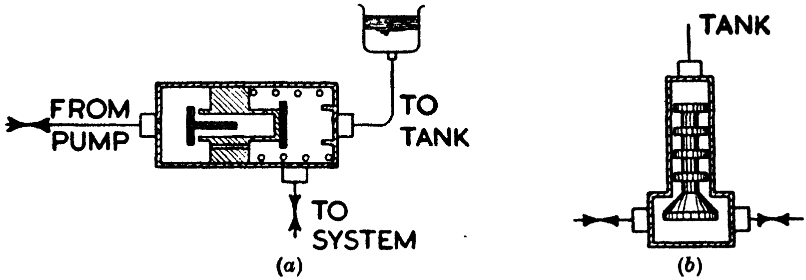


Fig. 91. Feeding Device

Type (*b*) is a simpler version of the same thing, using a poppet valve opened by gravity and closed by the rush of fluid past the crude "labyrinth gland" formed on the valve stem.

9.5. Air Bleed Valves and Steam Traps. An air bleed valve is one which allows the removal of air from a hydraulic system more or less automatically.

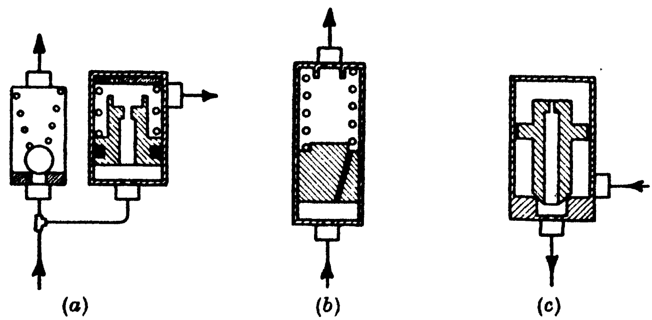
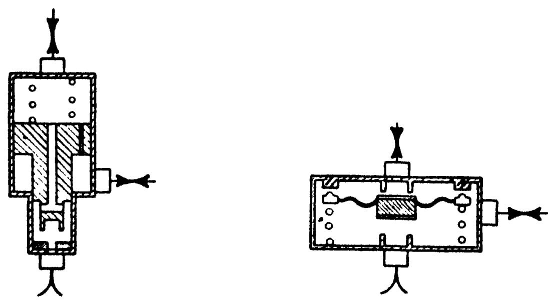


Fig. 92 (*a*), (*b*). Air Bleed Valves   Fig. 92 (*c*). Steam Trap

Fig. 92 (*a*) shows a valve which can be fitted to the delivery line of a pump to bleed the delivery of air. Normal flow passes through the lightly loaded relief valve to the system, the bleed valve proper being sealed off owing to pressure acting on its piston. If the pump starts to deliver air, the pressure at once drops and the piston spring moves the piston to open the outlet valve in it, allowing air to escape (preferably back to tank, since it will be mixed with oil froth, etc.). When the pump delivery again is free of air the loss of pressure of the oil (as opposed to the air) through the small restriction hole in the piston will again move it against its spring and seal the system off. The relief valve is necessary to ensure that there is sufficient back pressure to operate the piston, but it can replace the normal delivery non-return valve.

Similar types of valve are used to bleed household central heating systems, in this case omitting the relief valve as shown in Fig. 92 (*b*). This valve is really a hydraulic fuse (see page 124).

A steam trap is the converse of an air bleed valve, since it enables condensed water to escape from a steam system, but closes to prevent escape of steam. Fig. 92 (*c*) shows a typical valve,



(*a*) (*b*)

FIG. 93. QUICK RELEASE VALVES

and it will be seen at once to be similar to certain types of relief valve; normally the valve is shut by steam pressure, a small bleed being allowed past the flange and down the central hole. Should water condense, the pressure loss past the flange increases, lifts the valve, and allows water to escape; as soon as steam starts to escape the piston becomes balanced again and closes. By using a tapered sleeve between the flange and body, the restricting orifice can be adjusted by moving the sleeve up and down and thus the sensitivity of the trap altered as required.

9.6. Quick Release Valves. A quick release valve is one which will allow the rapid escape of air to atmosphere at some local point, usually close to a jack, etc., so that the air does not have to return all the way to the selector. Similar valves are known for vacuum systems, the direction of flow being reversed. In a sense a valve of this type can be considered as a special case of a relay selector. A similar object is attained with the systems described under "Flow Accelerators" in Chapter 17.

Fig. 93 (*a*) and (*b*) show known types of quick release valve. Type (*a*) has a piston, normally spring-loaded to close an escape port. When the main selector is exhausted the air flowing from the pipe line causes a difference in pressure on the piston owing to the restricted passage through the hole in it and the higher pressure in the jack or receiver line lifts the piston, allowing air to escape to atmosphere. As illustrated, the additional central port in the piston helps to vent the selector line as soon as the valve has lifted, the smaller diameter of the piston masking the escape of air from the jack initially; this masking diameter acts as a low pressure slide type relief valve to maintain the small pressure under the piston necessary to lift it against the spring. The central passage also allows the piston to be balanced, the pressure on each side acting on the annular area only.

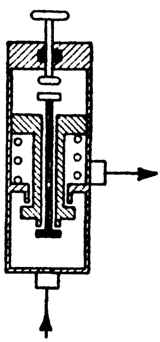


Fig. 94. Valve Closing after Time Lag

A simpler and perhaps more effective solution is shown at (*b*). Air entering the valve, opens a large diameter diaphragm, which acts as a low pressure relief valve and involves little loss of pressure. Before the outer valve seat actually lifts, the pressure on the centre of the diaphragm pushes down the central plug; this seats on the exhaust seat and cuts off the venting of the jack. When the air in the main pressure line is allowed to exhaust (or strictly speaking, when the flow to the jack ceases, before the pressure line is exhausted) the large valve seats; the pressure in the jack line now acts on the diaphragm in the reverse direction and lifts the venting valve allowing direct escape of air to the atmosphere.

9.7. Valves Closing after Time Lag. There is a type of valve —strictly a selector or on-off cock—which, once opened, closes after an interval of time. Many of these valves have been designed for normal household water-supply systems to prevent waste of water. These have a certain interest in pressure systems, and in some respects can be considered as a type of hydraulic fuse (see Chapter 8.5).

Fig. 94 shows a design of more interest than the simplest types, but typical in principle. When the knob is pushed, the central pilot valve is opened allowing fluid pressure to enter the upper chamber, and unbalance the piston, which then opens the main valve. The valve then closes gradually as the spring and pressure force acting on the unbalanced head of the valve forces the fluid from the upper chamber past its close fitting diameter or a suitable restriction hole; meanwhile, since the pilot valve is closed the pressure supply cannot again enter the upper volume.

9.8. Oil Separators. An oil separator is a bottle which enables some of the oil in suspension in the air of a pneumatic system, coming from the compressor, to be separated out and drained away. One system is to pass it through a large metal filter, the oil condensing or collecting on the outside of the filter and eventually dripping down into a sump, which can be emptied from time to time.

Another system often used at the same time as the other, is to admit the air to the separator by a tangential entry, so that,

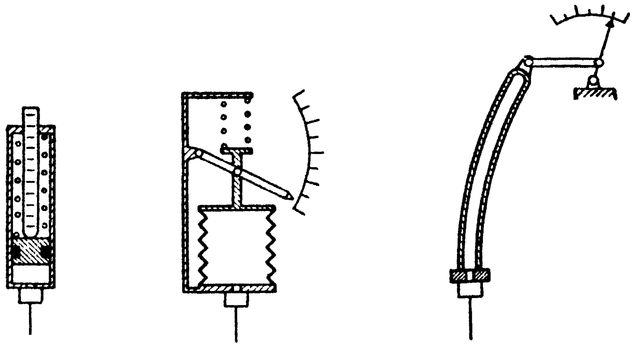


FIG. 95. PRESSURE GAUGE
(*a*) = Spring type. (*b*) = Bellows type. (*c*) = Bourdon type.

as it expands, it whirls and the oil is separated out by centrifugal force, draining into the sump as before.

9.9. Pressure Gauges. Pressure gauges are usually of two types, the piston, bellows or diaphragm type, and the Bourdon tube type. Fig. 95 illustrates these diagrammatically. Type (*a*) is the mechanism of a normal tyre gauge. Pressure lifts a piston against a spring with a slight error due to gland friction, and pushes out a loose rod or indicating device, suitably calibrated. If the spring rate is constant (which it should be) the deflection of the spring is proportional to pressure. To eliminate gland errors, dead weight pressure gauges are used, for example, to calibrate normal types. These have closely fitting, but glandless, pistons loaded by dead weight in place of a spring.

Type (*b*) is equivalent to type (*a*), but uses a bellows or a diaphragm. These are usual for low pressure applications on barometers, etc.

Type (*c*) is a much simplified representation of a Bourdon tube gauge of the classic type. A curved tube tends to straighten under pressure since it has its maximum internal volume when straight. If the motion of its end is coupled to a lever, an accurate representation of the pressure is given. In practice the tube is coiled to increase its length, and the motion of the pointer multiplied by a gear train.

Fig. 96 shows a useful gauge relay or protecting device which can be fitted in the gauge connecting line. Should the gauge fail, the fluid in the hydraulic system will not be lost.

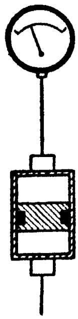


FIG. 96. PRESSURE GAUGE RELAY

9.10. Flow Meters. A flow meter is a device for indicating the rate of flow of liquid or gas; two common types are known. The first makes use of a fluid motor through which all the fluid must pass and thus by connecting the motor to a counter the quantity of fluid passing can be measured within the limits of accuracy of displacement of the motor—usually very high. This type is obviously more suitable for liquids than gases since the effects of compressibility of liquids can be ignored.

The second type depends on the flow velocity, the best known type being a Venturi meter which records velocity of flow in terms of a pressure drop in the throat of the venturi. If the orifice coefficient and dimensions of the Venturi are known, the quantity of fluid passing can be calculated from its velocity. Another well-known instrument gives a direct reading on a scale as a small, loose fitting float is blown up a tapered glass tube, the gap increasing as the float rises. The calibration of course, is only correct for a particular fluid (generally a gas) and at a particular temperature, etc.

CHAPTER 10

PIPE FITTINGS

10.1. Pipe Couplings. Pipe couplings exist in far too many different forms to be examined in detail here, but nearly all are minor variations (not necessarily improvements) on the main types shown in Fig. 97. Type (*a*) is the classic soldered or brazed cone connection, usually with a 60° included angle cone. Type (*b*) is an expansion joint where the end of the pipe is flared out

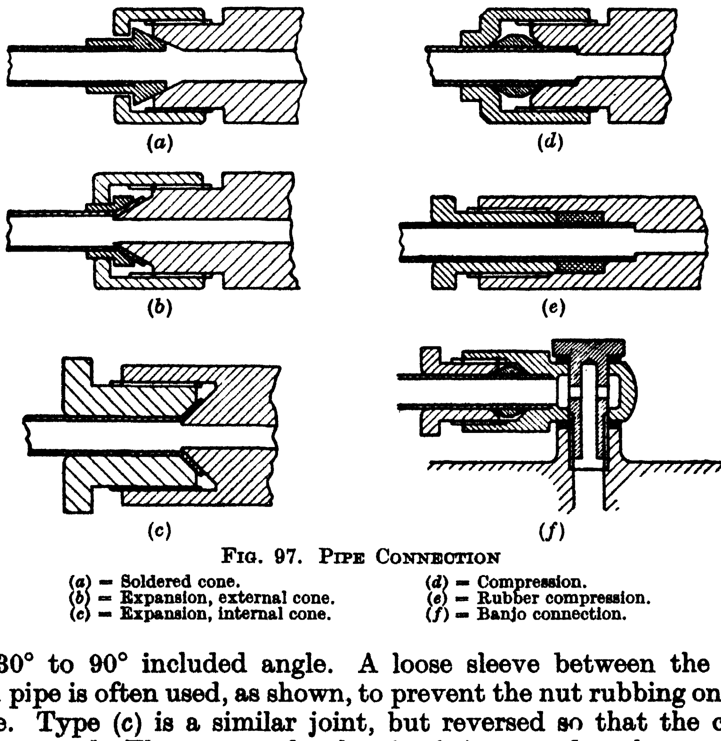


FIG. 97. PIPE CONNECTION

(*a*) = Soldered cone.
(*b*) = Expansion, external cone.
(*c*) = Expansion, internal cone.
(*d*) = Compression.
(*e*) = Rubber compression.
(*f*) = Banjo connection.

at 30° to 90° included angle. A loose sleeve between the nut and pipe is often used, as shown, to prevent the nut rubbing on the pipe. Type (*c*) is a similar joint, but reversed so that the cone is protected. The support for the pipe is improved at the expense of cost, since the cone is obviously difficult to make at the bottom of a hole.

Type (*d*) is a compression joint, using an olive (usually in brass) whose edges are squeezed down into the pipe by the joint being tightened. Often the wedging action is augmented by using a 45° chamfer on the nut and body, and a 30° half angle on the olive; or by a radius on the nut and body and a cone on the olive. In fact there are numerous proprietary joints of this type and even more patents for olives with special shapes to improve the grip, but few are worth the extra manufacturing cost. One interesting variation, however, is to use a rubber "olive" as shown diagrammatically in (*e*). Tightening the joint forces the rubber to swage the pipe slightly inwards.

The only other main type of connection worth mentioning is the banjo or orientable type as shown at (*f*).

10.2. Flexible Pipes. Flexible pipes or hose are a necessary adjunct to any pressure system. The pipe itself is usually made from natural or synthetic rubber reinforced with braided cotton or steel wire layers. Cotton-reinforced hose is available, for example, with working pressures of about 1 000 lb/sq in. in a bore size of $\frac{1}{4}$ in.; steel-braided hose of this bore is available for working pressures of 5 000 lb sq/in. or more. A $\frac{1}{2}$ in. bore hose with cotton reinforcement will operate at about 500 lb/sq in., and in steel at about 1 500 lb/sq in. The strength depends on the number of layers of reinforcing material. Although simple pipes have been, and are still made by clamping the hose to an inner metal pipe (e.g. bicycle or car tyre pump hose), the actual assemblies used on a pressure actuating system is likely to be of more specialized design.

The most important feature to the user is the type of end fitting adopted, since the hose itself is of proved quality and reliability and, in these respects, is often better than the end fittings. The infiltration of oil down the end fitting and between the layers of hose is a common fault caused by poor end fittings. This infiltration causes blisters to appear on the hose which when pricked are not necessarily full of anything else than gas or oil vapour, but are a sign that the life of the hose will not be long. Various types of ends are illustrated in Fig. 98 (*a*) to (*g*).

Type (*a*). In this case the end fitting is in brass, and in one piece. The brass (or brass-plated steel) is an advantage, since the rubber is self-bonding to it.

The brass also facilitates the difficult operation of trepanning and hollow-tapping the internal bore. With this connection the oil has to pass along in two directions to leak. Torsion and tension loads are distributed over the whole surface and there is no tendency to shear the outer and inner layers of rubber.

The inside diameter of the outer casing of the end fitting has small flat teeth in order only to grip and not break the rubber, the main adhesion being by friction.

Type (*b*). This is similar to type (*a*) except that coarse teeth are added to facilitate bonding and to grip the core positively; the cover has to be ground away at the end to expose the core and allow insertion of the hose before swaging. This mutilates

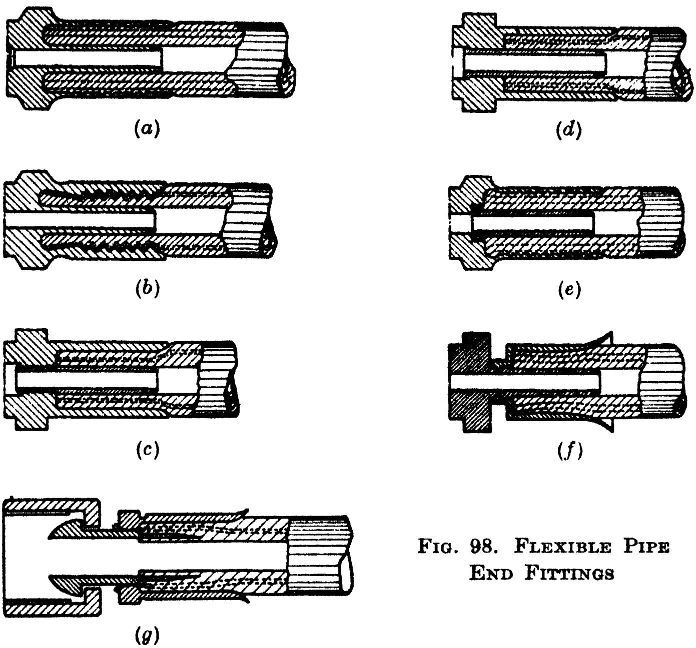


FIG. 98. FLEXIBLE PIPE END FITTINGS

the end of the hose and increases the risk of leakage. There is also a single effective length of seal.

Type (*c*). This is also similar to type (*a*), but the central pipe is screwed and soldered in place. If the soldering is defective the oil has access to the core of the rubber and may infiltrate. It has also only one length to travel to leak. Torsion loads are not so adequately dealt with and may shear the solder. It is an attempt to produce a cheaper substitute for the solid type; the use of furnace copper brazing, however, overcomes the objection to solder.

Type (*d*) is very similar to type (*c*) and suffers from the same defects. The internal nipple is not soldered but clamped in place, and the chances of infiltration are relatively great.

Type (*e*). This is better than many other substitutes for the first type, since the central brass nipple is peaned in place after insertion in a pocket inside the end fitting, but although the chances of leakage or movement of the separate nipple are reduced, a solid fitting is still better.

Type (*f*). This differs from the others in that the end nut is part of the central nipple and torsion of the ends is transmitted to the hose by the small diameter of this nipple. The outer casing of the fitting serves to swage down the hose and it is rolled over into a groove in the inner nipple to prevent end shift, but is not fully effective in torsion. A single length against leakage is available, but this is made less effective by adding a coarse pitch

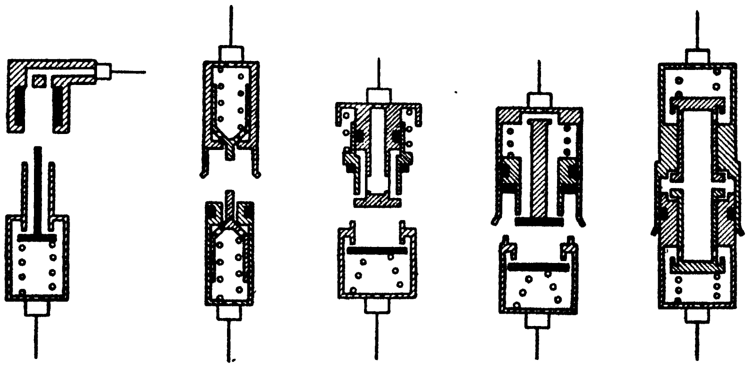


FIG. 99. DISCONNECTING COUPLING
(*a*) = Common type. (*b*)–(*e*) = Self-sealing.

thread to the outside diameter of the inner nipple; the fitting is at best a substitute for the solid fitting.

There are variations of type (*f*) where the outer sleeve is not swaged, but the inner nipple is screwed into the inside of the hose after the outer ferrule has been slipped over the hose. These types are excellent for low and medium pressure, and can in fact be dismantled and re-made on new hose with simple tools.

Type (*g*) grips the steel braid between the two parts of the end ferrule and therefore is mechanically very strong. The seal against leakage is effected by slitting the rubber lining and inserting the sharp edge of the nipple into it. To prevent the rubber lip lifting under reverse flow, a thin lining tube (not shown in the sketch) is used in the bore.

10.3. Disconnectable and Self-sealing Couplings. A useful form of pipe coupling is one which is readily detachable and automatically seals one or both of the uncoupled pipes against loss of fluid. Fig. 99 (*a*) shows the commonest and obvious type which seals only one line; this is the mechanism of the well-known motor tyre pump connector, or the coach-to-coach steam and air line couplings on railway trains. The essentials are a non-return valve, mechanically opened by the action of connecting one pipe and some form of seal between the two members. Various forms of clamp are known, including quick-acting toggle or compression devices to save the time of screwing the parts together.

Type (*b*) is the basic type of self-sealing coupling which seals both lines. It consists of two valves, some means of opening both when the coupling is made, and a seal between the two bodies. In the construction as shown diagrammatically, when the coupling is made, a small volume of air will be introduced, or oil lost when the connection is undone. To avoid this, various couplings have been designed which disconnect more or less dry. Type (*c*) shows one solution which uses one lift valve with a flat face exposed to the other member, and a reversed valve whose seat on a sleeve moves away when the joint is made. In order to balance the latter to close, the sleeve sealing diameter is made larger than the seat. Types (*d*) and (*e*) are very similar to type (*c*).

Most types of self-sealing coupling are streamlined so that the pressure losses through the valves, ports, etc., are at a minimum. Practical designs, therefore, differ considerably from the diagrammatic simplification of types (*c*), (*d*) and (*e*).

PART II: PRESSURE SYSTEMS

CHAPTER 11

ELEMENTARY POWER SYSTEMS

11.1. Single-acting Jack Systems. The methods of operating a single-acting jack are too obvious and well known to require much description.

Fig. 100 (*a*) shows the standard method of operating a spring return jack with a 3-way selector. Circuit (*b*) is equivalent, except that a hydraulic accumulator replaces the spring. As the accumulator circuit is closed, a piston separating oil and air is unnecessary (see page 129).

Circuit (*c*) is a useful method of controlling a single-acting jack by means of a 2-way selector (cock), in conjunction with a permanent leak to tank. This device is used in many cases to simplify the selector element, and is seen in several types of cut-out, etc.; there is a small continuous leak, however, which is usually acceptable, while the size of this leak governs the return speed.

All of these circuits can be connected to either end of the jack, etc.

Circuit (*d*) is an ingenious method of using an accumulator in a sealed differential area system (see below). The whole system is under accumulator pressure, and when the pump by-pass cock is open, the jack extends on the effective area of the jack piston rod. When the cock is closed, the pump can suck oil from the top of the jack or accumulator and deliver it into the annular area of the jack, closing this unit. The surplus fluid due to the volume of the rod is forced into the accumulator and raises its pressure, ready for subsequent discharge when the cock is again opened. In this circuit the accumulator must be connected as shown, the reverse connections on the jack not being possible.

A useful method of inching a jack in small steps is shown in the system illustrated in Fig. 100 (*e*) in much simplified form. Pressure is applied by a selector to a relay unit—not necessarily pressure-intensifying as shown—which acts as a simple single stroke pump to move the jack a small amount, depending on its delivery volume. The pump delivers through a non-return valve, and is provided with some sort of feeding reservoir (see Chapter 12 for information on this point). To enable the jack to return,

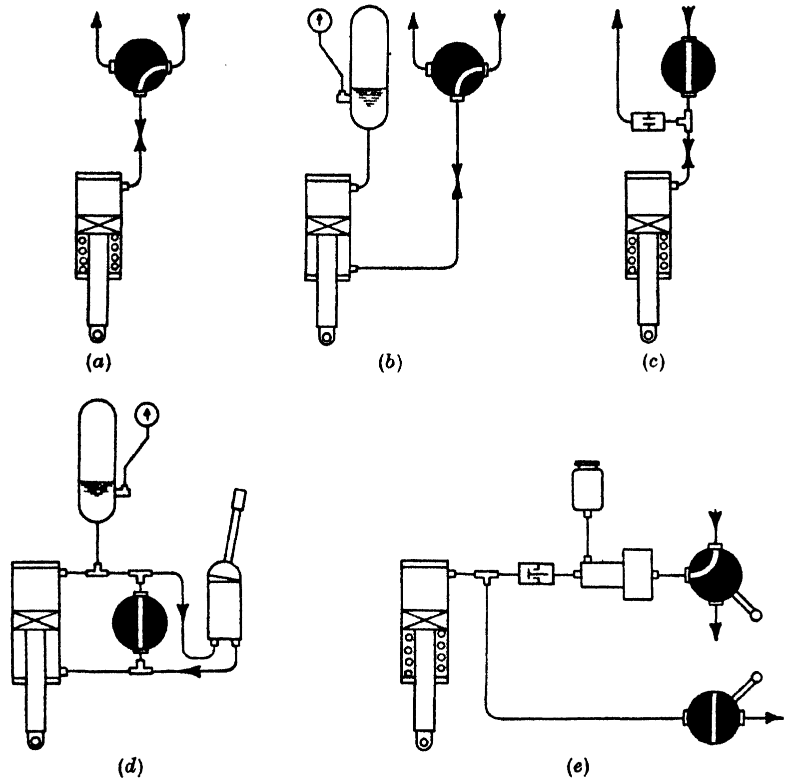


Fig. 100. Single-acting Jack Systems

(a) = Spring return.
(b) = Accumulator return.
(c) = Permanent leak.
(d) = Differential area.
(e) = Method of inching a jack in small steps.

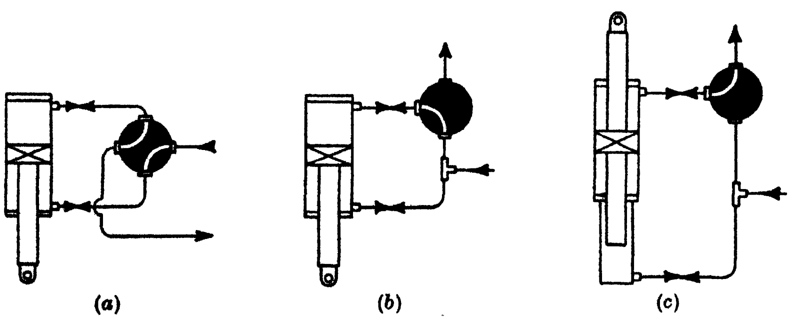
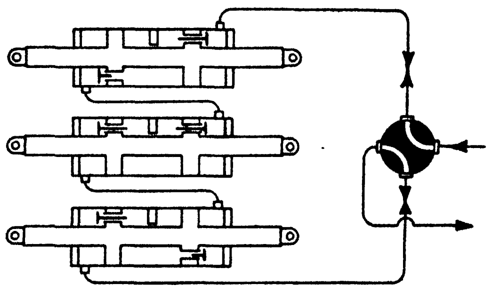


Fig. 101. Double-acting Jack Systems

(a) = Normal. (b) = Differential area. (c) = Reversed differential area.

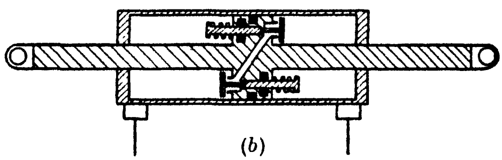
it can be released by a cock as shown or by a second relay system which "inches" the jack backwards.

11.2. Double-acting Jack Systems. Fig. 101 (*a*) illustrates the well-known double-acting jack circuit using a 4-way selector. Variations of this, using multi-travel jacks, etc., have already been discussed in Chapter 5. Circuit (*b*) is the "differential area" circuit. The pump pressure is applied to the annular area of the jack continuously, and the other end of the jack is either



(*a*)

FIG. 102 (*a*). JACKS OPERATING IN SERIES



(*b*)

FIG. 102 (*b*). METHOD OF MOUNTING SYNCHRONIZING VALVES IN THE JACK PISTON

connected to pressure as well, or vented to tank by a 3-way selector; thus the jack operates to extend on the effective area of the piston rod, the annular volume being by-passed from one side to the other.

This circuit is particularly useful because—

1. It uses a 3-way instead of a 4-way selector.
2. The effective volumes and areas in each direction can be varied at will without the limitations of a normal double-acting jack, and can easily be made equal.

Circuit (*c*) is a differential system which by virtue of the more complicated jack extends under pressure on one volume, and closes with pressure in both volumes, i.e. the converse of circuit (*b*).

Fig. 102 (*a*) illustrates double-acting jacks mounted in series. It will be seen that the fluid acts directly on one piston, moving the remaining jack pistons owing to the presence of trapped fluid columns between them. When the pistons reach the ends of their travels, synchronizing valves are opened which make sure that each jack reaches its stop and that any excess or lack of fluid in one particular line is made good. Although when the valve is open the piston is short-circuited and thus can exert no force, what will happen, if there is any reaction on the jack rod, is that the valve will open just long enough to effect synchronization, and then the piston will stay in a position with the valve just shut. Since each jack but the last has operating pressure on both sides, the pressure in each jack will not be the same. Suppose each jack is lifting a weight of 1 000 lb, and that the annular section of the pistons is 1 sq in. Then the operating pressures are as follows—

Last jack: 1 000 lb $\div$ 1 sq in. = 1 000 lb/sq in.
Last but 1 jack: 1 000 lb + (1 000 lb/sq in. $\times$ 1 sq in.) $\div$ 1 = 2 000 lb/sq in.
Last but 2 jack: 1 000 lb + (2 000 lb/sq in. $\times$ 1 sq in.) $\div$ 1 = 3 000 lb sq/in.

Obviously, therefore, the operating load on each jack is the pressure multiplied by the jack area and divided by the number of jacks, the sum of the loads being equal to the total pressure on one jack area.

Fig. 102 (*b*) illustrates another way of mounting the valves in the piston head, using spring-balanced valves.

This series system can be used with accumulator return, as for a simple system. For other aspects of it see Chapter 12—Remote Control Systems.

11.3. Open Centre Systems. The open centre system has already been referred to in Chapter 5, where the necessary selectors are described. The feature of this system is that the selector allows the pump pressure to idle to tank in an intermediate or central position of the selector, usually obtained automatically by the selector returning to neutral when the pressure rises at the end of the jack travel; otherwise a relief valve must be used in case the selector is not returned at once to the central position manually.

To enable this system to be applied to several jack circuits, the selectors are connected in series as shown in Fig. 103. When any one selector is operated the open centre is closed, but more than one jack circuit cannot be operated at once, the priority being determined by the series connection of the jacks. The jacks can be locked in a particular position by the selector, or by using hydraulic locks.

11.4. Use of Power Accumulators. The action of a power accumulator on a hydraulic system is obvious, and requires few comments. The normal method of connecting is obviously as shown in Fig. 104 (*a*). The complication introduced in practice is that unless the accumulator is of sufficient capacity to operate the jack fully, the action may continue until the accumulator is empty, and then the pump may re-charge the accumulator partly instead of completing the motion of the jack. This depends on the specification of the accumulator and the requirements of the jack throughout its travel, but often requires detailed investigation.

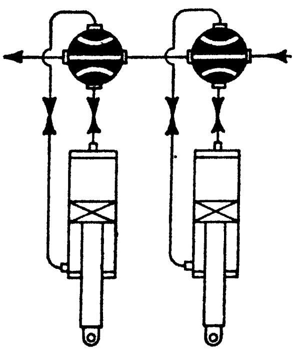


FIG. 103. THE OPEN CENTRE SYSTEM

It would be possible to overcome this difficulty by adding a shut-off cock to the line connecting the pump to the accumulator, so arranged to be closed automatically by the descent of the piston, and only opened to allow re-charging by the action of the jack reaching the end of its travel; this would mean a mechanical connection between jack and cock, or an automatic system similar to the automatically centralizing selector operated by the rise of pressure at the end of the jack travel. (These are hardly worth the complication in the average system.)

Fig. 104 (*b*) shows an accumulator connected up so that its piston when it falls moves a slide cock to isolate the pressure connection; the latter is opened by a piston only when the system pressure lifts a relief valve and fluid acts on the piston. A permanent leak to tank allows the piston to operate in the other direction to allow the slide to close. This system uses the mechanism of the centralizing selector of Fig. 49 (*b*).

Another device is shown at (*c*) (Fig. 104). In this case the accumulator incorporates a valve which prevents the accumulator from being discharged until it is fully charged. When the piston drops it pushes closed a valve normally held open by a spring; this valve when closed isolates the selector, system, etc., from the pump, which however can lift the piston, and when the accumulator is full the piston lifts the valve off its seat again, the pump pressure keeping it closed until this moment. The accumulator can now discharge again normally.

Sometimes two accumulators can be used to advantage, as in the so-called two-stage braking systems referred to later in

Chapter 12.4. A power accumulator system of this type is shown in Fig. 104 (*d*). A high pressure accumulator is filled from the

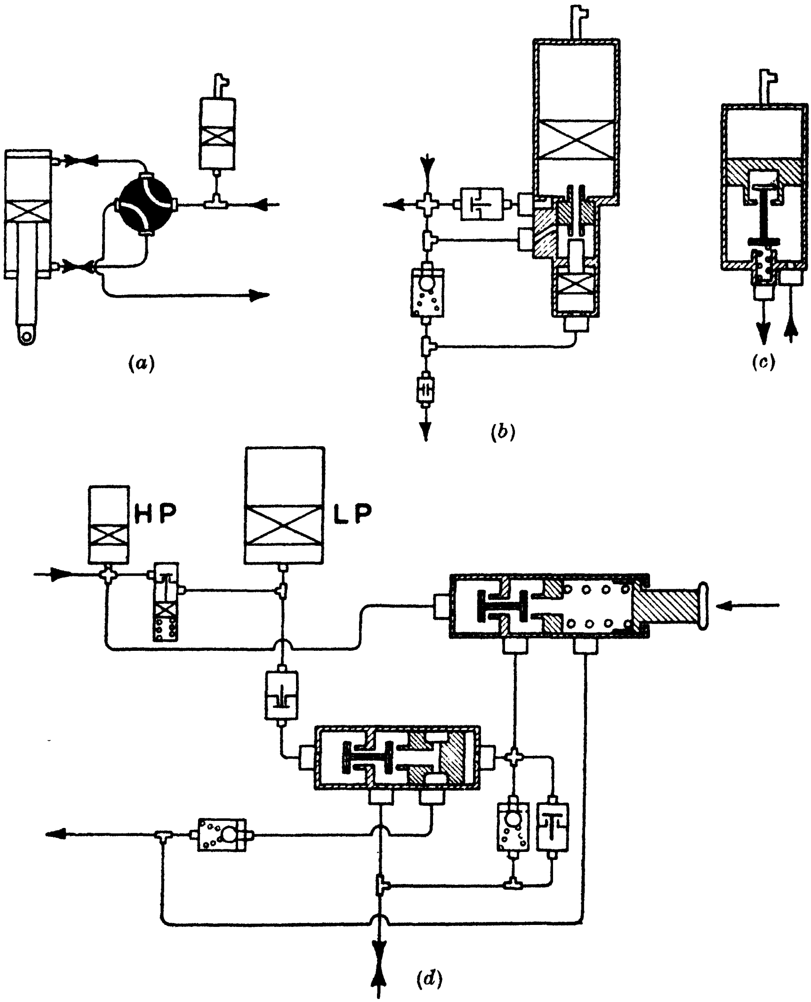


FIG. 104. POWER ACCUMULATOR

(*a*) = Applied to a jack system.
(*b*) = Fitted with isolation valve system.
(*c*) = With safety device.
(*d*) = Two-stage system.

pump line, and, through a reducing valve, a low pressure large volume unit is kept at a lower pressure. A normal manually controlled power brake valve controls the admission of oil direct from the high pressure accumulator to a second relay operated brake valve, and also through a relief valve, to the brake units themselves. The relay brake valve in turn admits oil from the low pressure accumulator to the brake system, the object being to ensure rapid filling of the latter, advance of the brake shoes, etc. The object of the relief valve just mentioned is to ensure that there is sufficient pressure to operate the relay, and to restrict discharge of the high pressure system until the brake pressure rises. Thereafter operation continues normally on the high pressure control valve; discharge of high pressure fluid back to the low pressure accumulator is prevented by a non-return valve, and escape through the tank return line of the relay brake valve is not possible, since the piston of the latter is balanced to the tank-closed position due to a difference in pressure on its two faces arising from the relief valve. When the brakes are released, the action is normal except that a very low pressure relief valve can be used, installed either as shown or controlling both return lines, depending on the physical lay-out involved, to prevent fluid draining from the brake system.

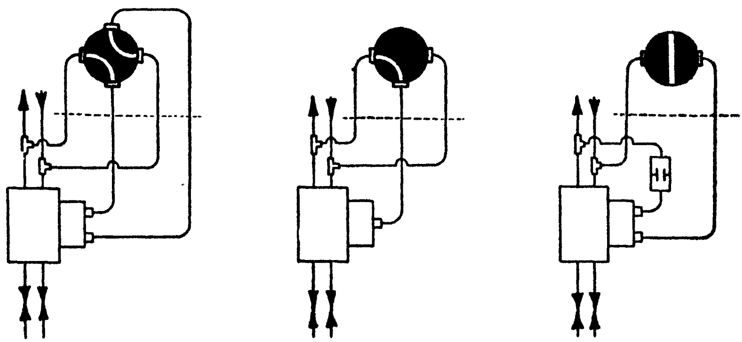
The advantage of such a system is that faster action can be obtained by the use of relays, the location of the relay and low pressure accumulator being close to the brake units; there need not necessarily be a difference of pressure in the two accumulators, but it is unnecessary to use the extra bulk of a second high pressure one when a low pressure one will suffice.

11.5. Relay Systems. A relay system is one in which a subsidiary circuit operates the main selector, which in turn controls the principal circuit. Such systems are useful because it is often possible to economize in the length of pipe runs of the main system by using small pipes or electric wires to signal to the main selectors placed so as to shorten the power circuit. In this respect an exact parallel can be drawn with the use of electrical relays to avoid extensive heavy current wiring.

The relay selectors themselves have already been described in Chapter 5. The special requirements of electro-hydraulic or -pneumatic systems are described in Chapter 16. Any normal remote control systems described in Chapter 13, or even servo systems as described in Chapter 15, can and have been used for the remote control of the selector, and the combinations are more or less obvious.

Fig. 105 shows the pipe runs necessary between pilot and relay selectors. Type (*a*) uses a 4-way pilot selector, four pipe runs, and a double-acting mechanism to operate the main relay selector. Type (*b*) uses a 3-way pilot selector, three pipe runs, and either a single-acting mechanism with spring return, or a differentially connected one on the main selector. Another known method uses a pilot selector of the type illustrated in Fig. 70 (*q*), giving half or full pressure in the single pressure line to the relay selector to give the two extreme positions of the latter. Type (*c*), however, by using a permanent leak system can use a cock and only two pipe runs.

The illustration refers to hydraulic systems where the fluid from the pilot selector must be returned to tank. In the case of a pneumatic system, however, one pipe line can be dispensed with on types (*a*) and (*b*). At the expense of a permanent loss



(*a*) (*b*) (*c*)

FIG. 105. RELAY SYSTEM

(*a*) = 4-way pilot. (*b*) = 3-way pilot. (*c*) = 2-way pilot.

of air, type (*c*) can be carried out pneumatically with a single pipe, the control being effected by opening or closing the small leak. Remote control of a selector by a hydraulic system with a single pipe is possible by means of one of the remote controls of the types described in Chapter 13, but not by pressure control in the sense of the other solutions.

Special problems arise in relay systems affecting the selector; for example, does the selector require continual pressure to keep it set, or will it stay where selected? Should means be provided to control the selector manually, or to over-ride in an emergency? If the relay system fails, should the selector move automatically to a given position? The answers to these problems affect the selector design.

11.6. Special Requirements of Pneumatic Systems. The control of a jack by compressed air introduces special system problems with their own solutions. If the load on the jack is constant, the air, usually from a bottle, expands into the jack to extend it. The pressure in the bottle drops off as the jack moves, and has obviously to be calculated accordingly. If the peak load of the jack is towards mid-travel, say, once the air pressure has moved the jack past the peak point, the supply of air can be reduced, or even shut off, allowing the air in the closed system to expand to complete the motion.

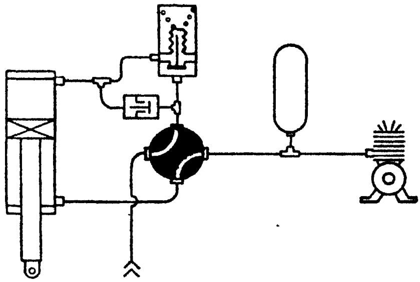


FIG. 106. USE OF REDUCING VALVES IN PNEUMATIC SYSTEMS

In the same way a double-acting jack may have different operating pressures in the two directions of motion (this is highly probable).

Since some means of preventing excessive consumption of air is often desirable, particularly because of the time taken to compress air, electro-pneumatic control is sometimes used so that limit switches operated by the jack will shut off the air supply as soon as the jack reaches the end of its travel, irrespective of the pressure in the jack (see Chapter 16). Alternatively, reducing valves may be used in the low pressure part of the system to lower the pressure in that part to a minimum, and thus avoid excessive compression and wastage of the air. This system is illustrated diagrammatically in Fig. 106. A non-return valve allows return or exhaust flow past the reducing valve, although some types of reducing valve will allow return flow without it.

CHAPTER 12

REMOTE CONTROL SYSTEMS

12.1. Single-acting Remote Controls (Spring Returned). For the purposes of this book a remote control is considered as a means of reproducing motion of a lever, wheel, etc., on a second and remote lever, wheel, etc., more or less accurately, the essential feature being the synchronism of the "transmitter" and "receiver"; simple jack systems controlled by selectors will not fall in this category.

The most elementary type of remote control is shown in Fig. 107 (*a*), and consists of two jacks joined together by a single pipe run. If the system is initially full of fluid, movement of the transmitting jack, pump, or master cylinder as it is often called, will cause a corresponding motion of the receiver piston, with an accuracy determined only by the compressibility of the fluid, and mechanical play, etc.

Such a system, however, suffers from the faults that there is no provision for making up small fluid loss, nor for thermal expansion or contraction of the fluid. Since most hydraulic fluids expand at least ten times more than normal material for the pipe lines for a given temperature rise, the effects of thermal change on synchronism can be serious.

Illustration (*b*) shows the normal method of feeding the pipe line of a hydraulic brake system. When the piston is fully released, it uncovers a small port in the cylinder wall connected to tank, releasing any surplus fluid or admitting more if some has been lost or the volume has contracted. The only weakness of this simple system is that the gland has to pass over the small hole in the cylinder wall, and if high pressures are used this may cause eventual damage to the seal.

Another way of achieving the same object is shown at (*c*); this uses a feeding valve as described on page 132 (Fig. 91 (*b*)), but will only work reliably if the speed of movement of the transmitter is never slow, and is always sufficient to close the feed valve. On the return stroke, provided the transmitter volume is greater than the receiver volume (which it must be), there is no risk of the valve remaining shut. Illustration (*d*) shows another means for feeding the pipe line through a valve carried on the piston and opened by a small cross-pin when the transmitter is fully off; this overcomes the difficulty of passing a seal over the feeding hole of type (*b*). The piston assembly, however, is much longer than on other types.

Type (*e*) is a single pipe control with spring return, the springs being arranged on both transmitter and receiver, by virtue of the

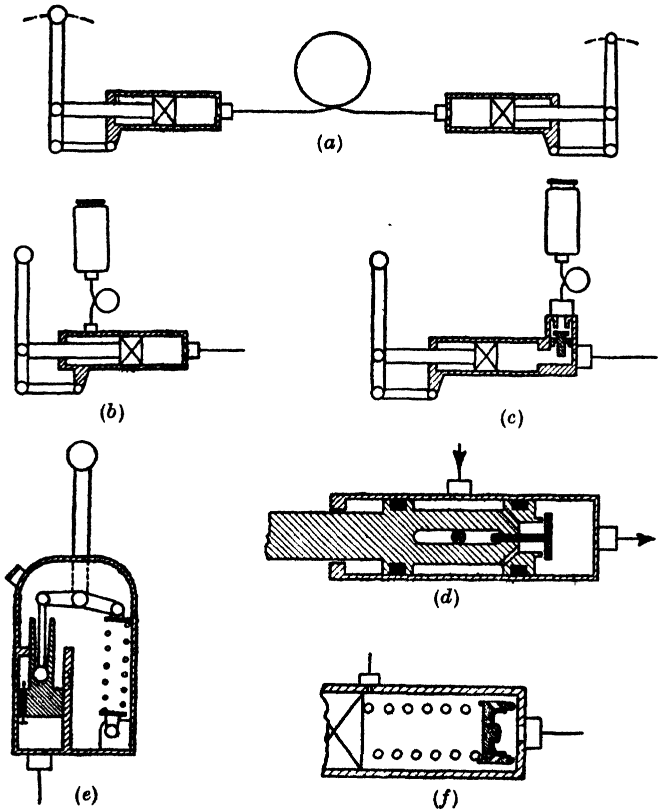


FIG. 107. REMOTE CONTROL SYSTEM (SINGLE-ACTING)

(*a*) = Simple system.
(*b*) = Feed system.
(*c*) = Feed system with special valve.
(*d*) = Feed valve in piston.
(*e*) = Controls with balanced return springs.
(*f*) = Valve assembly used on vehicle hydraulic brakes.

geometry chosen or by the use of cams, to exert corresponding forces, so that the control will stay put at any intermediate position. Motion in one direction causes compression of the fluid and the effort that can be exerted is limited only by mechanical considerations. In the reverse direction the torque which can be transmitted can only be that corresponding to the initial tension of the springs, reverse torque causing relief of this tension and its corresponding pressure. The control is fully subject to temperature dilation trouble, causing loss of phase, but can be re-primed and re-synchronized by moving back to the stop in the direction which opens the valve in the piston (or mounted alongside it).

One of the advantages of this control, however, is that the oil is normally under some pressure; this has a marked advantage in that all air bubbles, etc., in the fluid are compressed and the control does not feel "spongy". A similar object is achieved in some small degree in vehicle braking systems (illustration (*b*)) by means of a valve assembly shown at (*f*). Normally the transmitter operates through a non-return valve formed by a rubber cup washer which collapses to pass fluid. On the return stroke the returning fluid has to lift a light relief valve formed by the valve assembly against the spring return force. By this means the pipe line and receiver are maintained under a small pressure sufficient to improve the feel of the control, and to prevent the system pressure from dropping below atmospheric, which might allow air to get in. In addition the valve assembly is extremely useful in allowing the transmitter to act as a pump for priming or bleeding purposes.

To overcome the inherent difficulty of thermal dilation, the "shadow pipe" system of Fig. 108 (*a*) has been proposed. In this the transmitter body is so mounted that a second jack or pump, and a shadow pipe run in parallel with the main pipe and as near it in capacity as possible, are arranged to compensate automatically for dilatation of the main transmitting volume. It will be seen that equal expansion or contraction of the fluid has no effect on synchronism owing to the differential lever mounting of the two jacks. The shadow jack, however, has to be spring loaded with initial tension so that it does not move when the transmitter is operated, and this limits the force to be transmitted. Full compensation occurs only if the temperature changes affect both systems equally. In (*b*) is shown a substantially similar device using a double piston spring-loaded compensator in place of the differential lever of the earlier mechanism.

It has been proposed to use the shadow pipe of this type of control as an emergency pipe in the event of failure of the main pipe, using change over valve gear.

Another method which will not, however, necessarily compensate for local variations in temperature is to use a type of thermostatic compensator. If the coefficient of volumetric expansion of the fluid is, say, ten times that of the pipe line, etc., and if the pipe line is filled with, say, glass beads (glass being almost expansionless compared with the fluid) occupying nine-tenths of the internal volume of the pipe, then the relative expansion of the pipe and the remaining one-tenth volume of

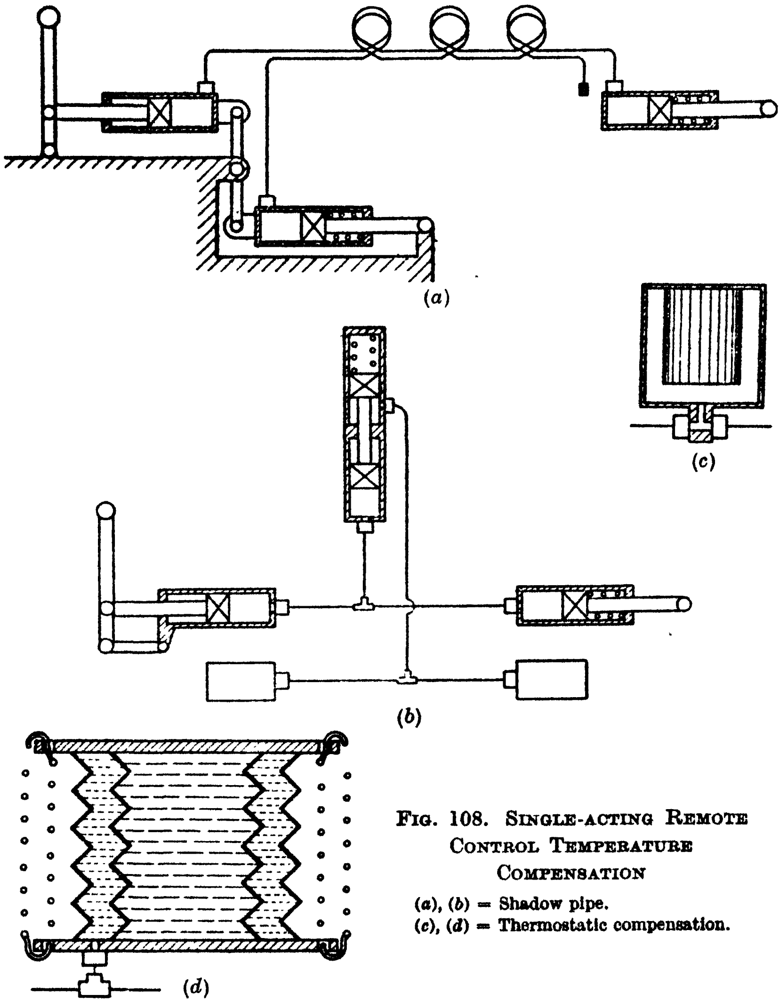


FIG. 108. SINGLE-ACTING REMOTE CONTROL TEMPERATURE COMPENSATION

(a), (b) = Shadow pipe.
(c), (d) = Thermostatic compensation.

fluid are the same. As the presence of glass beads is likely to be inconvenient, a more practical solution is to use a compensator as shown diagrammatically at (c) of Fig. 108. This has a metal casing with an internal solid plug, etc., made of, say, Invar or glass, such that the expansion of the casing is greater than the expansion of the fluid volume in the compensator by the surplus expansion of the fluid from the pipe lines. At (*d*) is shown a similar device using thermostat bellows, with a suitable choice of volumes, thermostatic liquid or gas, etc.; this device, however, has to be spring loaded and this limits the maximum pressure in the transmission system.

12.2. Double-acting Remote Controls. By "double acting" is meant a remote control which exerts a direct action through a fluid column in both directions and does not rely on spring return. It is true, however, that certain of the double-acting systems can exert only a limited effort due to the presence of a spring-loaded compensator or equivalent device, and that the return action is positive only within certain limits.

The elementary double-acting remote control is that shown diagrammatically in Fig. 109 (*a*); it will be seen to consist of two jacks connected together so as to operate in series (see also page 144). The fluid volumes of the jack need not necessarily be equal, provided the system is symmetrical.

Other mechanical arrangements of the units are shown at (*b*) and (*c*); in the case of (*b*), operating loads cause friction losses on the pivots, etc., while care has to be taken that the geometry of the two units is correct (as drawn it is slightly incorrect, the motion of the receiver not being exactly that of the transmitter). In the case of (*c*), although the unit is long, it has fewer glands than type (*a*).

In all cases, to obtain satisfactory performance feed valves will have to be fitted to the transmitter, and usually to the receiver as well. The solutions illustrated in Fig. 107 (*d*) and (*e*) are the usual ones.

To re-establish or alter synchronism the usual scheme is to have a short-circuiting valve on the transmitter, which can be opened to allow this unit to be re-phased. If the transmitter lags the receiver, pulling it to the receiver stop and then opening the by-pass valve will allow the transmitter to be pulled to its stop. If the transmitter is ahead of the receiver, the same process in the opposite direction is necessary. A full cycle opening the valve at each end is the obvious routine.

Another method of re-synchronizing is to use a device as shown at (*d*) (Fig. 109), which is connected to both lines, and by screwing the separating piston back and forth, the relative phasing of transmitter and receiver can be altered at will.

With most double-acting remote controls it is possible to mount three or more in series (Fig. 109 (*e*)), so that two transmitters can operate one receiver (i.e. dual-control) or one transmitter can operate two or more receivers (as on the throttles of two engines). However, this usually introduces additional inaccuracy as the pipe lines are not all the same length, and temperature compensation is made more difficult.

Fig. 109 (*f*) shows an interesting control using a tension wire sliding in a pipe in conjunction with a fluid column. The wire

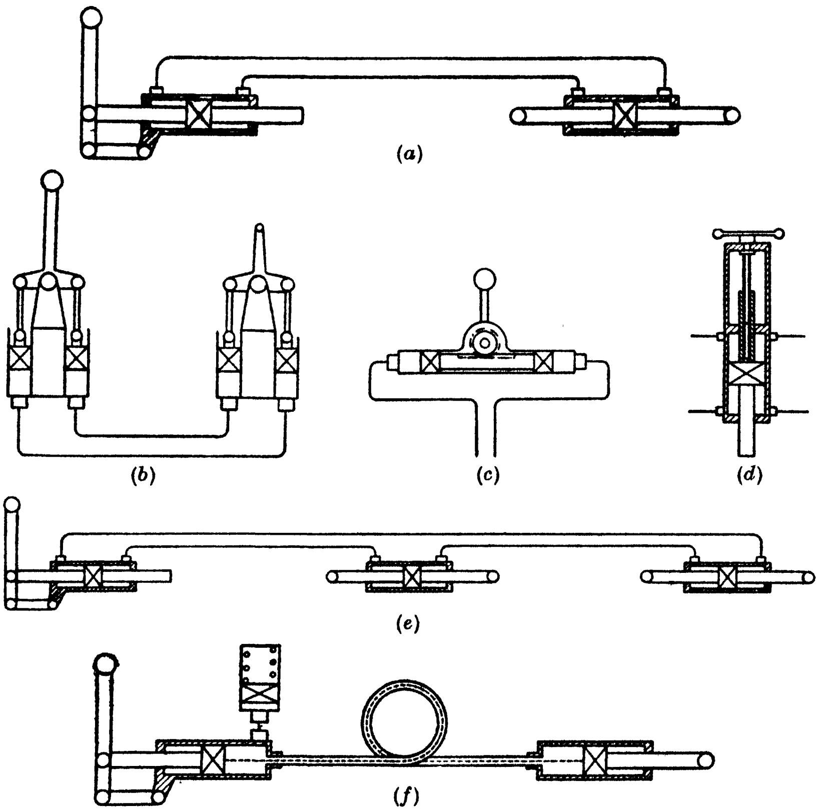


FIG. 109. DOUBLE-ACTING REMOTE CONTROLS

(*a*) = Simple axial type.
(*b*) = Rocker and links.
(*c*) = Rack and pinion.
(*d*) = Phase compensation.
(*e*) = Several controls in series.
(*f*) = Tension wire remote control.

transmits tensile loads, the fluid compression. For satisfactory performance pre-loading of the oil by a spring-loaded accumulator, etc., as shown, is desirable to take up slackness due to air bubbles, etc., and to act as a reservoir against thermal changes. Owing to the wire, temperature change does not affect synchronism (provided the wire and pipe have equal expansion coefficients), but except on straight runs the friction of the wire, in spite of it being well lubricated, is liable to be high.

Means of temperature compensation are known in addition to the thermostat systems described for the single-acting control. Fig. 110 (*a*) shows a method using a spring-loaded piston in each pipe line, with a common spring so that equal pressure is developed in each pipe line. Thermal changes merely cause the two

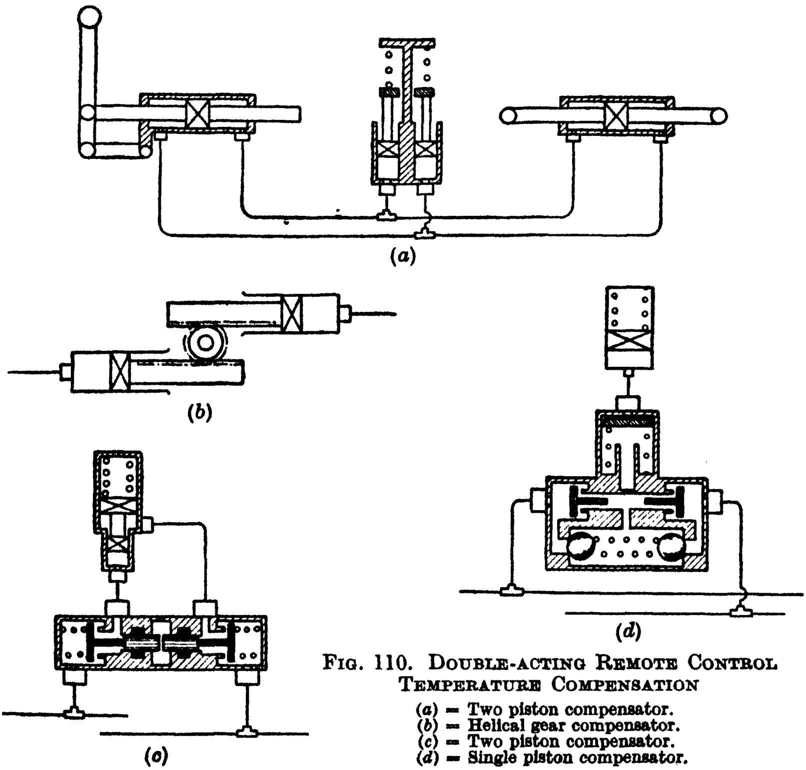


FIG. 110. DOUBLE-ACTING REMOTE CONTROL TEMPERATURE COMPENSATION

(*a*) = Two piston compensator.
(*b*) = Helical gear compensator.
(*c*) = Two piston compensator.
(*d*) = Single piston compensator.

pistons to rise or fall together, varying slightly the system initial pressure; operating loads, however, cause a difference in pressure in the two lines, which up to the initial pressure of the compensator will not cause it to move. This operation has the advantage of initial system pressure together with complete and accurate temperature compensation. The compensator may have two pistons as shown, or a single one with stepped diameters. To reduce the initial pressure, or eliminate the pressure limitation of the system, the two pistons can be connected by means of a right angle helical gear drive, so chosen that although the rack on one piston can turn the gear, the gear cannot move the second piston as the gear is irreversible in this direction. This means that the two pistons can move together an equal amount due to thermal effects, but not differentially due to operating loads (Fig. 110 (*b*)).

Another means of preventing the compensator from lifting when the operating pressure exceeds the compensator pressure is shown in Fig. 110 (*c*). A unit is connected between the system pipe lines and the compensator; it contains two valves with equal spindle diameters, normally abutting and both open. When the pressure in one line rises as during operation, the valves

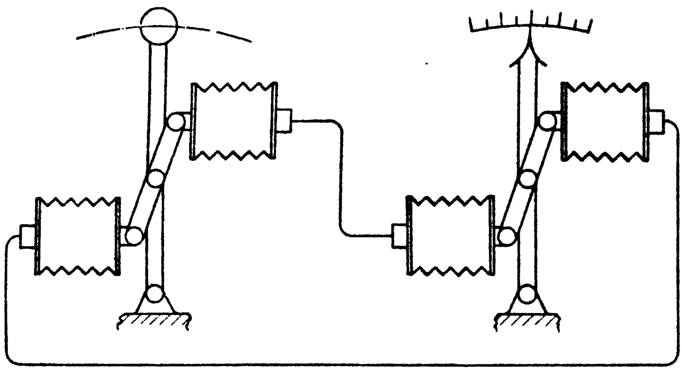


FIG. 111. DIFFERENTIAL LEVER REMOTE CONTROL

move over and isolate the connection to that particular cylinder of the compensator.

Another type of compensator is shown at (*d*). This uses a single spring loaded piston loading both lines. The outlet of the compensator is restricted by a valve device allowing small flows only, and which closes if the flow is rapid, as with a broken pipe (i.e. it is a hydraulic fuse—see page 124). Non-return valves allow losses in the pipes to be made up, and equally calibrated relief valves take care of thermal expansion. This type of compensator would appear to be inferior to the two-piston type.

A by-pass valve fitted to the transmitter can be used to correct synchronism as mentioned above, with all these devices. The system can be filled and the compensator charged by pumping fluid in the transmitter by means of a hand-pump with the by-pass valve open; if the latter is then shut, the compensator conveniently stays compressed owing to one pipe being locked, while the external pump is removed.

Fig. 111 shows another known system actually applied to remote reading instruments; the transmitter and receiver levers are connected to a pair of bellows (or, obviously, jacks) at each station, through a differential lever joining the two bellows. Thermal changes merely cause the differential lever to rotate about its own axis without shifting the latter; operating loads, however, shift the axes of the two levers, closing one pair of bellows and extending the other.

Most double-acting remote controls can be fitted with the double type of hydraulic lock illustrated in Fig. 77 (*c*), page 110, so that the receiver is made irreversible; this, however, is at the expense of accuracy, since some transmitter travel is required to move the hydraulic lock piston and open the valve.

12.3. Rotary and "Spherical" Remote Controls. Hydraulic remote controls exist which reproduce continuous rotary motion of a handwheel, etc., on a transmitter at a remote receiver; another type of control has a knob on the transmitter which can be moved over a sector of a sphere (or two planes) with a corresponding motion at the receiver end.

Fig. 112 (*a*) illustrates one type of rotary control, using the same compensation mechanism as that illustrated in Fig. 110 (*a*). Rotation of the transmitter crankshaft causes the three pistons to move up and down, reproducing the motion exactly at the receiver or motor end. It is important that the motion of the piston shall be "symmetrical", i.e. truly harmonic, since the total volume of the system is constant; this can be achieved by circular eccentrics operating on flat followers on the ends of the pistons. Owing to the use of three pistons, return springs on the pistons are not necessary; the three pistons are necessary anyway to make the control self-starting. The compensator operates as before, being immovable up to a given (spring) load due to operating loads, but moving up or down to compensate for the equal changes of volume due to temperature change. This type of control is capable of reproducing continuous motion with a very great accuracy.

In order that small leakage in a single pipe will not cause backlash during rotation, a receiver of the type shown in Fig. 112 (*b*) may be useful. This has an additional degree of freedom between the wobble plate and the driven crank, such that the wobble plate takes up a position in contact with all three pistons, slight leakage affecting synchronism but not introducing backlash, when a compensator is used.

Similar systems using three or more simple transmitting systems with some form of feed system to each pipe line are known; these have the advantage that leakage, etc., is made good, but are not temperature compensated. For most applications such a refinement is not necessary, and the ability to reproduce rotary motion even inaccurately is all that is necessary.

Fig. 113 (*a*) shows the transmitter element of a "spherical" control. This has three pistons disposed equally around the lever

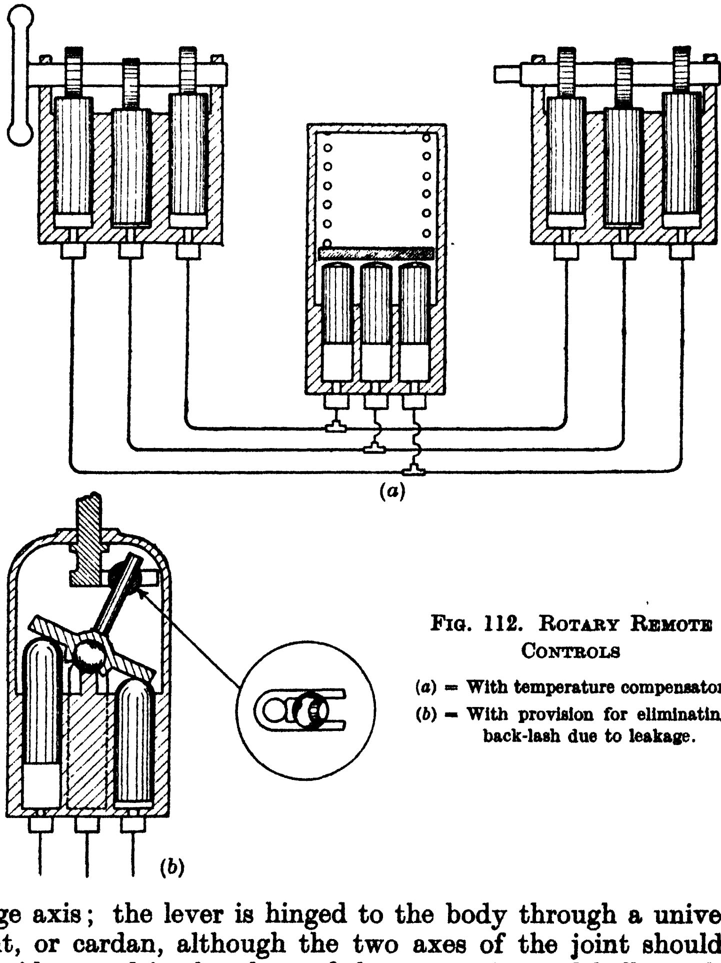


Fig. 112. Rotary Remote Controls

(*a*) = With temperature compensator.

(*b*) = With provision for eliminating back-lash due to leakage.

hinge axis; the lever is hinged to the body through a universal joint, or cardan, although the two axes of the joint should be coincident and in the plane of the connecting rod balls, and not as shown in the sketch, separated for clarity. The lever can thus move over a sector of a sphere, but not rotate; this causes the piston to oscillate in a manner which can be reproduced exactly at the receiver end. The pistons will have to be fitted with priming valves as described for the simple systems. Since the motion of the piston is not truly harmonic, the kinematics of the whole system is imperfect and compensation by the means described for the rotary control will be difficult if not impracticable; in addition the pre-loading of the system will cause friction loads on the ball joints, etc., to the detriment of the performance of the control.

An equivalent "spherical" action can be obtained by connecting a lever to two double-acting controls at right angles, so that

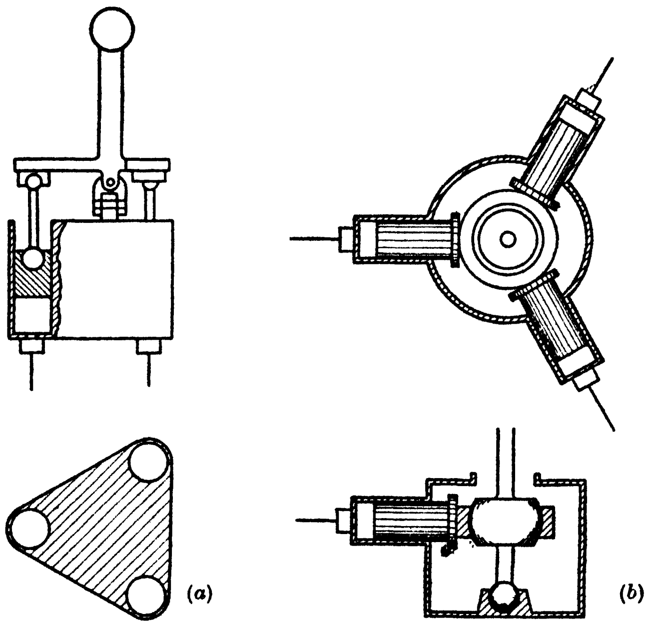


FIG. 113. "SPHERICAL" REMOTE CONTROLS

the motion of the lever in two planes is transmitted separately; this, however, uses four pipe lines instead of three.

Fig. 113 (*b*) however shows a "spherical" control which is kinematically correct, and can be used with a three-piston compensator. The motion of a rocking lever moving in two planes is transmitted to the pistons through a ball joint and circular ring contacting flat followers on all three pistons. This type of transmitter could obviously be used for full rotary motion by mounting the central roller on a rotating eccentric.

12.4. Pressure Variation Remote Controls. So far the remote controls described operate by displacement of a column of liquid and are therefore not suitable for use with compressible gases, such as air. There is a type of control using reducing valves, which can, however, give fairly accurate remote control with air systems. Since it uses an external source of air pressure it is not strictly a remote control, rather a servo system (see Chapter 15), but as it is used as a remote control it is described in this section.

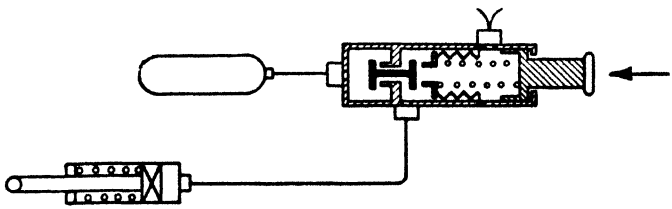


FIG. 114. PRESSURE VARIATION REMOTE CONTROL

Fig. 114 shows the system. The transmitter lever operates a "power brake" type of valve (see Chapter 7), and causes a variation in pressure in the pipe line to the receiver jack proportional to the deflection of the transmitter spring, and therefore the lever knob. This in turn causes a corresponding deflection in the receiver spring; thus, provided the receiver operating load is small compared with the spring force, the transmitter

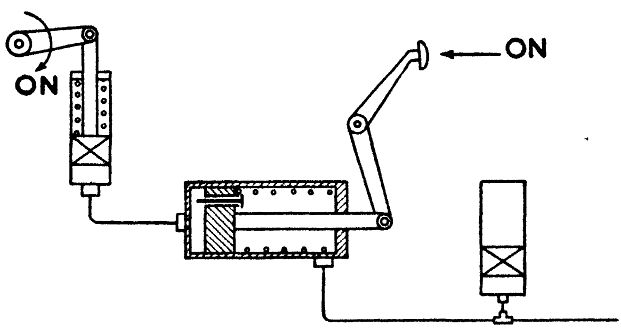


FIG. 115. REVERSE PRESSURE SYSTEM

and receiver can be more or less synchronized. Obviously, however, this system has limitations, but is of interest because it is one of the few using a single pipe line.

This system, however, is very effective for the remote control of jacks with very small travel, e.g. the diaphragm chamber of another reducing valve. Such valves controlling the flow of steam have been operated remotely by a single pipe line and a system of this type. Again, however, it can be said that this is hardly a true remote control system.

Fig. 115 shows a reverse pressure braking system which may be considered as a type of single-acting remote control. The brake is off when the pedal is released and the internal valve of the "transmitter" is open, allowing accumulator pressure to extend the "receiver" and compress its spring. When the pedal is depressed the receiver follows up the transmitter under the action of its spring, the accumulator keeping the transmitter valve closed. This system has some advantage when combined with a normal power brake which consumes accumulator pressure during application, the reverse system (applied for example to trailer brakes) compensating for this to some degree, or at least not requiring additional volume. Furthermore, should the system fail, the brakes will be applied by the action of the spring.

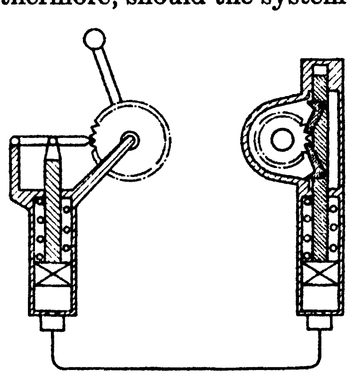


FIG. 116. IMPULSE REMOTE CONTROL

Fig. 116 shows very diagrammatically another type of remote control using a single pipe line, but obtaining rotary motion in either direction by sending impulses down the pipe line. Both transmitter and receiver carry a spring-loaded piston, the two springs being balanced with the pipe line full of oil. Rotation of the transmitter lever, by virtue of a mechanism shown diagrammatically as a sharp pointed lever and ratchet wheel, causes impulses of positive or negative (i.e. reduced) pressure to be sent down the line. Each impulse causes the receiver piston to move a corresponding amount in one or other direction; this movement is transferred to a second shaft through a mechanism shown diagrammatically as a ratchet wheel with two pawls, arranged so that while one pawl is causing a rotation the other is lifted out of the way, so that the piston rod carrying the pawls can return to the central position again. Some means (e.g. a spring-loaded click fitting the ratchet wheel) must be provided to brake the ratchet wheel during this return stroke.

Thus continuous rotation of the transmitter causes an equivalent step by step motion of the receiver. This system, although extremely ingenious, is obviously fraught with difficulties, not the least of which is the effect of thermal dilatation of the fluid.

12.5. Two-stage Remote Controls. A useful means of improving the performance of a simple single-acting remote control system particularly suitable for vehicle brake operation, is to use a two-stage transmitter. By this device initial motion is by means of a large volume low pressure cylinder, and when the system pressure rises, as when the brake shoes contact the brake drum, the low pressure cylinder is cut-out and a high pressure low displacement cylinder operates instead. In the following illustration the actual master cylinders are much simplified, and usually use collapsing seal cups instead of the separate valves shown.

An early system giving the two-stage effect is shown in Fig. 117. In this a normal simple master cylinder or transmitter passes fluid direct to the receiver or brake, through an open valve in the change-over device. When the pressure rises to a predetermined amount the piston in the latter moves against its spring

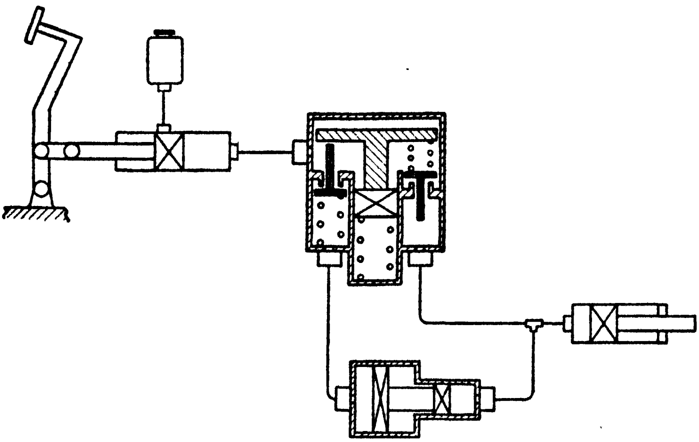


FIG. 117. TWO-STAGE REMOTE CONTROL SYSTEM

under line pressure, closes the valve in the direct line to the receiver (against a light spring not shown), and then opens a second valve normally held to its seat by a balancing spring, and admits fluid to an intensifier or hydro-pump, which then relays high pressure to the receiver.

While this system uses a change-over valve depending on line pressure, similar in principle to the duplex hand-pump described on page 34, the transmitter of Fig. 118 (*a*) uses a change-over at a predetermined point in its travel. Normally pedal pressure closes the tank feed valve and advances both pistons, the two being locked together by a simple ball lock. When the larger piston uncovers a groove in the cylinder, the inner piston advances alone, leaving the larger piston locked by the ball. This locking is obviously necessary to prevent the larger piston moving back under pressure, if it were, for example, spring loaded. This system has the advantage of simplicity, but the disadvantage inherent in the predetermination of the change-over point.

The third type of two-stage master cylinder exists in several forms (Fig. 118 (*b*), (*c*), (*d*), (*e*) and (*f*)). Each consists of two pistons of different diameter and coupled together so that they both deliver fluid at the same time, the low pressure piston

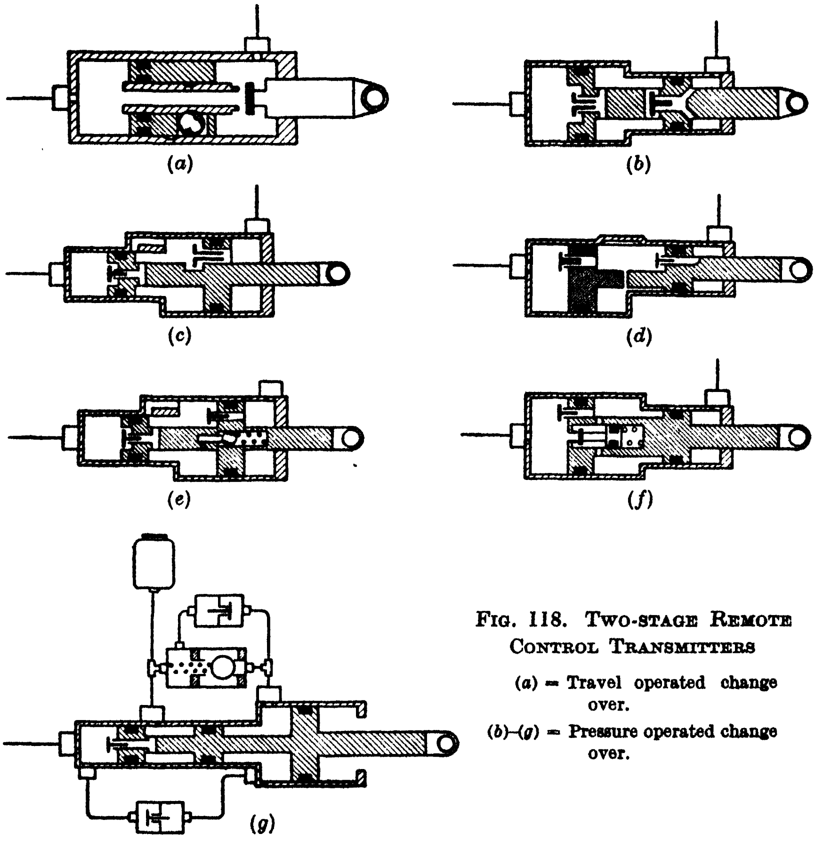


FIG. 118. TWO-STAGE REMOTE CONTROL TRANSMITTERS

(*a*) = Travel operated change over.

(*b*)–(*g*) = Pressure operated change over.

operating only to advance the receiver under the slight load when the brake shoes are advanced into contact with the brake drum. The functioning of these types seems rather uncertain, but some appear to operate reasonably well. Fig. 118 (*b*) shows a unit with the large piston ahead of the small one; initial motion delivers fluid from the large cylinder, cavitation in the high pressure cylinder being avoided by it filling up from the tank. When the pressure rises in the receiver, the low pressure piston is by-passed through the bleed hole in its feed valve; the effectiveness of the low pressure piston depends on the rate of flow through this restriction and is thus effective at relatively low pressure and a high rate of delivery only, if the restriction is not to be so small as to upset the by-passing of fluid past the low pressure piston when the high pressure piston is working. (The volume of fluid to be by-passed is obviously due to the difference in swept volumes of the two pistons.) The return action and feeding arrangements are straightforward.

Type (*c*) is similar, but has the position of the two pistons reversed. In this case initial motion allows fluid to by-pass the high pressure piston from the low pressure chamber, the latter having to by-pass through a restriction to the reservoir, when the pressure rises sufficiently to keep the valve in the high pressure piston closed. The return action and feeding arrangements are straightforward, an additional port in the high pressure cylinder being necessary, as shown.

Type (*d*) is similar to type (*b*), except that the cylinders are eccentric so that air pockets are eliminated on the upper surface of the unit, a by-pass groove allowing air bubbles to pass the high pressure cylinder. The pistons may be separate as shown.

Type (*e*) is similar to type (*c*), except that the by-passing of the low pressure piston is through a relief valve; this will improve the delivery action, but means that work is wasted in pumping oil through the relief valve. The actual value of such a system is, therefore, doubtful unless the "advance" stroke is large compared with the actual working (i.e. high pressure) stroke.

Type (*f*) is again similar to type (*b*) and type (*e*) as well, using a relief valve in the low pressure piston, but mounted ahead of the high pressure piston. The remarks on type (*e*) apply.

Type (*g*) works on a similar principle as well. As before, the low pressure piston delivers through the high pressure piston (actually into the cylinder through an external non-return valve). When the pressure rises in the receiver and thus in the low pressure cylinder, a relief valve by-passes the latter to tank. This relief valve, however, is fitted with a second seat so that if the flow through it is great, the ball seats again and blocks the flow; this means that when extra effort is exerted and the flow rate is great, the low pressure piston will deliver as well. Return flow is normal and feeding of the low pressure cylinder takes place through a non-return valve direct from the tank.

The disadvantage of all these systems is that the change-over point is likely to be felt, and what is really wanted is some truly progressive change of delivery.

Another two-stage braking system using power control is described in Chapter 11.

CHAPTER 13

VARIABLE SPEED SYSTEMS

It is particularly important on many hydraulic systems, as on machine tools, to be able to vary the speed of operation at will. This can be carried out in the following ways, sometimes more than one way being combined—

(i) By varying the pump output manually;
(ii) By using several pumps in combinations;
(iii) By restricting or throttling the output of an automatically variable delivery pump, or a pump accumulator system, or by throttling the inlet.
(iv) By by-passing part of the pump output with a flow dividing valve;
(v) By varying the volume of the operating jack.

13.1. Variation in Pump Delivery. Pump delivery can be varied by—

(i) Alteration in its speed;
(ii) Alteration of its stroke in a variable stroke type of pump;
(iii) Using two or more pumps of different delivery in parallel so that by stopping and starting the pumps in various combinations different total deliveries can be obtained.

The first system is an easy one when the pump is electrically driven, although the electric motor involved is comparatively complicated for normal requirements. Mechanical variable speed gear boxes have been used successfully with constant speed electric drive.

Several of the pump mechanisms previously described can readily be adapted to give a varying output by reducing the working stroke manually by means of a control wheel, etc.

The third system is simple enough, but varies the output in fixed steps. Two pumps in parallel can give three ranges of output corresponding to—

Pump A, Pump B, Pump A plus B.

Three pumps in parallel can give seven steps corresponding to—

Pump A,	Pump A plus B,	Pump B plus C,
Pump B,	Pump A plus C,	Pump A plus B plus C.
Pump C,		

Since, however, variable stroke pumps are readily available, such a complication as three pumps in parallel hardly seems worth while, although the two pump system is probably excellent for such duties as presses, etc., where a great part of the working stroke is at low pressure, and where a relatively cheap type of pump can be used, cutting out in favour of a smaller delivery high pressure pump for the final working stroke. Automatic isolation of the low pressure pump can be effected by a valve as shown in Fig. 31 (*a*). Any normal type of automatic cut-out will operate in the low pressure system to by-pass it, without interference from the other pump.

13.2. Restriction of Pump Output. With a variable delivery pump the flow of oil to the system proper can be metered through a restriction, the delivery of the pump automatically adjusting itself to the reduced flow. An automatic flow control valve or throttle as described in Chapter 8.3 is to be preferred to a simple restrictor. This is an extremely simple system, but is liable to variation of speed owing to change in viscosity of the oil, temperature effects, etc., and the metering restriction may have to be adjusted from time to time to keep the speed constant. On the other hand, it is possible to evolve a restriction compensated for changes. Viscosity compensation systems are referred to in Chapter 17.5.

By fitting the flow control valve in either jack line, control in one direction only can be exercised. By fitting in the tank return line, control in both directions can be exercised, but note that as the volumes of the jack returning to tank may not be the same in both directions, the degree of speed control may not be similar.

Fig. 119 illustrates a typical machine tool application of this type, with the flow control valve in operation only at the end of the travel in the advance direction, being short-circuited by a cam operated valve over the remainder of the travel. The system is an automatically reciprocating one, with a pilot selector feeding a relay slide selector handling the main flow (see also Chapter 14). Obviously various arrangements of cams and restrictors can be used to vary the feed rate over the working stroke as desired.

With an accumulator system charged from a pump, in conjunction with an automatic cut-out valve, a similar degree of speed control can be effected by suitable restriction, the pump cutting in from time to time to re-charge the accumulator. The additional variable of accumulator pressure, however, introduces further irregularity.

In many cases restriction of the flow out of a jack will be preferable to restricting the inlet to it, because this will pressure-load the system, reduce elasticity problems and hold the jack against reverse or negative loading.

A further known method is to restrict or throttle the inlet to the pump. This will tend to make it cavitate and may cause frothing of the output; it appears, however, to be successful in certain applications.

Although the normal flow divider is usually installed in a single line, it may be convenient in some cases to connect the by-pass connection in the inlet to a jack, and connect up the pressure controlling piston and the restrictor in the outlet line

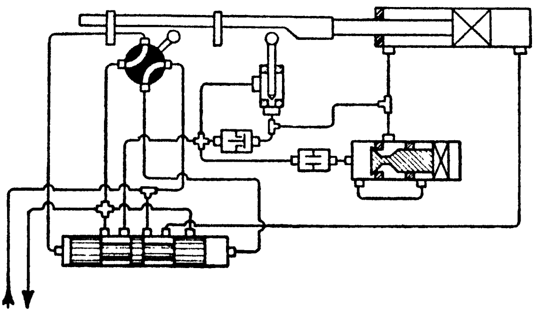


FIG. 119. MACHINE TOOL SYSTEM WITH SLIDE OPERATED FLOW CONTROL VALVE TO VARY SPEED

of the jack. In this case the back pressure entering the restrictor is kept constant to control the flow rate but the pump pressure rises progressively as the work load pressure requirements may dictate, the by-pass port acting as an automatically adjusted relief valve to absorb surplus flow. This has the advantage over a normal flow divider of putting both sides of the jack under pressure and making the unit at least partially resistant to negative jack loads.

13.3. Use of Flow Dividing Valves. The flow dividing valves of various types already described in Chapter 8.2 can be, and are, used to control the speed of a system by by-passing part of the pump output, even if at the expense of a slight wastage of power. It is possible to use a selector incorporating several ports, which in turn control the flow of fluid past several different flow dividing valves, giving different rates of flow for each position of the selector.

13.4. Variation in Jack Volume. Another means of obtaining variable speed from a constant delivery pump is to use jacks of different volumes (i.e. at different pressures), either in parallel, or using a multi-volume construction as described in Chapter 4. If, for example, the machine tool slide, etc., is fitted with two operating jacks, by suitable selection varying speed of operation can be obtained corresponding to—

1. Use of jack A;
2. Use of jack B;
3. Use of jacks A and B together.

If $B = 2A$, the speeds are in the order 1, 2, 3. The combination of two jacks and two pumps can obviously give 9 speeds, but at the expense of considerably more complication than would appear to be present with a variable delivery pump.

Another interesting possibility is to use a 6-way selector of the type shown in Fig. 41 (g), so arranged that the jack can extend on the full area of the cylinder or on the area of the rod by the differential area system. Extension speeds can thus be obtained corresponding to—

(a) Piston rod displacement;
(b) Full piston displacement;

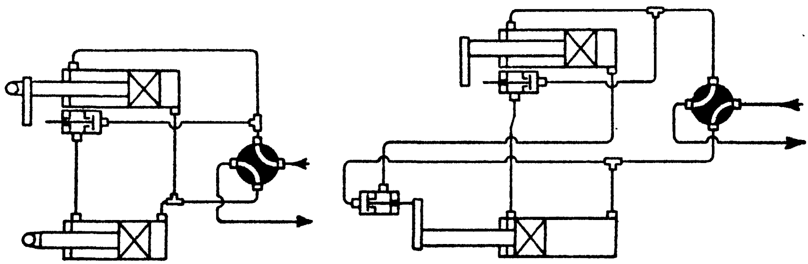
but the closure speed remains constant.

CHAPTER 14

SEQUENCE SYSTEMS

A SEQUENCE system is one where one jack completes an operation before a second is permitted to operate, i.e. in sequence. The usual method of achieving this is with sequence valves (see page 69).

14.1. Simple Sequence Systems. The simplest sequence system is that shown in Fig. 120 (*a*). It will be seen that the fluid closes



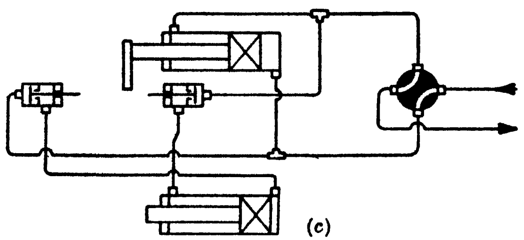


FIG. 120. SEQUENCE SYSTEM

(*a*) = One direction only.
(*b*) = Sequence 1—2—2—1.
(*c*) = Sequence 1—2—1—2.

one jack, the second being isolated until the first reaches the end of its travel, opens the sequence valve and allows the second jack to move. This last remark is not strictly correct, since the second jack may have already moved under the action of external force tending to close it, and thus true sequence operation will occur only if there is no tendency for this to happen. The defects of sequence systems of this type have already been referred to in the discussion on sequence valves themselves in Chapter 5, but will be repeated; when the sequence valve opens there is a momentary drop in pressure as the fluid passes to the second jack and this may allow the first jack to extend again, thus shutting the sequence valve. The effect in practice is to set up chatter of the valve, and means of overcoming this in the valve may be provided.

In place of a mechanically operated sequence valve a switch may be tripped by the first jack and an electrically operated valve opened.

This system provides sequence operation in one direction, but both jacks will move in the other direction together, the sequence valve allowing free return flow.

To obtain sequence operation in the reverse direction, two sequence valves are needed. This system, giving reverse operation in the opposite order to forward motion (i.e. 1–2–2–1), is shown in Fig. 120 (*b*). This is more or less self-explanatory.

To obtain equivalent but unsymmetrical motion (i.e. 1–2–1–2 instead of 1–2–2–1), the system shown at (*c*) can be used, the second jack always lagging the main jack which operates both sequence valves.

There is no limit to the number of jacks which can be operated in sequence by increasing the number of sequence valves accordingly.

Jack construction with the valves integral or built inside in various ways are known; since the sequence valve is usually connected to one volume of the jack which operates it, this construction may save piping, etc. On the other hand it is obviously not necessary for the valves to be operated by the jacks themselves, and they are often operated by some part of the mechanism moved by the jack.

14.2. Complicated Sequence Systems. The word complicated is used to distinguish sequence systems more involved than the simple types just described; the difference is one of degree only, and fortunately the complicated systems are comparatively rare since they are excessively complex.

One known system is used on aircraft undercarriage retraction to enable the undercarriage doors to close after the undercarriage has extended and open again while it retracts, closing again finally. This sequence is obviously: 1–2–1–1–2–1, 1 being the door jack, and 2 the undercarriage.

The following description of a German system is reproduced, by permission, from *Aircraft Engineering**—

"The sequence of flow when raising the undercarriage is illustrated by the diagram in Fig. 121, the circuit being reduced to

* *Aircraft Engineering*, October, 1946—"Hydraulic Systems in German Aircraft," by M. Hodgson. See also C.I.O.S., Report XXXI—41, published by H.M. Stationery Office.

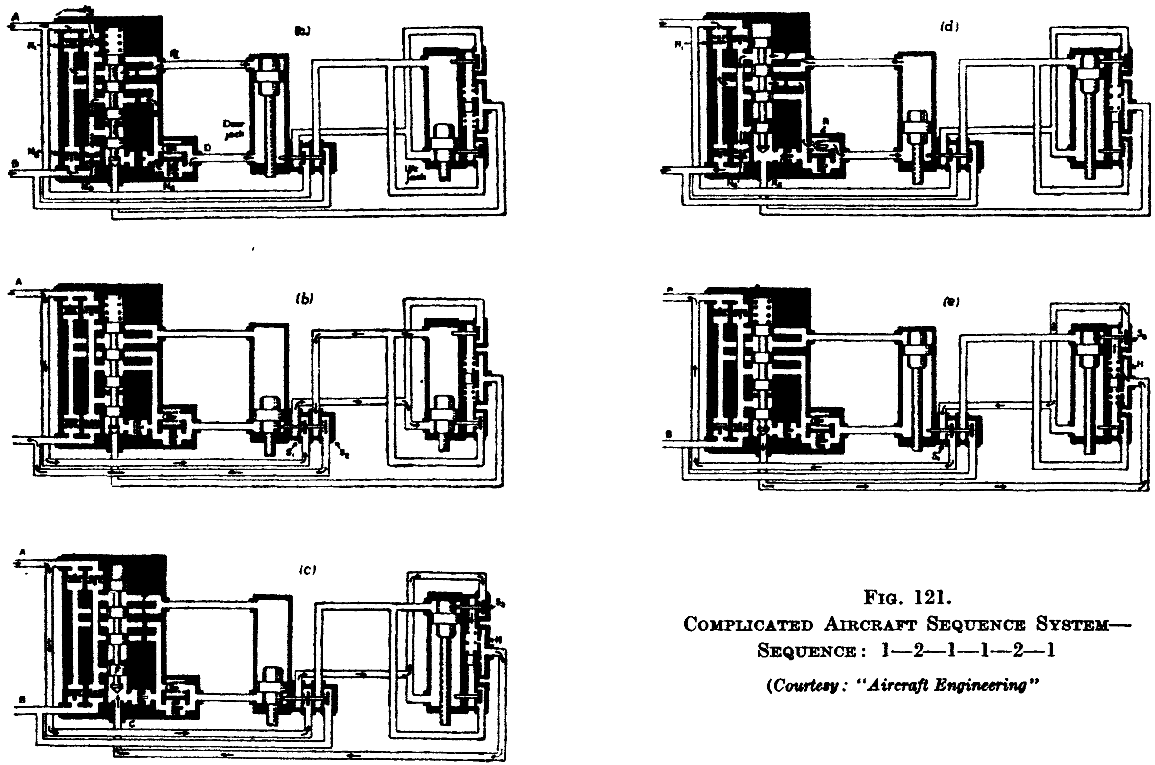


Fig. 121.
Complicated Aircraft Sequence System—
Sequence: 1—2—1—1—2—1

(Courtesy: "Aircraft Engineering")

its simplest terms by the omission of the emergency arrangements. Co-ordination of the movements of the doors and the wheel jacks is performed by mechanically operated sequence valves, housed in the jacks, and a component called a flow reversing valve, which consists of a plunger valve and several non-return valves.

"The first requirement when 'undercarriage up' is selected is to open the doors, without affecting the wheel units. In doing this (Fig. 121 (*a*)) fluid from the selector valve enters the connection (A) on the reversing valve, opens the non-return valve (N_1), and crosses the plunger housing by the passage (1), emerging from the connection (E) leading to the upper end of the door jack. As the jack extends to open the doors, the fluid expelled from its lower end enters the reversing valve by the connection (D), passes through the non-return valve (N_6), and along the passage (6) to another passage (3), the opposite ends of which lead to the inlet sides of two non-return valves (N_2, N_4), and flows to the selector valve, from where it is returned to the reservoir.

"As the doors reach the fully-open position (Fig. 121 (*b*)), the piston of the jack lifts the sequence valves (S_1, S_2). Fluid from the selector valve then passes through the valve (S_1), to the piston rod end of the undercarriage jack, causing it to retract and raise the undercarriage. The fluid driven out of the upper end of the jack returns to the reservoir through the sequence valve (S_2), and the selector valve.

"To close the doors after the undercarriage is up, the piston of the undercarriage jack opens a sequence valve (S_3), allowing pressure to displace the piston of a shuttle valve (H); pressure is thus applied at the connection (C) of the reversing valve where it lifts a valve and acts on a face (F) of the plunger. The consequent movement of the plunger closes the passages (1, 6) and opens passages (2, 7), the effect being to cross over the connections between the selector valve and the door jack (Fig. 121 (*c*)).

"Flow then takes place through the non-return valve (N_1), the passage (2), and the valve (R) to the lower end of the door jack, thus closing the doors (Fig. 121 (*d*)). Fluid from the opposite end of the jack returns to the reservoir by way of the passage (7) and the non-return valve (N_4). The operation of raising the undercarriage is now complete and the selector valve snaps into neutral, in which condition both lines between the selector valve and the reversing valve communicate with the reservoir. The spring load can then return the plunger to its normal position, as the pressure previously trapped under the piston (F) escapes as shown in Fig. 121 through the shuttle valve (H), the sequence valve (S_3), and the sequence valve (S_1), which in this direction of flow acts as a non-return valve.

"Operations similar to those just described occur when the undercarriage is being lowered, the effective non-return valves being those marked (N_2, N_3) instead of (N_1, N_4), and the order of jack movements being controlled by the sequence valves (S_1, S_2) and one at the piston rod end of the undercarriage jack.

"Incidentally, the function of the non-return valve (N_5) in the reversing valve is to ensure that the plunger does not return to

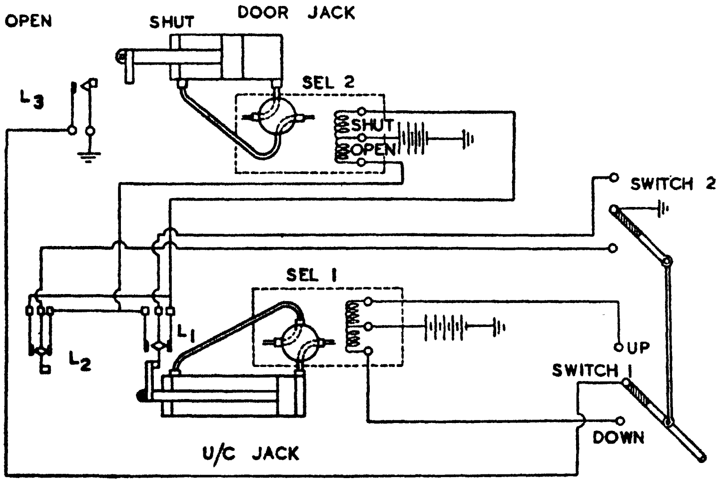


FIG. 122. COMPLICATED SEQUENCE SYSTEM GIVING SEQUENCE: 1—2—1—1—2—1

its normal position prematurely. When the door jack begins to retract, it releases the sequence valves (S_1, S_2), which therefore close; seepage of fluid from under the piston itself as well as past the shuttle valve piston, is made up by fluid passing through the valve (N_5), sufficient pressure for this purpose being developed by fitting the relief valve (R) at this point."

A similar object can be obtained, perhaps more simply and effectively, by the electro-hydraulic system shown in Fig. 122. Both door and main jacks are controlled by electro-hydraulic 4-way selector of any standard type. Assuming that both jacks are closed, the action is as follows: the main switches 1 and 2 are put to the down position. The undercarriage selector is not moved to the down position, because in series with switch 1 is a limit switch $L3$, which will only be closed when the door jack is fully extended. This occurs shortly because the door jack selector circuit is closed through the contacts of limit switch *L*2. When the undercarriage jack has, in fact, extended, the limit switch *L*1 having been thrown over from one contact to another as soon as it began to extend, the limit switch *L*2 is thrown over also, thus reversing the door selector to close the doors again. When "up" is selected, this selector is again reversed by means of switch 2, so that the doors again open, allowing the limit switch *L*3 this time to select the undercarriage "up". The change-over once more of *L*2 and *L*1 closes the doors finally.

The switch *L*3 is obviously normally open, while the inner sets of contacts on *L*1 and *L*2 are normally closed, the others open. Additional limit switching may be necessary if the selectors

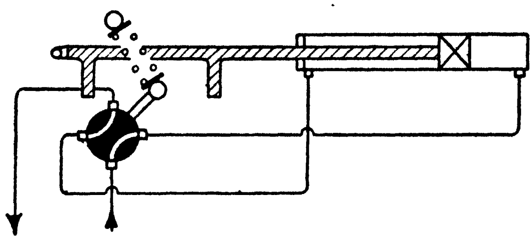


FIG. 123. AUTOMATIC REVERSING SYSTEM

are to be de-energized when a cycle is complete, but this is an electrical problem not affecting the hydraulics.

14.3. Automatic Reversing Systems. Sequence systems which allow continuous or more or less continuous reciprocation of the jack, etc., are used on machine tools, and such devices as hydraulic or pneumatic windscreen wipers.

One particular machine tool system has already been referred to in Fig. 115. This can readily be understood after a description of the basic circuit as shown in Fig. 123. The jack moves in one direction normally, the fluid entering through a selector; when the jack reaches the end of its travel, or in the case of machine tools, a predetermined point in its travel, it moves the selector over so that the motion is reversed. This continues at the end of each stroke to give automatic reciprocation. The special requirement is the addition of a toggle mechanism to the selector, to make sure that the unit opens properly, and also to give the necessary "snap" to the action. The selectors themselves are usually of the slide type suitably adapted for this particular duty, the special type shown in Fig. 48 (*i*), and making use of a hydraulic "toggle" being particularly useful.

To improve the accuracy of the stroking operation on machine tools, such as grinding or honing machines, so that the tool can cut or operate close to a shoulder or wall, electro-hydraulic control is sometimes used, employing limit switches to reverse the flow in electro-hydraulic selectors.

As applied to small units, such as vacuum, pneumatic or hydraulic windscreen wipers, the same principle is used with various forms of in-built selectors. One particular mechanism is shown in Fig. 124. In this the central piston has two plungers of different diameter, and thus works on the differential jack system. Fluid pressure permanently acts on the right-hand plunger urging it to the left and rotating the pinion by the rack, clockwise. As drawn, the central selector plunger admits fluid to the left-hand or larger end, and the pinion will rotate anti-

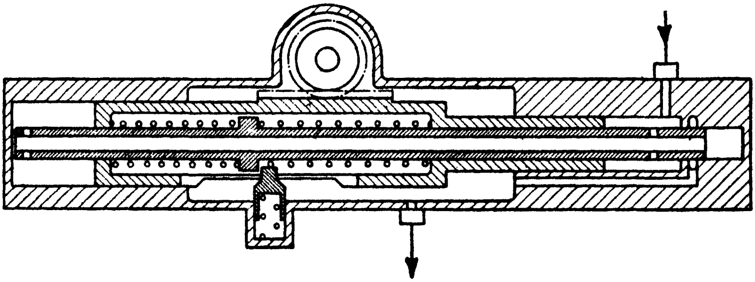


FIG. 124. WINDSCREEN WIPER SYSTEM

clockwise. The inner selector plunger is prevented from moving to the right with the piston assembly (under the action of its spring) until the piston reaches the end of its travel, and the detent is released by a cam on the piston; the selector slide now moves rapidly to the right and connects the larger cylinder to tank, since fluid flows down the central hole and out of the holes in the end corresponding with the tank recess or groove. Thus the action involves the continuous compression of one or other spring and the release of the detent from either side of the collar on the slide. The same mechanical principle obviously can exist in many variations, not necessarily internal devices, although the use of a differential piston is advantageous as simplifying the selector mechanism (i.e. a 3-way instead of a 4-way selector).

Another solution is to use what might be called a pressure detent. One method is shown in Fig. 125. As drawn the jack closes under pressure; when it reaches the end of its travel, the pressure rises, moves the left-hand annular piston against its spring and eventually overcomes the ball-and-recess, or equivalent, detent, allowing the central valve assembly to move to the right and engage the right-hand detent. This reverses the flow, the valve acting as a 4-way selector, the force to move it being provided by the compressed spring, the annular pistons having a suitable abutment.

The action is thus continuous, the change of direction occurring each time the pressure rises. If the pressure rises in mid-travel for some reason or other, the reciprocation takes place at once; this has an advantage in the case of a wiper acting on a snow-covered windscreen, as the wiper arc can gradually get less, instead of the unit stalling.

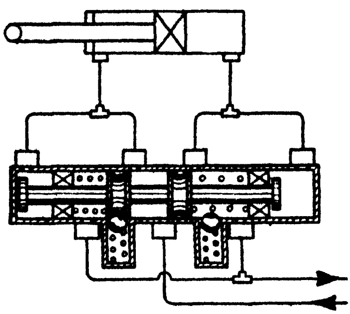


FIG. 125. AUTOMATIC REVERSING VALVE USING DETENT

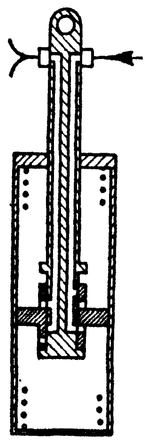


FIG. 126. JACK WITH DELAYED AUTOMATIC REVERSING VALVE ARRANGEMENT

The jack incorporating its own selector illustrated in Fig. 126, may perhaps be classified as a delayed automatic reversing system, as will be apparent from the description. The unit is designed for operation by vacuum, but could be adapted for normal pressure use. When the vacuum selector valve is opened, the piston and rod are sucked one way (closed as drawn) and eventually, after compressing the spring, the rod abuts. The piston continues to compress the spring as long as the vacuum is maintained. If, however, it is released for a moment, the spring moves the piston along the rod and changes over the 4-way slide selector formed by it and the rod end. When the vacuum is next applied the jack moves to the other position. This system is thus an automatic reversing system, but delayed at each end of the stroke until a cock is opened momentarily. Such a system has very limited application.

CHAPTER 15

SERVO SYSTEMS

SERVO systems are those where the motion of a relatively lightly loaded transmission system is reproduced with a degree of force multiplication, usually considerable, at the receiver. In other words, a small effort is boosted into a large one at the receiver. (The expressions "power assisted control," "boost control," "booster," and "follow-up control" are used as well, but the better title is probably the "servo-control".) Hydraulic systems are generally used, but pneumatic types are known too.

Although systems using a controlled variation of pressure or force to effect the required servo action are known (e.g. brake system—see Chapter 7), this chapter refers exclusively to that type of servo where the movement of the pilot control or transmitter is exactly reproduced at the control receiver, with a more or less fixed degree of force multiplication.

Servo systems are not new inventions.* They have been used to effect hydraulic power steering of ships and steam turbine governor control for many years. Equivalent electrical servo systems have been known for the remote control of guns for almost an equal period. In recent years the power boosting of heavy vehicle steering gear in America has become common, using both hydraulic and pneumatic systems, while servo vehicle brake systems (although often not of the type under discussion) have also been widely used throughout the world.

There are two main types of fluid pressure servo system, one is the so called "on-off" or "error operated" type and the other a balanced circuit type. The first is the more usual and consists of a hydraulic valve which is moved "on" or "off" with a small "error" to effect the necessary power multiplication. The second to be described first relies on the flow of fluid continuously in a "Wheatstone bridge" type of circuit using, perhaps, a selector

* The first application of the hydraulic servo or follow-up principle is probably that used on the "Great Eastern" ship and described by Gray in 1867 in a paper before the Institution of Mechanical Engineers†; here the helmsman turned a screw connected to his steering wheel, the screw being carried in a nut formed inside a bevel gear connected by shafting to the rudder. When the screw was turned the axial motion in the nut opened a steam valve to operate the rudder engine. The follow-up action due to the nut rotating brought the screw back to its original position.

† *Proc. I. Mech. Eng.* 1867, page 267.

generally similar to that illustrated in Fig. 48 (*i*) or by arranging the circuit as in Fig. 79 (*a*).

15.1. Balanced Circuit Servos. A typical arrangement of the balanced circuit is used in automatic honing machines. The four resistances consist of two fixed resistors, a variable resistor actually calibrated in terms of desired size, and a fourth variable resistor controlled in size by a small lever touching the part being honed. Across the bridge is a diaphragm unit; the diaphragm will be responsive to unbalance in the circuit and will be urged one way or the other, depending on whether the size resistor is

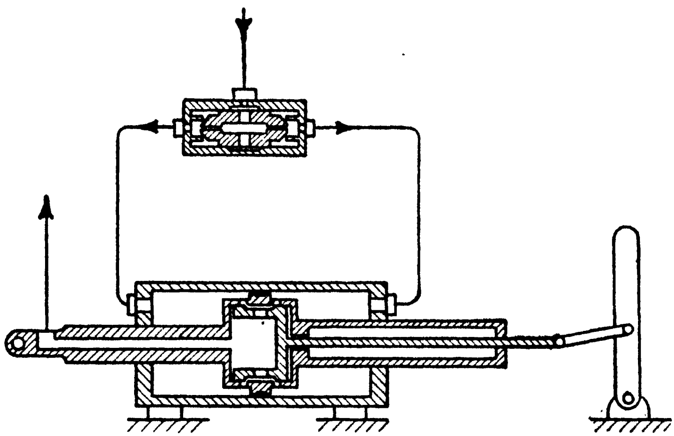


FIG. 127. A BALANCED CIRCUIT SERVO SYSTEM

indicating that the size of the work is larger or smaller than the calibrated or desired size resistor. The diaphragm is made to operate a micro-switch to apply means of adjusting the hones until the size indicated reaches the desired size, the circuit balances and the micro-switch opens.

There are many other applications of similar devices on this principle mainly using air escaping past a variable orifice operated by the machine process or device being controlled. Fig. 148 illustrates a typical application where a balanced circuit might be used although in the case illustrated a simple system is used.

A hydraulic servo system based on balanced flow is shown in Fig. 127. This uses a single pump with flow split in a flow divider, or could use two equal pumps; the branches are connected to either end of the jack, the central valve allowing fluid to escape with a pressure drop back to tank. The valve takes up a central balanced position and as it is moved the circuit becomes out of balance moving the piston of the jack to follow upon the displaced valve member. The resemblance to the Wheatstone circuit will again be at once obvious, the flow divider being equivalent to two fixed resistances, the variable resistances being in the valve member. In place of a fixed flow divider, a selector element similar to that shown in Fig. 48 (*i*) but with a slide of constant diameter may be used. This gives control of a jack by simultaneous variation of all four resistances.

Extraordinary accuracy of reproduction with valve devices of this type on machine tool hydraulic servos has been achieved. Errors appreciably less than 0·001 in. between valve member and jack can be easily obtained.

15.2. Design Problems with "On-off" Servos. The main problem which should be considered at this stage is whether the control shall have "feel" (i.e. be reversible) or be irreversible. In other words, shall the effort exerted by the operator be that required to overcome—

(*a*) The friction of the control valve, etc.;

(*b*) The opening load of an unbalanced control valve;

(*c*) The force exerted by a deliberately inserted reaction or "feel" spring;

(*d*) Some fixed fraction of net reaction or equivalent load on the controlled unit or receiver.

Friction to some degree or other is certain to be present, but with good design it can be very small; it is usually substantially constant and independent of the force exerted and is thus more important at light load.

As has already been mentioned, certain selectors or control valves, particularly poppet valve types, require a given force to overcome the fluid pressure load on the valve, but once open this force may drop. This gives an undesirable characteristic to the control if the initial load is high. If small it is not unpleasant to handle, since it eliminates excessive sensitivity to small movements of the operator's fingers.

Reaction springs are used commonly in brake control gear to give artificial feel, and are used in some other servo systems as well. It seems easier to use the genuine feel effect obtainable by allowing the control to exert a direct action of a certain fraction of the total effort, particularly since the true feel may not have the linear characteristics of a spring.

The second design problem may be to devise a method of over-riding or disconnecting the control in case of failure of the fluid system. This can be done by—

(i) Open-circuiting the control jack;

(ii) Short-circuiting the jack;
(iii) Disconnecting the jack mechanically from the receiver mechanism;
(iv) Complete duplication of the control.

Either open- or short-circuiting the jack is the simplest method; in the former both sides of the jack are connected to tank, the whole system idling under zero pressure; in the latter the pressure on both sides of the jack becomes the same, not necessarily zero —this of necessity demanding a symmetrical jack. With both these systems gland friction in the jack will cause a not inappreciable operating friction which may be unpleasant. In addition, the presence of oil in the jack makes it act as a hydraulic damper and causes further restriction of movement.

Disconnection of the jack completely is another method possible where the jack is not part of the control system proper, although requiring extra space and mechanism. The disconnection can be effected automatically by loss of pressure or by deliberate selection by the operator.

Complete duplication of the control is used on some aircraft systems, where the extra complication is justified.

A third problem, and one which should be considered with the earlier problem just referred to, is the possibility of a single failure (pipe, etc.) causing the control to move, even momentarily, in a particular direction; this might be referred to as a defect unbalancing it. In certain applications this would have more or less catastrophic results if the unbalancing was serious or lasted an appreciable time. Any control system should be studied for safety under such conditions.

15.3. Basic Mechanisms. The basic "on-off" servo jack mechanism is shown in Fig. 128. It will be noted that it comprises a jack, a selector valve controlling it, a connection from the operator to the selector, another connection from the controlled device, etc., to the selector, and a differential lever mechanism or its equivalent between these two control runs and the selector. It is important to note that a remote control run must exist between the controlled device and the operator, although part of the run deals only with relatively light loads.

The action is simple enough and well known. When the operator moves his lever, the jack, which is full of fluid and hydraulically or pressure locked by virtue of the selector valves being closed, remains fixed and the differential lever pivots about it, opening the selector; this in turn admits fluid to the jack to move it in a direction which moves the differential lever until it is again in the balanced position. It should be noted that the ends of the lever move laterally the whole travel of the jack, and that the operator's control run has the same travel as the jack. The travel of the selector lever is half this, assuming the differential lever is divided centrally, and this means that it must have considerable over-travel, since the first fractional motion must open and close the various valves in it. In the case of a hydraulic servo and ignoring fluid compressibility, the backlash of the operator's control and the positional accuracy of the jack are some slight multiple of (depending on leverage, but probably twice) the selector backlash.

In this simple system the operator can move his lever the whole of its travel while waiting for the jack to follow up; if

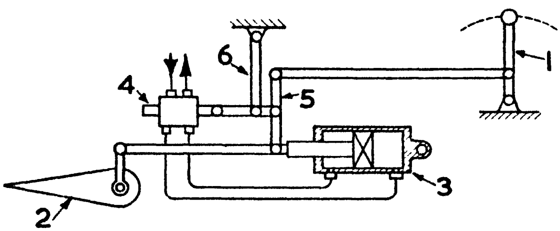


FIG. 128. THE BASIC "ON-OFF" SERVO MECHANISM

1. Pilot's lever.
2. Rudder, flap, etc.
3. Jack.
4. Control selector.
5. Differential lever.
6. Locating arm omitted from later diagrams for simplicity.

the pressure source has failed he has no direct indication that the jack is not moving. This system is used in the so-called preselective system, enabling the operator to select any jack position quickly, the jack moving at a predetermined rate to the chosen position. For aircraft flying control, or vehicle steering or brake boost systems, it is usual to limit the travel of the selector so that the operator can only get out of phase by a small amount. If he continues to exert a force on the control his hand or foot moves at the speed that the hydraulic system will allow; if the system is inoperative he meets resistance and is thus warned.

FEEL. In the system of Fig. 128, the operator must exert an effort which is sufficient to overcome the selector load only (generally half of it due to the differential lever). The true reversibility or feel effect, referred to above, can be obtained in two ways; by fluid pressure or mechanically. Fig. 129 (a) shows the fundamental method of obtaining feel by pressure, the degree being a function of the diameter of the feel cylinder; in practice the feel cylinder is incorporated inside the (slide) selector. (See Fig. 48 (*f*).)

A much better and simpler method of obtaining feel is by direct mechanical leverage, as illustrated diagrammatically in

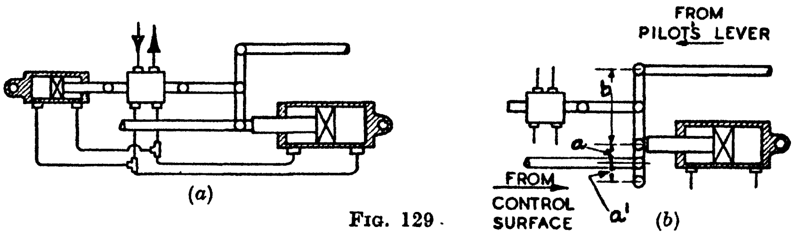


Fig. 129.

(*a*) = Hydraulic "feel" cylinder. (*b*) = Lever disposition to obtain mechanical "feel".

Fig. 129 (*b*). The small offset "a" causes a reaction in the operator's control of b/a times the output rod reaction, i.e. a fixed percentage feel. Incidentally, if "a" is negative as at "a′", the control is overbalanced and has "negative" feel, although this does not appear to have any practical interest.

Other Mechanisms. Fig. 130 shows another variation of Fig. 128 with the lever arranged differently. In this case the

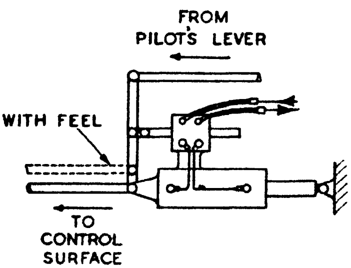


Fig. 130. Another Servo Jack Mechanism

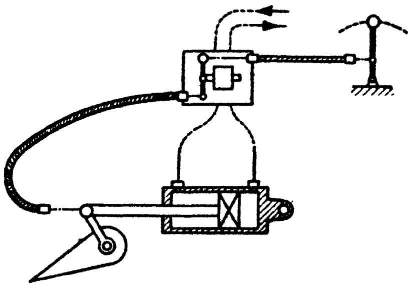


Fig. 131. Remotely Controlled Differential Lever

selector moves with the jack and thus laterally from side to side over the whole travel of the jack. Instead of the centre of the lever having a fixed balance centre, as in Fig. 128, the balance position is obtained when the lever reaches a particular orientation, e.g. vertical. This has the advantage that the selector can be fixed to, and integral with, the jack, thus saving piping, but it requires flexible connections to the power system owing to the movement involved. The simple method of obtaining feel will be noticed. Fig. 131 shows another useful variation, where both connections to the differential lever mechanisms are by remote controls, which obviously need not be mechanical, and in fact are often electric. This system, while inherently less accurate than the more direct methods, due to the remote controls, is interesting for some applications.

Fig. 132 shows what might be considered as a particular case of Fig. 130, where the lever has zero length, since the valve is mounted direct on the jack rod or cylinder. This type of servo jack is, of course, similar to that shown in Fig. 127 with a different arrangement of porting. It is more logical, however, to consider it as a particular case of the ordinary lever servo with the lever of zero length.

ALTERNATIVE DIFFERENTIAL MECHANISMS. All the well-known differential lever variations can be used, such as the bevel or

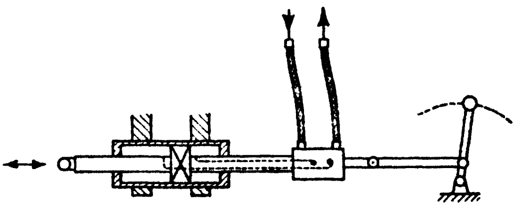


FIG. 132. DIRECT OR LEVERLESS VALVE MOUNTING

spur gear differential, cams, etc. Complete elimination of the lever is possible by moving the selector lever on the one hand, and the body of the selector on the other, thus obtaining the necessary differential motion. This is what is done in the system illustrated in Fig. 132, and the one referred to in Fig. 133 below.

15.4. Selector Valves. The controlling selectors used with on-off servo systems may be of any type, but actual solutions have used (*a*) rotary, (*b*) poppet, or (*c*) slide valves. Rotary face or barrel valves are not adequate for precision applications, since it is too difficult to obtain the necessary pressure tightness with the very small overlap between one port closing and another opening, which is necessary to obtain accuracy. They have been used on steering gear boosters and because they make the elimination of the differential lever relatively simple (see the preselective system of Fig. 133 below). Poppet valve selectors suffer from the disadvantage of opening load, already referred to, but this can in practice be overcome if the valves are small enough. Most valve solutions require car-tappet-like adjustment, so that valve backlash can be kept as small as possible, and to enable the bedding-in of the valves to be allowed for during maintenance, although at least one solution overcomes these difficulties. The poppet valve is relatively cheap, and a good solution for simple applications, although the provision of large over-travels is generally an inconvenience.

Slide valves are certainly the best for precision work as they are very light to operate, fully balanced hydraulically, and can

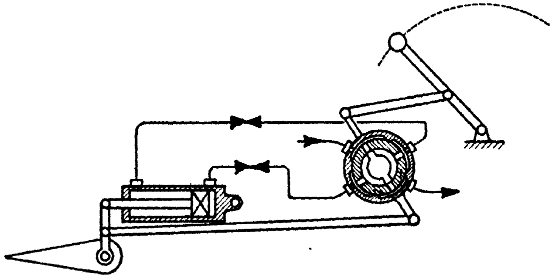


FIG. 133. ROTARY FOLLOW-UP SELECTOR

easily be provided with over-travel. For accurate motion without excessive lost motion, the overlap between one port closing and another opening must be very small (of the order of 0·004–0·005 in.), and this requires very accurate workmanship, being thus costly. As already mentioned on page 58 certain sensitive servo systems use slide valves with an intermediate sleeve

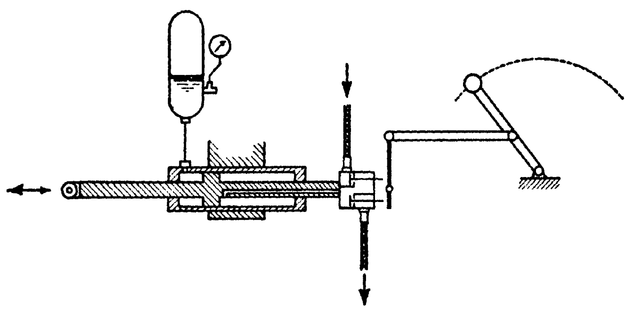


FIG. 134. SERVO JACK WITH ACCUMULATOR RETURN

between the plunger and casing, the sleeve being made to rotate or oscillate to eliminate "stiction" and reduce friction.

With certain servo systems provision has to be made for ensuring that the operator's lever stays in a given position when selected, since if the selector is stiff or requires appreciable force to keep it open, the lever if let go may move back instead of keeping the selector open. This is important for systems without feel, but if feel is used the lever obviously tends to reverse anyway.

Certain designs of control use a fixed amount of lost motion to operate the selector—see, for example, Figs. 135, 136, 139, 140 below. It is generally unnecessary to introduce this complication, and the simplest of the various systems do not use it.

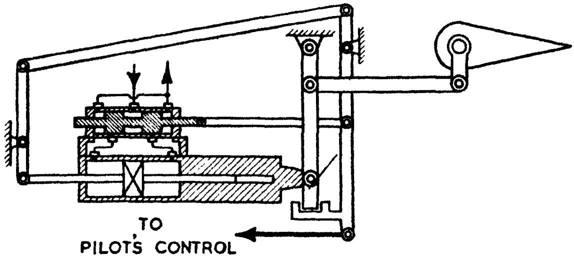


FIG. 135. AN AIRCRAFT FLYING CONTROL BOOSTER

In the case of steering boosters, the end thrust on the worm of a worm and wormwheel steering gear box has been used to move the slide valve.

15.5. Typical Solutions. 1. An aircraft preselective flap system is shown diagrammatically in Fig. 133. A rotary follow-up valve is used, the rotor being connected to one member (the pilot's lever) and the casing (actually an intermediate ring with an outer body arranged to admit and receive the fluid from the system) is connected to the other (the flap). This system is exactly

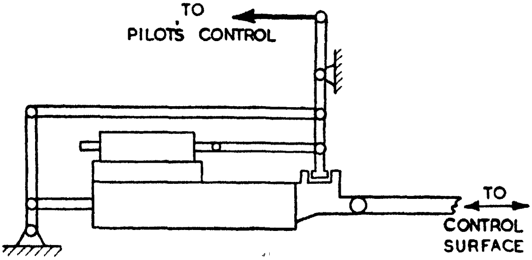


FIG. 136. SIMPLIFICATION OF THE MECHANISM OF FIG. 135

equivalent to that of Fig. 127 and has the same advantages and weaknesses.

2. Another aircraft flap system is shown in Fig. 134. Although this was used with an accumulator to lower the flap, the pump pressure being connected to one side of the jack only, it is strictly equivalent to the system of Fig. 132. Owing to the limited travel of the valves the system is not preselective.

3. A flying control booster system developed for a large aircraft

is shown in Fig. 135. This appears excessively complicated and it is a fact that it is kinematically equivalent to that shown in Fig. 125. The method of limiting the selector travel, and the feel or feed back connection will be noted. Although Fig. 135 represents diagrammatically the original mechanism, Fig. 136 shows the system re-arranged in a simpler fashion.

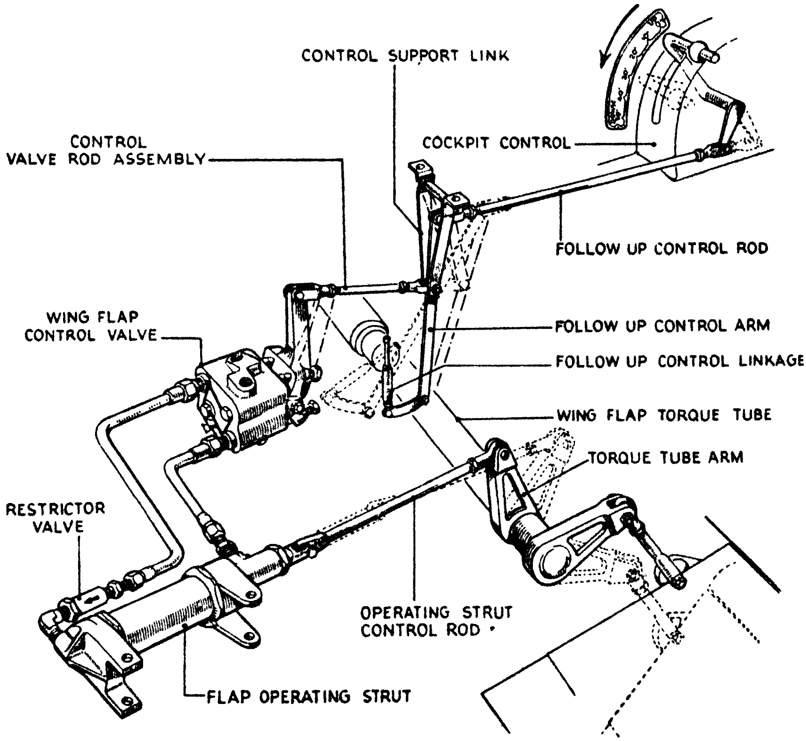


FIG. 137. AN AIRCRAFT FLAP SERVO SYSTEM

4. A well-known fighter aircraft is fitted with an excellent flap system, using the preselector arrangement of Fig. 127, although arranged with five or six notches on the lever gate at regular intervals over the lever travel; these are needed to hold the lever in position. This system is shown in Fig. 137, and will be seen to have a poppet valve selector with adequate valve clearance adjustment.

5. Another aircraft aileron booster unit is shown in Fig. 138, and is a unit of some complexity in comparison with other systems. The diagram, which is much simplified from the actual unit, shows that the differential lever is connected in

an unusual sense with the valve on one side and one control connection in the middle, instead of the valve in the middle. This has no mechanical significance except that the travels of the two control run points on the lever are not the same to rebalance the lever once this is moved. Also the feel, quoted as

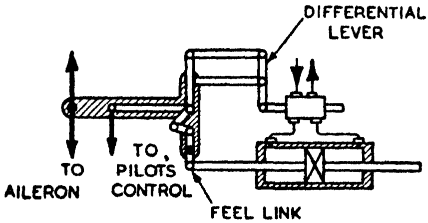


FIG. 138. AN AIRCRAFT AILERON BOOSTER SYSTEM

15 per cent, is obtained by a separate set of links connecting the jack to the aileron crank. This design appears to involve kinematic complexity for no particular reason.

6. A further aircraft solution is shown in Fig. 139. This uses a hydraulic feel cylinder externally or built into the selector as illustrated in Fig. 128. The scissors mechanism, allowing a degree of lost motion to operate the valve, and at the same time acting as a differential lever, will be noted.

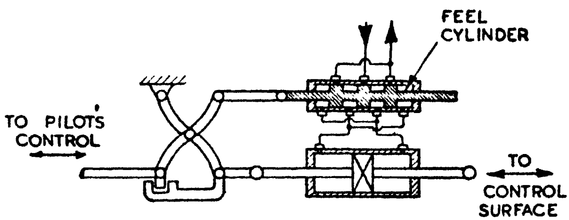


FIG. 139. AN AIRCRAFT BOOSTER WITH HYDRAULIC FEEL

7. The solution illustrated in Fig. 140 is similar mechanically and in complexity to the unit of Fig. 138.

8. The system shown in Fig. 141 uses the differential area principle of jack connection, the pump pressure being applied on the annular area of the jack to close it, or to both sides of the piston at once to extend the jack, the effective area on extension being that of the piston rod. This is convenient because the selector is simpler (a 3-way instead of a 4-way unit), and because the effective areas of the jack can be the same in both directions without using a through piston rod. The unit uses a slide valve with a declared port overlap of 0·005 in.

9. Another known aircraft system is the same as that shown in Fig. 132, the valve in this case is a 4-way 4-valve poppet

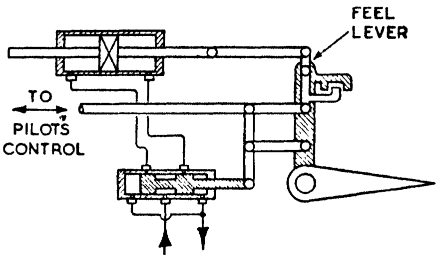


FIG. 140. ANOTHER AIRCRAFT SYSTEM

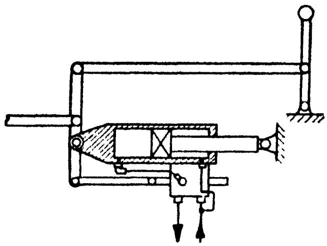


FIG. 141. A SERVO JACK SYSTEM USING THE DIFFERENTIAL AREA PRINCIPLE

selector generally similar to the unit shown in Fig. 137. Since the valves are small the opening load is small, and the performance of the unit is satisfactory from this point of view.

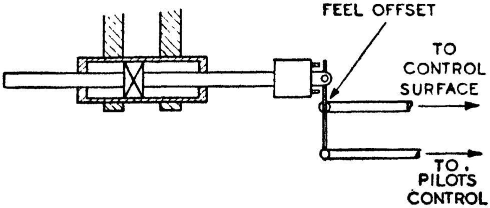


FIG. 142. ANOTHER DIRECT JACK SYSTEM

Feel can be obtained by mounting the jack on a reaction lever linkage as shown in Fig. 142.

10. Another flying control booster (Fig. 143) is a straight-

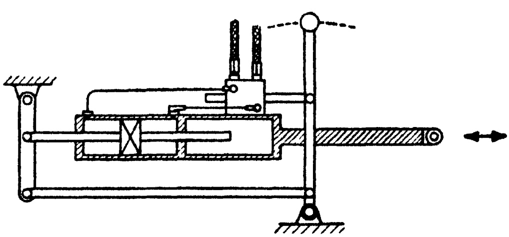


FIG. 143. ANOTHER AIRCRAFT BOOSTER SYSTEM

forward control of the type shown in Fig. 130, using a slide selector with a declared 0·002 in. port overlap. It incorporates an ingenious disconnecting device to enable the control to be uncoupled by pushing a knob mounted close to the pilot. The means for disconnection consists essentially in a telescopic strut locked in the normal position by a pin which can be withdrawn by pushing the knob of the selector referred to. Stops on the selector enable the pilot's effort to be transmitted directly. In the diagram the telescopic strut would be fitted in an extension

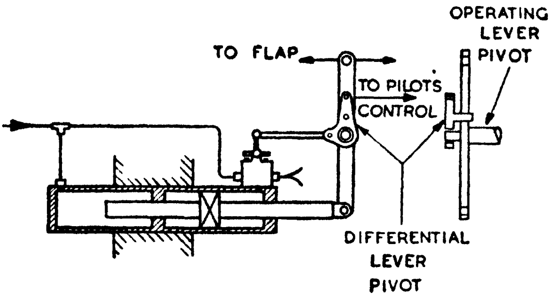


FIG. 144. A PNEUMATIC SERVO SYSTEM USING A REVERSED DIFFERENTIAL AREA JACK

of the left-hand end of the jack piston rod between it and the swinging lever.

11. A pneumatic aircraft flap system is illustrated in Fig. 144. It is feel-less and since the amount of free travel to operate the valve is limited, it is not preselective. The differential lever is carried on the main operating lever, and uses an enlarged hole co-operating with a pin on the latter to obtain the free travel

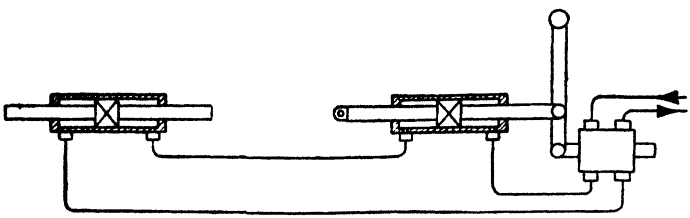


FIG. 145. A SERVO SYSTEM IN CIRCUIT WITH A REMOTE CONTROL SYSTEM

conveniently; this device is used on the well-known vacuum brake servo for commercial road vehicles.

12. Another system which is basically similar to other types is shown in Fig. 145. This uses a hydraulic remote control between the "transmitter" lever and the output "receiver", but connected in series with a selector coupled to the operator's lever differentially. The usual priming and synchronizing valves of a double-acting remote system, as described in Chapter 12, will,

in practice, be included. This system can be considered as a normal servo jack coupled to the device being controlled by a double-acting remote control, but with one jack of the latter combined with the servo jack.

13. V servo valve element, suitable for air or oil, is shown in Fig. 146. In this case the control valve is mounted in some part of the mechanical rod (etc.) system coupling the operator to the unit being operated, and it thus transmits the whole unboosted effort mechanically. The operator's effort acts on a small slide valve in the selector, and when the load exceeds the initial tension of the spring, the valve opens and admits pressure to the booster jack; the total travel of the valve is small and will not be noticed by the operator. The initial tension of the valve spring can be varied to give any desired artificial feel characteristics, without the true progressiveness of genuine reaction feel; this, however, may be an advantage. Servos on this principle are well known for vehicle braking.

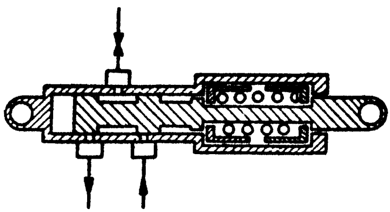


FIG. 146. A PNEUMATIC SERVO VALVE WITH ARTIFICIAL FEEL

14. A booster system, this time using a variable delivery pump, is shown in Fig. 147. Although designed for aircraft the principles have been, and can be, applied to other applications.

The system is applicable to all three flying controls, aileron, rudder and elevators, and consists essentially of an electrically driven variable delivery pump and a servo jack and necessary linkage for each of the three controls. It includes a fourth hydraulic circuit operating "feel" jacks or cylinders, and an automatic locking system.

Each pump, except the "feel" pump, is of the radial variable stroke and reversible type as used for many years on ships' turrets.

The mechanism of the system is illustrated in the central or neutral position. The lock lever (1) engages the differential lever due to the spring (2), and the hinge point (3) remains fixed, allowing the pilot's effort to be transmitted directly through the levers (4) and (5). At the same time fluid in the jack (6) is allowed to by-pass from A to B, etc., since the valve (7) is in the open position due to its spring.

Should the pilot wish to use the servo system, he switches on the power supply through the switch (8). Then the pump (9) operates the various safety devices, the other pump not yet

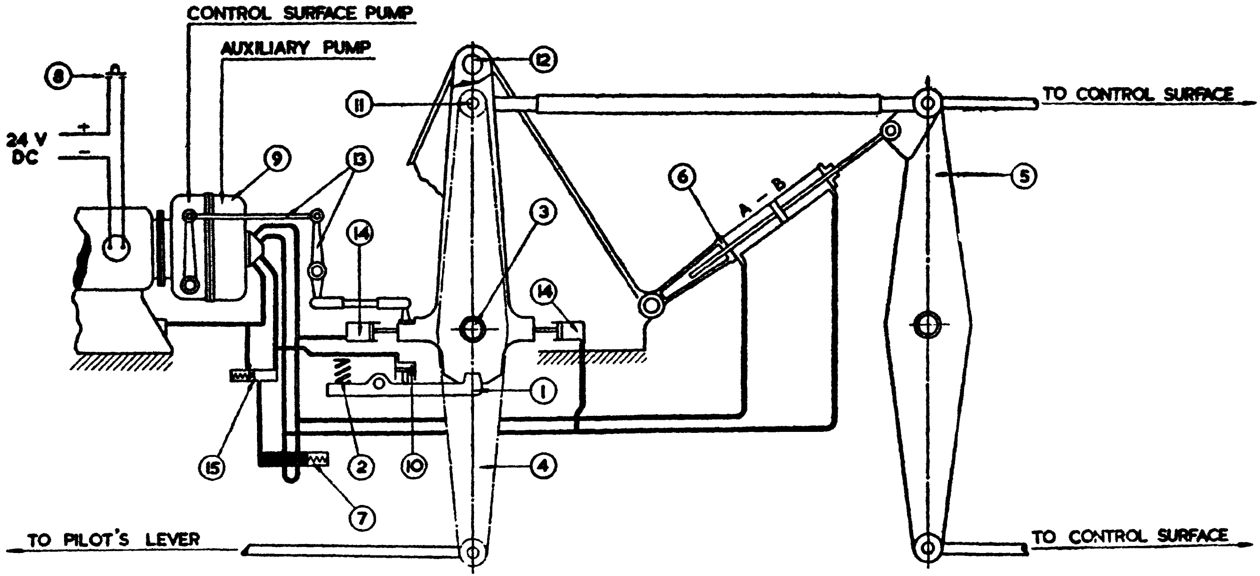


Fig. 147. Servo System using Variable Delivery Pumps

operating since the control is in neutral. The small jack (10) unlocks the differential lever pivot, and the valve (7) closes the short-circuiting of the jack (6). The control lever is now locked hydraulically by the oil in the jack.

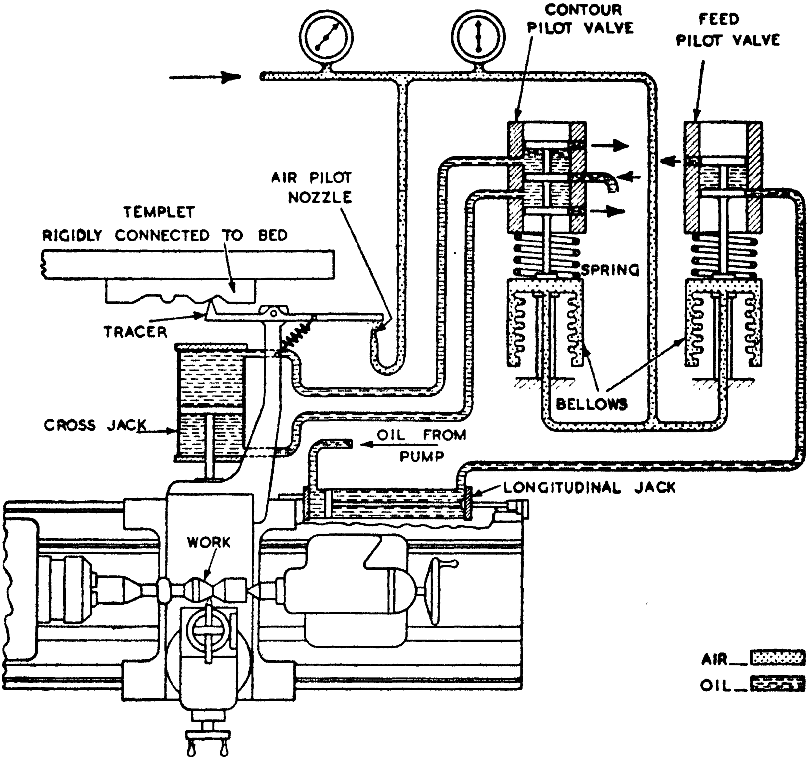


FIG. 148. HYDRO-PNEUMATIC MACHINE TOOL COPYING SERVO

When the pilot operates a control, the point (11) being fixed, the main lever rotates about the pin (12), causing the pin (3) to move laterally. The linkage (13) now causes the eccentricity of the pump to change from its central position, starting the delivery of fluid to the main jack and the feel cylinders (14). Delivery continues until the jack has moved the control surface over until the lever (4) is again in the balanced position once more cutting off pump delivery. At the same time the feel cylinders cause a small predetermined reaction on the pilot to give him some sensation of the effort exerted by the servo.

During the running of the power system the safety valve (15) maintains a constant pressure in the auxiliary circuit. Should the power supply fail for some reason the springs (2) and (7) relock the differential lever and by-pass the main jack again.

15. Another interesting application of a hydro-pneumatic servo system, as applied to a copying mechanism on a machine tool (for die sinking, for example), is shown in Fig. 148. As illustrated, the mechanism is applied to profile turning on a lathe. Pressure from a constant source is allowed to escape past a small bleed varied in size by one end of a differential lever, pivoted to the cross slide and acting on a profile templet. The air pressure in the pipe line is thus controlled as described on page 127, and reacts on two bellows to counterbalance the springs in the two slide type selectors. The cross slide is operated by a double-acting jack and the 4-way selector, the feed by a second jack, the escape of oil from which, and thus the feed, is controlled by the 3-way selector. When motion is started the feed takes place at a certain rate through the feed selector, and the cross-slide advances until the differential lever balances between the contour templet and a degree of opening of the restrictor or bleed hole which will produce enough pressure to balance the 4-way selector and stop the cross slide; thereafter the cross slide follows the templet. The second selector is not necessary, except that as the cross slide goes in and out the depth of cut will vary and thus the work done by the tool also; more metal will be removed when the pointer is rising up an incline on the templet (reducing the air pressure) than going down one (increasing the air pressure), and thus decrease in air pressure is made to reduce the feed also.

CHAPTER 16

ELECTRIC CONTROL SYSTEMS

MANY hydraulic or pneumatic systems use electric control of parts of the system, usually to facilitate the control itself and to replace long pipe runs with electric wires. The most elementary electro-hydraulic or pneumatic system is in the use of an electric motor to drive the pump, with, for example, a pressure switch to effect the necessary cut-off. This is commonly seen in the normal garage compressor set. The systems to which this chapter refers, however, are more complex.

A good example of the advantageous use of electricity is in an electro-hydraulic system where the selectors are solenoid or motor controlled, with a simple wiring system to a remotely placed switch. This enables the low power control action to be electrical, the high power action being hydraulic, with great economy of pipe, fluid, and often with a gain in transmission efficiency, when the overall fluid pipe runs are reduced. Where, however, a hydraulic or pneumatic relay system is used, there is usually little to be gained by the use of electricity instead, from the point of view of friction losses, etc., in the pipe runs.

Where complicated sequence systems are required (as already described in Chapter 14), electric control is recommended, since switches, relays, etc., are to be preferred to a multiplicity of sequence valves and piping. The relatively complicated actions set in train by the press-button switches on modern lifts—which enable the successive stopping of the lift to be pre-selected by pressing several buttons, enable the buttons to be illuminated when the selected action is commenced, enable the lift to descend automatically after a time lag, etc.—all these are possible by purely hydraulic control, but would then be impracticably complicated. The purely electric problems of such systems are not within the scope of this book, but the main elementary pressure circuits will be described, circuits which can be combined with electric control to give any desired performance.

Another application of electric control is in servo systems (see Chapter 15), where the original source of power, which is to be multiplied by the servo system, is purely electric. Examples are in the magnification of the current output of a photo-electric cell, so that the movement of some mechanism could be controlled in response to the amount of light falling on the cell; the governing of steam turbines or Diesel-electric sets by means of the current output of a small tachometer-generator which is used to control a small hydraulic valve, and thus a powerful jack to move the

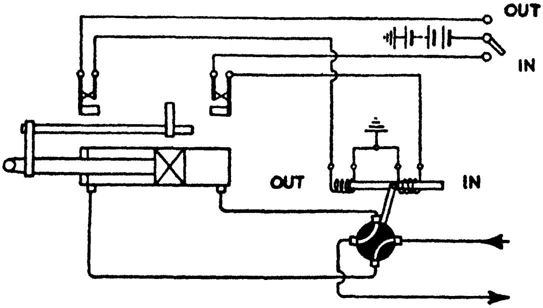


FIG. 149. ELEMENTARY ELECTRO HYDRAULIC SYSTEM WITH LIMIT SWITCHES

steam or fuel inlet, etc.; or thermostatic control by the servo magnification of the current flowing through a pyrometer.

Any pressure system using a pressure remote control, whether of the servo type or not, can generally have the pressure control part replaced by an electrical equivalent; this chapter is, therefore, not exhaustive, and should be considered with Chapters 12, 14 and 15.

16.1. Elementary Systems. The most elementary system is shown in Fig. 149, and is a means of controlling the reciprocation of a jack by an electro-hydraulic or pneumatic selector. If the selector is continuously rated and can be left switched on the whole time, it has simply to be connected to a single-pole changeover switch. As, however, this is unlikely, limit switches are generally used to switch off the current when the jack reaches the end of its travel, as shown in the circuit illustrated. These limit switches are often made adjustable so that the jack stroke can be varied as on machine tool applications.

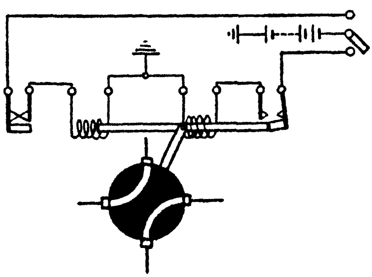


FIG. 150. SELECTOR WITH LIMIT SWITCHES

Instead of using limit switches on the jack, it is possible to use them on the selector as shown diagrammatically in Fig. 150; this is particularly suitable for a motor-driven selector, since the selector will be irreversible and will stay where selected when de-energized.

Another elementary method is to use a "toggle" type of selector which requires an impulse of current only to move it

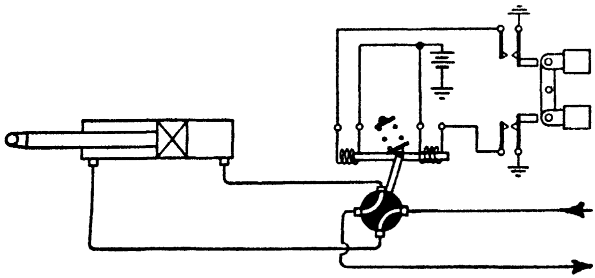


FIG. 151. IMPULSE-OPERATED SELECTOR

to a desired position, staying there for mechanical reasons. This system is shown diagrammatically in Fig. 151, using push-button switches, which must be depressed for the short time required to allow the selector to move. It may be necessary to use a

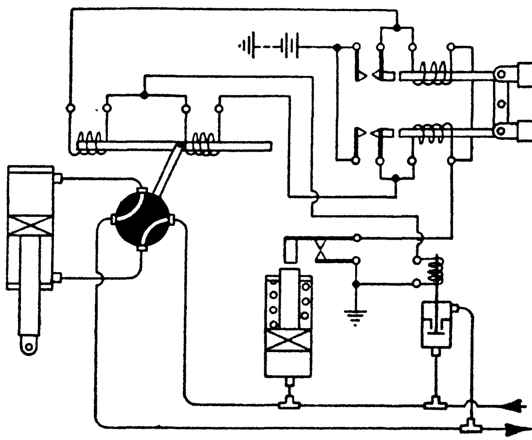


FIG. 152. THE HOLD-ON SYSTEM

thermal or time lag protecting circuit breaker to prevent damage to the selector windings if a button is held down too long.

With all these systems it is a simple electrical problem to stop and start the pump each time the selector is operated, if desired.

16.2. The Hold-on System. Another system uses electrical means to hold the switch closed when once pushed, releasing it only when the motion is complete and the pressure rises to cut out the pump. Fig. 152 shows this system. The action is as follows: when a button is pressed, the circuit is made from battery, through the switch contacts involved to the selector, through a solenoid coil in the pump cut-out to earth. Thus the selector is moved and the cut-out by-pass cock closed simultaneously, enabling the pump and jack to operate. At the same time, however, a circuit is closed through the pressure switch contacts to the hold-on coil in the press button, energizing it and keeping the button down when the finger is removed (obviously it may be illuminated simultaneously). When the jack reaches the end of its travel, the pressure rises and eventually the pressure switch contacts open, de-energizing the switch, which jumps out, and also opening the pump by-pass cock to unload the pump.

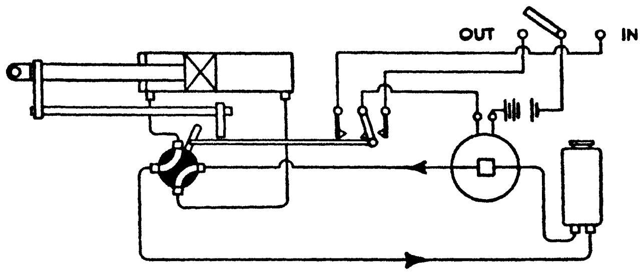


FIG. 153. DELAYED AUTO-REVERSING SYSTEM

This also enables the pressure switch to close again ready for the next selection.

In place of hold-on coils on the switch proper, the same effect can be obtained by holding relays in the circuit.

16.3. Delayed Auto-reversing System. A known system of electric control, which avoids the use of an electrically driven selector, is shown in Fig. 153. In this the selector is controlled by the motion of the jack as for an automatic-reversing system (see Chapter 14), and would normally reciprocate automatically. However, the selector is coupled to a single pole change-over switch (obviously by some cam or toggle action, not shown, to give a snap action), which is wired in series with the pump driving motor and the main selector switch. Instead of the jack reciprocating automatically the motor is cut off and the motion is stopped, but the selector is left in the correct position so that when the switch is thrown over for motion in the other direction, the motor starts up and the selector does not have to be moved. In other words, the selector is pre-set at the end of each stroke. This system is admirably simple, but has the possible disadvantage that the motion cannot be reversed in mid-travel. If the main switch is opened in mid-travel the action stops, but cannot be reversed.

16.4. Multi-position System. Electrical switching means is the easiest method of stopping a jack at intermediate points in

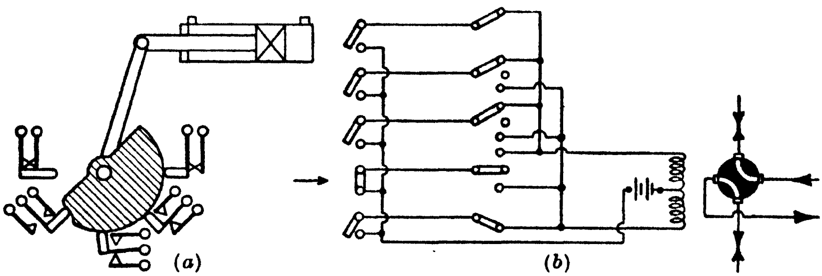


FIG. 154. MULTI-POSITION SYSTEM
(a) = Switch cam arrangement. (b) = Switch wiring circuit.

its travel, without the use of multi-travel jacks or other means described in Chapter 4.

Fig. 154 illustrates a method of achieving this object. Fig. 154 (*a*) shows a jack operating a cam mechanism and five switches. It will be noted that the end switches are normal limit switches, but the intermediate three (corresponding to three intermediate positions) are single pole change-over types. The cam is arranged to change these switches over from one contact to another, with an intermediate open-circuit position possible during the change. Fig. 154 (*b*) shows the electrical circuit. The operator selects closed one of five switches (e.g. by a rotary switch); an electrically operated selector is energized to start the jack moving, and, therefore, the follow-up switch gear just described. The jack stops when the corresponding follow-up switch is reached by the cam, and is open-circuited in the mid-position between its two sets of contacts (or simply opened at the end of the travel). The switch circuit shown is required because it is necessary for the selector to know which way to move when moving from an intermediate position, and thus the switch before, and the one after the one already in use, on the follow-up switch gear, must be "preselected" automatically to make this possible.

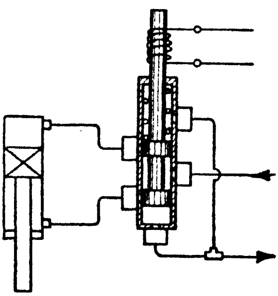


FIG. 155. SOLENOID OPERATED GOVERNING SELECTOR

16.5. Miscellaneous. As already mentioned, electro-hydraulic systems are often applied to governor or other control gear on turbines and other types of engine or motor. Chapter 15 has already dealt with the main aspects of this type of servo control. Apart from the obvious step of using an electro-hydraulic selector operated by a switch, itself controlled by the follow-up lever, or mechanism, as already described in that chapter, one known

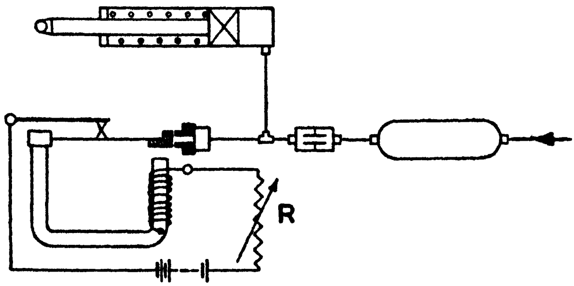


FIG. 156. ELECTRO PNEUMATIC VIBRATOR

method is that shown in Fig. 155. In this the usual slide valve is operated by a solenoid, the pull of which is balanced against a spring. The machine being controlled by the jack is made to vary the current in the solenoid so that the position of the slide due to the pull of the solenoid balances the performance of the machine in the required manner.

Fig. 156 shows another unusual electro-pneumatic system. The position of the jack is controlled by varying the pressure in it which balances a spring. The pressure from the system, passing through a restriction, is allowed to escape through a small cock opened by the movement of a reed, which, at rest, obstructs the escape of air. The flow of air from this unit is, therefore, proportional to the movement of the reed, which is actually made to vibrate by an electrical vibrator. The position of the jack is, therefore, controlled by the frequency and amplitude of the vibration of the reed, in turn controlled by a rheostat. As a development of this system the reed may merely be excited by a simple coil without using it as a vibrator, and in this case appreciable air will escape when the reed is excited with its own natural frequency. A double-acting jack may be controlled by two reeds, each excited by a different frequency from a common current source.

CHAPTER 17

MISCELLANEOUS SYSTEMS

17.1. Pump Control Systems. No pressure system will work satisfactorily unless some means of limiting the pressure when the system is inoperative, is introduced. In the simplest system, such as a vehicle hydraulic jack or an air-pump inflating a tyre, the reaction on the operator's hand or foot enables a safe limit to be put to the pressure reached; except on the simplest system, however, a safety valve is used to limit the pressure automatically.

Where a power-driven pump is used, some means of limiting the pressure is obviously necessary. Since the use of a simple relief valve means that the pump delivers at maximum pressure the whole time the system is inoperative—obviously uneconomical from the point of view of wear and tear on the system and consumption of power—alternative systems for "unloading" the pump have been devised.

The various pump control systems can thus be classified as follows—

(*a*) Relief valve.
(*b*) Open circuit.
(*c*) Variable pump delivery.
(*d*) Automatic cut-out or unloader.
(*e*) Automatic clutch control.
(*f*) Electric cut-out.
(*g*) Electro-hydraulic or pneumatic control.

Relief Valve System. As already mentioned above, this consists simply in allowing the pump output to by-pass through a relief valve set slightly higher than the maximum system operating pressure. The full power output of the pump being passed through the relief valve, heat is generated in it and passed to the tank by the fluid, and the system cannot be expected to be reliable without cooling if the power of the pump is large. Many systems at 500–600 lb/sq in., however, have operated satisfactorily in this manner.

Open-circuit System. This consists in using the open centre jack system, as described on page 145, probably with an additional relief valve as in the previous paragraph. Normally the pump idles back to tank through the open centre; when the selector is moved the pump delivers under pressure to the system. At the end of the travel of a jack, etc., the pressure rises, automatically centering the selector in some cases, or opening a relief valve until the selector is manually centred in others (see Chapter 5 for descriptions of the selectors).

An equivalent system uses a separate by-pass for the pump, which is opened and closed manually by a separate lever instead of being built into the main selector.

VARIABLE PUMP DELIVERY. If an automatically variable delivery pump is used (see Chapter 3), there is no need for any unloading system because the pump will adjust itself to the needs of the system and will cease to deliver when the selectors are closed or the jacks, etc., are at the end of their travels.

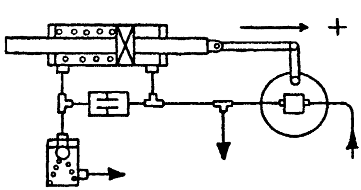


FIG. 157. PUMP DELIVERY CONTROL SYSTEM

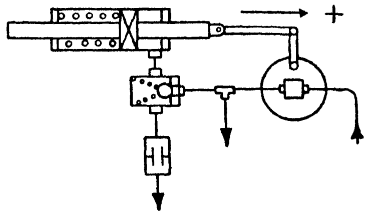


FIG. 158. ANOTHER PUMP DELIVERY CONTROL SYSTEM

In the case of a manually variable delivery pump, the delivery lever, which usually varies the pump stroke, can be coupled to the selector lever. In simple cases the pump can be reversed, enabling a jack to be reciprocated without any selector being used.

Various forms of valve device, as used in automatic cut-outs (see below), can be adapted to vary the stroke of a pump when the pressure reaches a predetermined point to neutralize the delivery. One method is shown in Fig. 157. It will be seen that a spring normally holds the pump lever in the delivery position, the piston on the operating jack being in hydraulic balance. When the relief valve lifts at a predetermined pressure, the pressure in the pump output will rise slightly owing to throttling of the feed to the relief valve, and this unbalances the stroke control piston, moving it to the left and reducing the delivery of the pump. The pump will then deliver only just enough fluid to maintain the necessary unbalance of the piston. Fig. 158 shows another method of achieving the same object. In this the pump output lifts a relief valve to admit pressure to the stroke control jack; fluid can leak away permanently through a restriction causing a back pressure on the relief valve (which raises the pump pressure equivalently), sufficient to maintain the stroke control jack at zero or almost zero.

AUTOMATIC CUT-OUT OR UNLOADER. This method uses an automatic cut-out of the type described in Chapter 6; the valve enables the pump to be by-passed or unloaded either to zero pressure or a low pressure for lubricating the pump, etc. (or even to operate low pressure services), leaving the main system under pressure. To give the system some capacity and prevent the pump continuously cutting in and out, an accumulator or air bottle is generally used in the system or delivery side of the cut-out.

For the best performance of the system the cut-in pressure should be close to the cut-out pressure; to give a degree of unbalance to the cut-out (see Chapter 6 for reasons) an appreciable difference is needed; thus the actual ratio is a compromise and generally about 0·8–0·9 on a good cut-out. If the cut-in pressure is relatively low, the risk is run that if the system is stalled in mid-travel for some reason or other, it may not re-start automatically without moving back and starting all over again.

AUTOMATIC CLUTCH CONTROL. If the pump is fitted with a clutch, an automatic control equivalent to a cut-out is to be preferred to manual control of the clutch. The pump clutch will either require pressure to engage it, or being spring engaged will require pressure to disengage it, or can be spring toggle engaged and disengaged. The various valves suitable for this control are described in Chapter 6.

With this system an accumulator to give the circuit capacity is needed, as before.

ELECTRIC CUT-OUT. If the pump is electrically driven, the motor can be stopped and started by using a pressure switch of the type described in Chapter 6. Any desired ratio of cut-in to cut-out pressure can be obtained by suitable arrangement of the switch. Obviously the switch will operate through an electrical relay.

With this system an accumulator to give the circuit capacity is needed, as before.

ELECTRO-HYDRAULIC OR PNEUMATIC CONTROL. When electric control of the selectors is used, as described in Chapter 16, it is obviously a simple electrical problem to switch on or off the pump, or a solenoid cock acting as a by-pass, or a solenoid selector controlling the clutch, at the same time as the main selector is energized, and equivalent means can be used to de-energize the pump electrical circuit at the same time as that of the selector where desired. The electrical aspects of this problem are not within the scope of this book.

17.2. Emergency Systems. An "emergency" system can be defined as one attached to a main system, and able to operate part or all of the latter in special or emergency circumstances, usually when some part of the main system has failed.

Fig. 159 shows the application of an emergency system to the extension of a jack, using a shuttle valve at the jack to enable the normal or emergency extension lines to be used as desired. Shuttle valves have already been described on page 130 (Chapter 9). It will be seen that normally the emergency line is vented

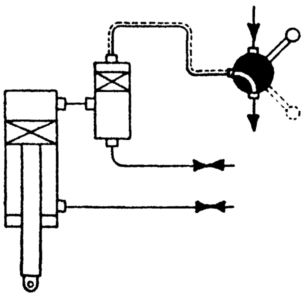


Fig. 159. Emergency System with Simple Shuttle Valve

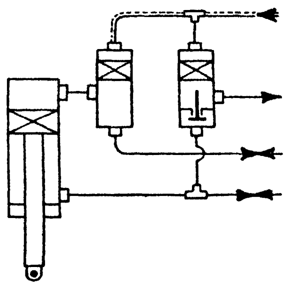


Fig. 160. Emergency System with Simple Shuttle Valve and Release Valve

so that the shuttle piston can seat and isolate the special line; but when the emergency selector is moved, pressure can be applied through the emergency line, the main line being isolated by the shuttle valve. If the main line is full of fluid, or selected as well, the shuttle may not be able to move properly, but this will either not matter or will be observed and one or other line vented.

A cock can be used instead of an emergency selector, if the risk of fluid leakage or expansion moving the shuttle piston over to block the jack connection can be taken.

A difficulty with this system is that unless the jack is selected "down", the fluid in the "up" or annular volume will lock the jack hydraulically, and the fluid pressure here may be increased in the ratio of full to annular areas, and will probably be excessive. One means of guarding against this is to add a hydraulic lock acting as a jettison or release valve, as shown in Fig. 160. It will be seen that emergency pressure opens the valve and releases the "up" fluid in the jack, if not already released.

Another method of preventing wrong selection of the emergency control is to interlock it with the main selector mechanically, or hydraulically as shown in Fig. 161. This uses a small jack which blocks the emergency selector lever when the "up" line to the jack is full of oil or under pressure. This is not, however, so sure or positive as the mechanical method, since the jack is dangerous even if only partly filled with fluid.

The method of applying an emergency control to an accumulator-jack system is shown in Fig. 162. In this case a hydraulic lock is fitted between the jack and accumulator, and acts as such. It also enables emergency pressure from a cock (not

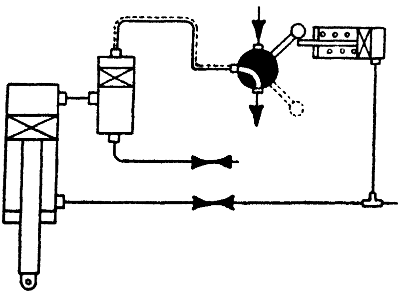


FIG. 161. EMERGENCY SYSTEM WITH SAFETY INTERLOCK

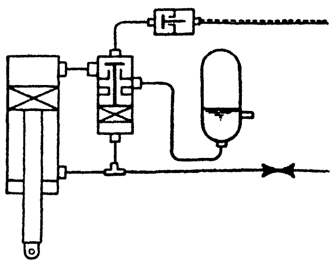


FIG. 162. EMERGENCY SYSTEM FOR ACCUMULATOR SYSTEM

necessarily a selector) to pass through a non-return valve, close the lock valve and isolate the accumulator, and extend the jack. A danger here, too, is that pressure, or even back-pressure in the "up" line, will open the lock and allow emergency fluid to pass into the accumulator, with possibly undesirable results. A jettison valve or interlocking system can be used as before.

There are various methods of deriving the source of fluid for the emergency supply. The main ones are—

1. A second (e.g. hand) pump sucking from the normal reservoir, with or without a "stack pipe" for the main pump, so that there is a special emergency volume of fluid reserved for the emergency pump; or with a separate reservoir.

2. Stored energy in the form of an air bottle or accumulator feeding the emergency selector, and charged automatically, or by special routine.

3. Supplying the emergency system from the main pressure source, this covering failure of the pipes or components, but not the pumps themselves.

These are more or less obvious; one additional possibility, however, is illustrated in Fig. 163, and this guards against the failure of one service putting the emergency out of action for another. In this, the various emergency selectors are connected in series, the actual emergency part of each being shown as a cock, although 3-way units could be used by adding another valve to each. In the position shown, the emergency pump can deliver through all the selectors (as in an open-centre system) to the main system and can thus supplement the main pump. The main pump can also deliver through them, but can have an additional optional direct pipe line as shown. When any emergency system is used the appropriate selector is moved; this closes the through connection (or "open centre") and admits

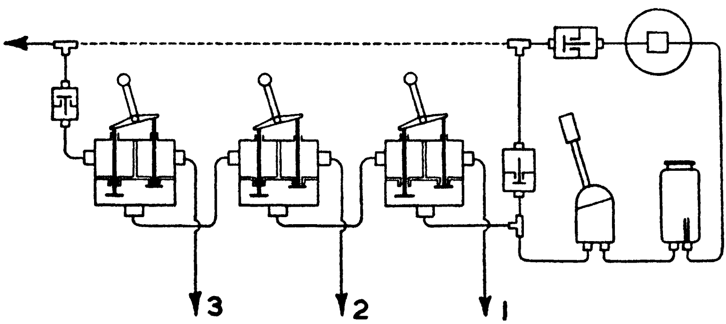


FIG. 163. SERIES EMERGENCY SYSTEM

fluid to the appropriate circuit. Only one circuit can be used at once, the priority depending on the order of the selectors. Either pump can operate the main system and either can operate any emergency system. If the main line or system is defective, the main pump can still operate the emergency units if the by-pass line is not used, and assuming that the main pump fluid has not already been lost.

Another known device is to use a float operated valve in the reservoir to isolate part of the system if the fluid should fall to a dangerous level, leaving the remaining fluid for particular and important duty. Obviously, very many variations of such a scheme are possible for particular duties.

17.3. Flow Accelerator Systems. The expression "flow accelerator" refers to a means for boosting the flow in a pressure system, usually for the purpose of filling the pipe lines or speeding up the exhaust of the jacks, pipe lines, etc. Reference has already been made to "two-stage braking" systems (Chapter 12), which are designed to accelerate the filling of the hydraulic system and increase the overall mechanical efficiency. In pneumatic systems, particularly for brakes where long pipe runs are involved, the time lag involved in compressing the air into the pipes and then exhausting it again may be a serious limitation. In the case of steam or vacuum brakes, the use of relays was introduced many years ago, so that each carriage, wagon, etc., could be fed from its own pressure reservoir.

Quick release valves are commonly used to vent a pneumatic or vacuum system to atmosphere; these are described in Chapter 9.

Two electro-pneumatic relay systems are illustrated in Figs. 164 and 165; these are designed to accelerate the flow in either direction by energizing a second electro-pneumatic selector in

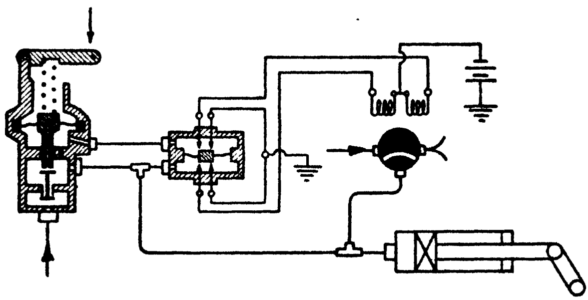


FIG. 164. FLOW ACCELERATOR SYSTEM

sympathy with the main control. In the case of the system shown in Fig. 164, pressure is applied from the bottle through a power brake valve to the brake jack in the normal way. The brake valve is fitted with an additional fixed diaphragm in the body restricting the passage of air pressure, through a small bleed hole, to the normal moving diaphragm chamber in the valve; thus a pressure difference exists across the fixed diaphragm while the balance of the moving diaphragm is taking place. This pressure difference reacts on the diaphragm of a switch box so that a pair of contacts are closed by movement of this diaphragm. These contacts energize a 3-way pneumatic selector close to the brake jack and admit additional air directly. When pressure is released by the power brake valve, pressure in the balance chamber can only leak away slowly through the bleed hole, and the diaphragm in the switch box is moved to close the other pair of contacts, opening the electro-pneumatic selector to atmosphere and venting the brakes locally.

Fig. 165 shows a somewhat similar system, except that the power brake valve is of normal type. Flow of air from this valve passes through another type of switch box, and when the flow is great as when the brakes are first applied, a hinged valve moves over to close a pair of contacts to energize the boosting selector as before. In the reverse direction the vane moves the other way when the escaping flow is great, and again opens the electro-pneumatic selector.

17.4. Reservoir or Pump Suction Boost Systems. On many installations, particularly on aircraft operating at high altitude, it is often necessary to boost the suction of the pump, even if the reservoir, by its location above the pump, provides some static head. When the operating fluid is cold or viscous the

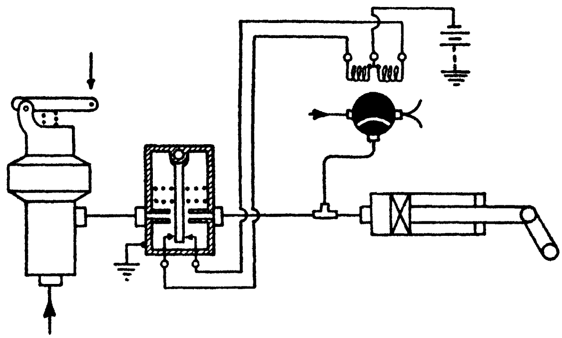


FIG. 165. FLOW ACCELERATOR SYSTEM

friction losses in the suction lines may be excessive and a degree of boost desirable.

Known means of providing some degree of positive pressure in the suction line are—

(*a*) Mounting the reservoir above the pump.

(*b*) Use of a first-stage low pressure pump, possibly in the reservoir, to supply the main pump.

(*c*) Use of a first-stage pump as above, driven by a hydraulic motor in the pump delivery line.

(*d*) Supercharging the reservoir by low pressure compressed air from an external supply.

(*e*) Fitting the tank with valves so that it supercharges itself every time there is a return of fluid into it.

(*f*) Use of a jet-pump operated by the pump delivery.

(*g*) Use of a jet-pump operated by the return fluid from the system.

Means (*a*) is obvious; means (*b*) is excellent if complicated; 2-stage pumps are known and since the first stage can be very simple, this is a good solution; the suction ability of the first stage, which must obviously have a large excess swept volume over the main pump, is however limited.

Means (*c*) is ingenious, but expensive, although absorbing relatively little power from the main pump; unless the pump lines are full, it may not be self-starting without priming, and this is a dangerous feature.

Means (*d*) is a good solution, being cheap and reliable, provided compressed air is available. From the aircraft point of view it is relatively heavy, since the additional weight of the reservoir, even for a boost of 5–10 lb/sq in. is appreciable.

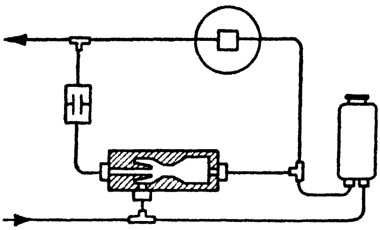


FIG. 166. PUMP SUCTION BOOSTER USING JET PUMP PRINCIPLE

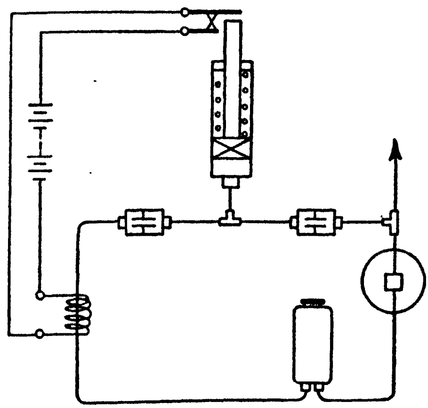
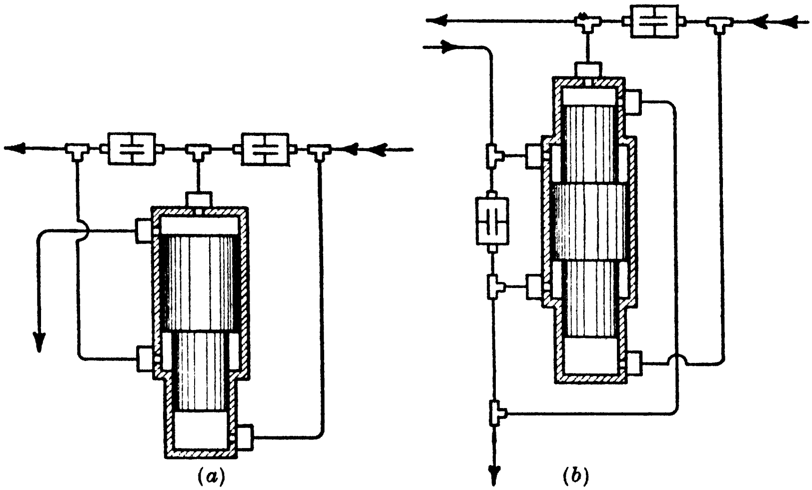


FIG. 167. VISCOSITY CONTROL SYSTEM —TEMPERATURE CONTROLLING

Means (*f*) and (*g*) both use the well-known jet pump which, although of low efficiency, can create enough pressure for the boosting of the suction of the pump. The principle of the two systems is shown in Fig. 166. In the first case, as illustrated, part of the pump delivery is bled to the jet pump; in the second case it is the return fluid from the tank which passes through the jet pump. The fluid ejects out of the nozzle of the pump with high velocity, and in passing through a venturi-shaped restriction creates a suction, lifts up additional fluid and passes it out under low pressure. The efficiency is not high, but sufficient to boost the suction without excessive loss of pump delivery. In the case of the use of return fluid, however, while the pump is not bled, the volume of fluid being returned, although greater than that bled from the pump in the other case, depends on whether the annular or full volume of the jacks, for example, is being returned. Which of the two systems is the better depends on the particular circumstances of each. For further information on jet pumps the reader should turn to textbooks on hydraulics or the steam engine.

17.5. Viscosity Control Systems. A system for controlling fluid viscosity is shown in Fig. 167. A bleed from the pump delivery passes through two restrictors and a heating coil to tank. The pressure existing in the volume between the two restrictors is proportional to the viscosity of the fluid, and when the fluid is thick the pressure is great enough to lift the pressure switch and switch on the heating coil. This warms the bleed fluid,



(a) (b)

FIG. 168 (a), (b). VISCOSITY CONTROL SYSTEM—FLOW DIVIDING

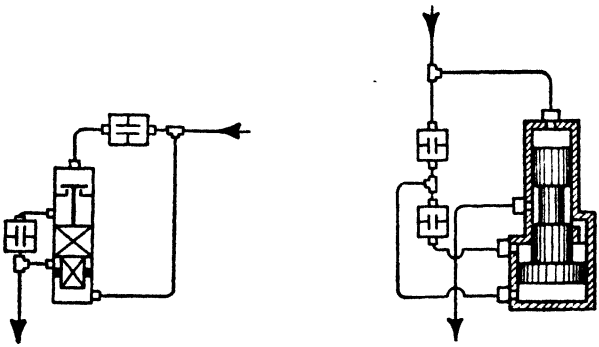
ultimately raising the temperature of the reservoir. Probably a thermostatic control on the latter would be more practicable.

Ingenious means of viscosity compensation is shown in the diagrams of Fig. 168 and 169. The principle of the first system as shown in Fig. 168 (a) will be seen to be the use of a by-pass type of flow divider which meters part of the flow to the jack, but in place of a spring acting on the piston of the divider, use is made of the pressure difference across a second restrictor orifice. Between the pump and jack are thus two orifices, the first reacting on the flow divider and the second being the normal feed rate restrictor. The first restrictor acts as the viscosity regulator if the pressure drop across it is more or less the same as across the feed restrictor; this is because any change in pressure drop across one restrictor would be balanced by a corresponding change across the other, the effect of viscosity thus cancelling out. For correct operation the areas of the piston acted upon by the various pressures should obviously be in balance (i.e. the large piston area should be double the smaller plunger area).

The construction of Fig. 168 (*a*) is for use in the main feed line to a jack; it will be noted that both the restriction orifices could in fact be contained in the piston.

The arrangement of Fig. 168 (*b*) has the viscosity restrictor in the main feed line and the feed rate orifice in the return line. This has advantages already referred to in Chapter 13.3.

The systems shown in Fig. 169 are similar to those just described except that a flow control valve of the variable throttle type is



(*a*) (*b*)

FIG. 169 (*a*), (*b*). VISCOSITY CONTROL SYSTEM—FLOW CONTROLLING OR THROTTLING

used, again arranged with a reaction due to the pressure drop across an orifice to take the place of the usual spring in the reducing valve element. Again the viscosity control arises owing to this pressure drop compensating for changes in drop across the fixed feed regulating orifice. The system shown in Fig. 169 (*a*) is installed in the feed line to the jack; the fluid passes first through an orifice, then through the usual differential reducing valve to the second feed restrictor. This valve is identical with the flow controller of Fig. 83 (*b*) with the addition of another reaction piston connected directly to the pump before the viscosity restrictor. Here again it will be realized that any change of viscosity does not affect the feed rate because the effect cancels out across the two restrictors.

Fig. 169 (*b*) is a similar system where the flow restrictor is ahead of the reducing valve, as in the case of Fig. 83 (*d*), again arranged with an additional reaction diameter and influenced by the pressure drop across the viscosity restrictor.

APPENDIX

A Note on Some Early Hydraulic Mechanisms

There are one or two hydraulic devices of antiquity which are worth describing, if only to put modern developments in a better perspective. A very considerable amount of development of valve gear took place following the invention of the steam engine, and early books on such machinery make interesting reading.

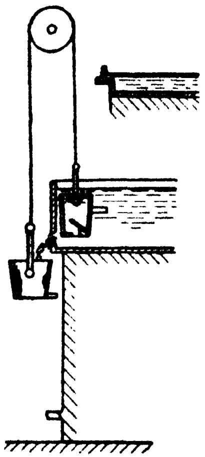


Fig. 170. Gaining and Losing Bucket System—circa A.D. 1600

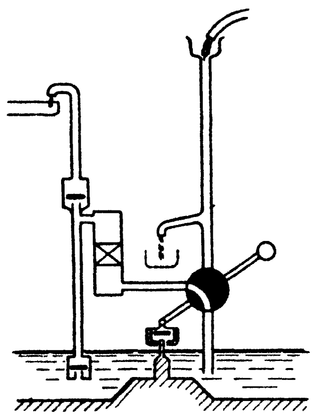


Fig. 171. Automatic Reversing Pumping System—A.D. 1618

A very fine account of pump and similar mechanisms is given by Reuleaux in his classic book.*

The following illustrations have been derived from Ewbank's early work on hydraulics,† and the even earlier study by Belidor.‡

Fig. 170 shows the method of raising water by the so-called "gaining and losing bucket" system, and dates from about 1600. Water flowing into one bucket eventually raises the second bucket, itself being filled through a non-return flap valve in its bottom. When both buckets reach the ends of their travel, they are tipped up automatically (the rising bucket first), the buckets

* "Kinematics of Machinery," by F. Reuleaux, London, 1876.

† "Hydraulic and other Machines," by Thomas Ewbank, London, 1842.

‡ "Architecture Hydraulique," by Belidor, Paris, 1739.

then being so weighted as to return to the filling position. Thus the energy of one head of water is used to raise a second head. This mechanism shows us an automatic reversing system, and an early use of a non-return valve.

Fig. 171 shows a more involved, but more modern-looking, system with the same object; this is ascribed by Ewbank to one Fludd and dates from 1618. Water pressure from one source flows through a three-way selector to a floating piston pump, raising the piston and discharging water from the other side to the output system, with, obviously, a reasonably small loss of head. At the same time a small bucket on an arm attached to the selector rotor is filled from a bleed jet off the main pipe, and

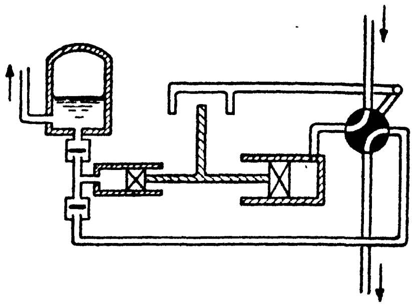


FIG. 172. EARLY PUMPING SYSTEM

eventually drops, rotating the selector to the position shown, allowing the piston in the pump cylinder to drop under its own weight (or a spring) and suck in a further volume of water. Meanwhile the small bucket is emptying because a small spindle lifts the non-return valve on its bottom, and when the water has drained away, the counterbalance on the selector lever rotates the rotor again, to restart the cycle of events. This ingenious system, which ought to command the respect of all modern hydraulic engineers, shows us an automatic reversing system, a sequence valve, a 3-way rotary selector, a floating separator piston, and a true relay action, since the original machine used waste water to pump clean water.

Fig. 172 shows a later pumping machine illustrated by Ewbank from Belidor. In this, descending water moves a large piston connected to a second smaller one, discharging fluid from the smaller cylinder to the output system. As the pistons reach their extreme position, the 4-way selector is moved over, connecting the large piston to exhaust, and the smaller one to the pressure inlet, thus causing return action and the filling of the small cylinder. Since the pressure in the outlet pipe is greater than the inlet, the output valve of the pump remains closed. Thus, the pump continues to reciprocate, using water equivalent to the volume of both cylinders to discharge the small volume at

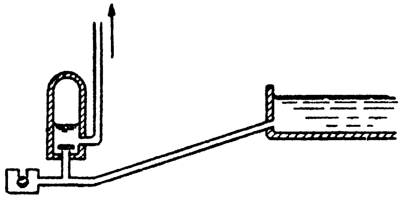


FIG. 173. THE MONTGOLFIER PUMP OF 1796

each cycle. An accumulator is used in the pump outlet to smooth out the delivery, or, by this means, as Ewbank puts it, "the discharge is rendered continuous". This system shows an automatic reversing system, a 4-way selector, a hydraulic intensifier, automatic means for filling the high pressure chamber of the latter, and an output accumulator.

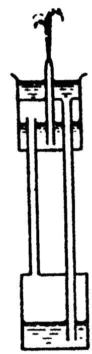


FIG. 174. HERON'S MAGIC FOUNTAIN (SIXTEENTH CENTURY)

Fig. 173 illustrates one of several early pumps which used water momentum or "hammer" to pump the liquid. The one illustrated is the Montgolfier Pump of 1796, although the invention is ascribed by Ewbank to Whitehurst in 1772. Water flowing down a pipe acquires momentum in escaping past a ball (or other) valve at the end of the pipe. The valve is of such a weight that it acts as a hydraulic fuse, closing the outlet when the flow reaches a certain velocity past it. The suddenly arrested water column then dissipates its energy in passing a volume of water into the accumulator chamber, and thus to the outlet pipe. Then with the flow stopped, the ball valve opens again and the cycle recommences. Actual systems require careful balancing to obtain the correct continuity. This invention shows the use of pressure waves or impulses in pipes, an accumulator and a hydraulic fuse.

Fig. 174 is an amusing device attributed to Heron in the sixteenth century.* Although as described it serves as a toy fountain, the principle has been used for pumping. Water in an

* "Spiritalia," by Heron, 16th Century. This book describes what must be the earliest slot machine, used in churches, and which, when a coin of at least a given weight was dropped in, delivered a small quantity of spiritual water, probably to the astonishment of the worshipper.

upper reservoir flows down into a lower chamber, raises the fluid level there, compresses the air in it and thus in the upper closed chamber; this in turn blows water out of the fountain nozzle. The height of the fountain jet will approach the height dropped of the "pumping" fluid, once the air is initially compressed.

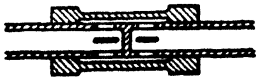


FIG. 175. EARLY TYPE OF SLIDE-COCK

Fig. 175, from Ewbank, illustrates a type of cock not already mentioned in this book. It is really an inversion of the ordinary slide valve, since the moving sleeve passes over ports in the central pipes, i.e. is on the outside instead of the inside of the body which carries the fluid connections. This principle can exist in various forms other than that illustrated, e.g. 3- and 4-way selectors, etc.

INDEX