(19) Europäisches Patentamt
European Patent Office
Office européen des brevets

(11) Publication number: **0 092 972**
B1

(12) **EUROPEAN PATENT SPECIFICATION**

(45) Date of publication of patent specification: **30.07.86**

(51) Int. Cl.⁴: **F 15 B 18/00**

(21) Application number: **83302247.8**

(22) Date of filing: **20.04.83**

(54) **Hydraulic control system.**

(30) Priority: **22.04.82 GB 8211628**

(43) Date of publication of application:
02.11.83 Bulletin 83/44

(45) Publication of the grant of the patent:
30.07.86 Bulletin 86/31

(84) Designated Contracting States:
DE FR GB IT

(56) References cited:
GB-A- 940 942
GB-A-1 029 527
GB-A-2 082 799
US-A-3 915 427

(73) Proprietor: **FAIREY HYDRAULICS LIMITED**
Cranford Lane Heston Hounslow
Middlesex, TW5 9NQ (GB)

(72) Inventor: **Fuell, Frederick James**
67 Minterne Avenue
Norwood Green Southall Middlesex (GB)

(74) Representative: **Ashmead, Richard John et al**
KILBURN & STRODE 30 John Street
London, WC1N 2DD (GB)

Courier Press, Leamington Spa, England.

EP 0 092 972 B1

Description

This invention relates to hydraulic control systems, using valves to control the supply of hydraulic fluid to an actuator. The invention is concerned with the control of such actuators which are to be used in the positioning of movable loads of substantial size with a high degree of accuracy and reliability. A particular but not exclusive application of such hydraulic control systems is in the control of actuators for use in the power operation of control surfaces of an aircraft.

In many applications of hydraulic actuators, it is desirable to position a movable load of some several tonnes with a high degree of accuracy while maintaining a high degree of protection against failure. Many such actuators are required to be controlled remotely by way of electrical signals from a remote control point and it is necessary to provide redundancy to accommodate the failure of various components in the hydraulic system itself or the electrical control system and hydraulic valves associated with it, so that control of the actuator may be maintained in the event of such a failure. One particular, but not exclusive, example of such an actuator is a hydraulic actuator used to effect the movement of an aircraft control surface, particularly a high speed aircraft. It has been proposed to use hydraulic piston and cylinder actuators to move the various control surfaces of an aircraft, the actuators being arranged to operate under servo control in response to movement of the control column or pedals of the aircraft by the pilot. Electrical transducers associated with the pilot control input elements have been arranged to provide electrical signals which in turn are fed to the actuator system as servo control signals to control the output position of the actuator itself. At the same time the pilot control input elements have been mechanically coupled to the control system and the actuator to provide a direct mechanical coupling by which control may be maintained in the event of failure in the electrical signalling system, allowing the pilot to maintain control of the aircraft. This has necessitated the provision of a mechanical linkage between the pilot control elements in the cockpit and the hydraulic actuator sited adjacent the control surface in question which while not requiring to transmit the full control forces have nevertheless involved precision of operation to provide accurate mechanical control.

With the need to develop aircraft having ever higher performance, and the development of electronics for highly sophisticated on-board computer systems, it has become desirable to take advantage of the greater aerodynamic efficiency which can be achieved with an aircraft which is inherently unstable. With such an aircraft the aerodynamic penalties associated with achieving inherent stability can be eliminated or minimised, but such an aircraft has to be "flown" continuously and it would be beyond the capability of a pilot to fly such an aircraft under mechanical control. With the development of computer systems, a computer may be used continuously to "fly" the aircraft and thus provide inherent stability.

In order to utilise such a computer system it is necessary to provide hydraulic control surface actuators which are electrically controllable under the influence of the computer and which provide sufficient reliability and redundancy in the control system to eliminate the need for stand-by mechanical connections with the pilot control elements in the aircraft, since they would be of no substantial use to the pilot as a fall-back system.

It is an object of the present invention to provide a hydraulic control system using electro-hydraulic valve control of an actuator which is suitable for use in an arrangement as described above, particularly in providing a required degree of redundancy against hydraulic and/or electrical failure and compensation for electrical hydraulic and/or mechanical tolerances or errors. Further and other objects and improvements achieved by the present invention will be apparent from the following.

According to the present invention a hydraulic control system comprises two actuators operating in parallel or tandem each having a separate supply of hydraulic fluid and a primary and a secondary electrohydraulic servo control valve interposed between each actuator and its respective hydraulic supply, each control valve having two operating coils for the electrical operation thereof, the said coils being connected to four electrical control lanes in such manner that each lane is connected to one coil of a primary control valve associated with one actuator and with one coil of a secondary control valve associated with the other actuator, the primary control valves having a linear input current output flow characteristic passing through the zero current flow point and the secondary control valves have an overlap at the zero flow point such that they provide no flow output for a part of the input current range either side of the zero position. The said overlap at zero flow point is preferably such as to provide no flow output for up to 15% on either side of zero of the input current range. The normal flow gain of each of the secondary control valves having overlap may differ from the normal flow gain of each of the primary valves without overlap such that the full control signal in either sense or direction applied to both types of valve achieves substantially the same maximum flow rate to the respective actuators. The system is preferably provided with means to feed an electrical feedback signal to the said control valves in response to a pressure differential across one and/or the other actuator, and in this arrangement the system is preferably such that a first pressure difference feedback signal generator is provided to give a feedback signal in response to a pressure differential across one actuator and a second pressure generator across the other actuator, each generator giving its feedback signal to the two electrical lanes connected to the primary control valve of its associated actuator. The pressure feedback signal magnitude is preferably arranged to be less than 10% of the valve control current.

The actuators in each case may control the supply of hydraulic fluid to a main power actuator.

The hydraulic control system in the present invention therefore comprises four electrical control lanes each serving in series one of the two coils of one of the valves of each actuator. One such valve which can be regarded as the primary valve in each case, has a nominally zero overlap and high pressure gain and the other, the secondary valve, has a significant overlap and a lower pressure gain. The object of the overlap is to ensure that within the null changes which are to be expected in service, the secondary overlapped low gain valve will remain in the overlap region thus eliminating the parasitic flow which would otherwise arise if the primary valve were standing open in one direction and the secondary valve in the other.

Each of the four control lanes is wired in series to a high gain primary valve without overlap of one actuator and the low gain secondary valve of the other. In normal operation of the system the performance of each actuator is based primarily on the performance of its primary valve, its secondary valve being provided for redundancy and failure survival purposes. When for example one valve associated with an actuator fails mechanically, the presence of another valve associated with that actuator enables the effects of the failed valve to be passivated by opening the second valve in the opposite sense to that in which the first valve is jammed therefore setting up parasitic flow. Control of the system output is then maintained by the other sound actuator.

When pressure feedback sensing means, for example a transducer, are provided for each actuator to overcome "fighting" and dead zones arising from mechanical/hydraulic errors by equalizing the pressures on either side of the actuator concerned, the pressure feedback means may also be used to identify failures for example mechanical failures in the control valves prior to use of the overall system, for example aircraft, relying on use of the hydraulic control system.

The control system of the invention thus utilises a dual valve arrangement associated with each of two actuators to enable the seizure or other defect of any valve to be overcome. By the use of two valves electrically in series electronic lane tolerances are distributed equally to the valves in both actuators. By provision of pressure feedback sensor to feed a pressure feedback signal into the control lanes and thence to the valves associated with the arrangement of primary and secondary valves described above, residual and electronic and/or mechanical tolerances can be overcome, the overlap/no overlap relationship of the secondary and primary valves respectively enabling antiphase fighting arising from tolerances to be washed out by the action of one valve per actuator despite the equal feeding of the feedback signal into the valves of both actuators.

Other objects, benefits and advantages of the invention will be apparent to those skilled in the art.

In order to promote a full understanding of the above and other aspects of the present invention an embodiment will now be described by way of example only with reference to the accompanying drawings, in which:—

FIGURE 1 is a schematic representation of a hydraulic control actuator with a hydraulic valve system for electrical control inputs;

FIGURE 2a is a more detailed schematic diagram of the arrangement of Figure 1 in a first control state;

FIGURE 2b is a similar diagram to that of Figure 2a with the system in a second control state; and

FIGURE 3 shows the arrangement of Figure 1 in more detail still.

With the arrangement shown in Figure 1, which might be for the actuation of an aircraft control surface or other item, there is provided a hydraulic piston and cylinder actuator 10 having a body 11 arranged to be mounted on a structure and an output piston rod 12 arranged to be coupled to a movable element carried on that structure. The hydraulic actuator 10 has in fact two actuators 10A and B arranged back to back, coupled through their piston rods and bodies, but controlled independently of one another. A dual hydraulic supply system (not shown per se) is provided, one system supplying the actuator 10A through a valve system indicated 14A and the other supplying the actuator 10B through a valve system indicated at 14B. Electrical control input signals may be provided to the valve systems 14A and 14B through four electrical control lanes indicated at 15, 16, 17 and 18 which are each independent of each other. The four lanes 15 to 18 couple the actuator 10 to a control input element (not shown) such as the control column in the pilot's cockpit. It will be noted that each of the electrical supply lanes 15 to 18 is connected to both of the control valve systems 14A and 14B. In the event of hydraulic failure in one of actuators 10A or B or their respective hydraulic supply systems, the system may continue to operate under the other actuator and supply system. In the event of failure of two or even perhaps three of the electrical signal lanes, control may still be maintained albeit degraded.

As shown in Figures 2A and 2B each of the valve systems 14A and 14B has a primary hydraulic valve 20 and 25 and a secondary hydraulic valve 21 and 24 so that there are two valve elements to control the flow of hydraulic fluid from each fluid source to the respective actuator 10A or 10B. The two electro hydraulic servo control valves 20 and 21 are arranged to supply the actuator 10A with hydraulic fluid from supply lines 22 connected to a first hydraulic supply system (not shown) having an associated return line 23. The valves 20 and 21 are electrically controlled in a servo system having feedback derived from a position sensor (not shown) responsive to the output position of the hydraulic actuator 10A. Two further electro hydraulic control valves 24 and 25 are similarly arranged to supply fluid from supply lines 26 to the actuator 10B with an associated return line 27.

Figure 2A illustrates a steady state situation where no actuator displacement is being demanded e.g. by the pilot. In the situation shown in Figure 2A the control valve 20 is assumed to be jammed in a fully open condition allowing full hydraulic pressure P to be applied to one side 40 of the actuator 10A. It will be seen in this situation that the valves 21, 24 and 25 only need to take a very small offset to produce a situation with the full pressure P on the opposite side 43 of the actuator 10B in the second hydraulic system thus preventing displacement of the rod 12. When movement is required of the actuator rod 12 with the valve 20 still jammed in the same fully open condition the system compensates by movement of valve 21 as shown to equalise the pressure (at approximately $\frac{1}{2}$P) in the two sides 40 and 41 of actuator 10A leaving actuator 10B to operate normally under control of valves 24 and 25. In the arrangement shown in Figure 2B the valves 24 and 25 will under no load or low load conditions settle so as to provide pressure at approximately $\frac{1}{2}P$ in both sides 42 and 43 of actuator 10B, movement at maximum rate or close to it of the piston 12 being effected by sufficient relative variation from $\frac{1}{2}P$ in sides 42 or 43 to overcome friction etc. At higher load requirements e.g. of a wing surface at high speed will require a greater pressure variation delivered again by appropriate movement of valves 24 and/or 25.

In aircraft control systems pressure P may be in the region of 20 to 30 MN/m^2, but systems utilizing the present invention may operate at whatever pressure required. Thus, as shown in Figure 2B when a maximum movement rate is demanded of the actuator 10, near maximum rate will be achieved since the valve 21 will be displaced fully in the sense required to neutralise the effect of the jammed valve 20, whilst the full control condition can be achieved by the valves 24 and 25 operating on the actuator portion 10B.

This arrangement provides a system with a very good failure survival capability. The system has a further advantage in that any electronic lane tolerances are fed identically into control valves associated with both actuators 10A and 10B supplied by separate hydraulic systems and thus do not create the anti-phase fighting of pressure illustrated in Figure 2A.

However further tolerances may exist e.g. in the mechanical elements and electrical actuating coils of the valves themselves and such tolerances may of sufficient magnitude themselves to cause antiphase fighting as illustrated in Figure 2A. The antiphase fighting condition can severely affect the threshold performance of such systems.

In order to overcome this, means are provided to measure the differential pressure across each actuator and to feed back to the electro hydraulic control system an electrical signal which is proportional to that pressure difference, substantially to equalise the pressures delivered to each side 40 and 41 of actuator 10A and to each side 42 and 43 of actuator 10B. Thus, for example, if there is pressure differential between sides 40 and 41 of actuator 10A in an otherwise no load condition, a feedback signal proportion to that pressure differential, will be required to be fed to valves 20 or 21 to restore substantially equality of pressure between sides 40 and 41. Since however the required feedback signal e.g. to valves 20 and/or 21 will be fed equally and in the same sense to valves 24 and/or 25 the objective i.e. equalisation to compensate pressure imbalance cannot be achieved without further provision.

In order to overcome this the system may be provided with the above described two valve per actuator arrangement using primary and secondary electrohydraulic valves of different overlap characteristics within each of the valve systems 14A and 14B associated with the respective actuators 10A and 10B. In particular, the primary valves 20 and 25 have subtantially linear electric current/hydraulic flow characteristics across their centre position, while the secondary valves 21 and 24, have a significant overlap at their centre position so that they do not respond to give hydraulic flow, to signals of say less than 15% of full control signal.

Such an arrangement is illustrated schematically in Figure 3. It should be noted that in this arrangement the control valves 20, 21, 24 and 25 are associated with the two portions 10A and 10B of an actuator 10 which is arranged in turn to drive a main power control valve (not shown) which similarly controls the supply from the two hydraulic systems to two separate portions of a main power actuator (not shown). Apart from this the arrangement is similar to that of the previous Figures and its operation is of course the same in principle.

In the arrangement of Figure 3 a pressure difference ~~feedback~~ signal generator 30 is provided to give a feedback to electrical lanes 15 and 16 in response to the pressure differential between the two sides 40 and 41 of the actuator 10A. A second pressure difference feedback signal generator 31 is arranged to give feedback signals to the lanes 17 and 18 in response to the pressure difference on the two sides of the actuator 10B.

If now the pressure feedback signal magnitude is arranged to be less than 10% of the valve control current then one valve (the primary valve) associated with actuator 10A or 10B will respond to the pressure feedback signal while the other (the secondary valve) will not. Thus the interphase hydraulic control conditions can be offset through the action of primary valves having no overlap, 20 and 25, while the other or secondary valves having overlap, 21 and 24 may be utilised to offset a hard-over condition as previously described. To achieve this the normal flow gain of the primary valves (20 and 25) and secondary valves (21 and 24) are chosen to be different so that the same full signal level of either sense or direction applied to both types of valves will adhieve essentially the same maximum flow rate out of both types of valve into their respective actuators 10A and 10B. Thus a hard-over condition in one direction in say the valve 20 can still be neutralised by displacement of the valve 21 in the opposite direction when movement is demanded.

Thus it can be seen that the invention provides a system which retains full electrical lane redundancy even after one of the hydraulic systems has experienced a failure. The system is also able to neutralise the effect of electric lane tolerances by effecting the command identically to valves associated with the two actuators 10A and 10B by wiring the control coils of the valves in one hydraulic system in series with the coils of the valves of the other hydraulic system. The use of more than one valve per hydraulic system enables a hard-over or other fault condition in one valve to be neutralised by the action of the second valve in the same system. A combination of the above features with the provision of equalising pressure difference feedback signals enables the prevention of hydraulic "fighting" between the two portions of the actuator. The use of valves of different characteristics in each of the two hydraulic systems permits all the features mentioned above to be embodied a at one and the same time.

The primary valves 20 and 25 will be axis cut i.e. will be linear across centre with no significant overlap. For aircraft use, particularly high speed aircraft, such a valve will have a high pressure gain not less than 28 MN/m^2 per milli amp per coil. It would also have a high band width performance unless special action is taken to reduce the flow gain of the first stage to reduce the overall standing leakage within the system. The secondary valves 21 and 24 would have a normal null bias tolerance of plus or minus 2%. This may deteriorate in service without having a significant effect on the overall behaviour of the system. Deviations in use up to perhaps 7% are believed to be likely, the figure of plus or minus 15% overlap proposed herein being to make allowance for a null bias deterioration of up to plus or minus 10%. For the pressure feedback control it is believed that a 10% signal for a 28 MN/m^2 differential is appropriate. With this the effect of the pressure feedback will not be significant when applied to the secondary valves and only the primary valves will normally contribute to the equalisation process.

Shown next to the coil arrangements of each of the valves in Figure 3 is a small graph showing the overlap characteristics of the valves concerned.

Claims

1. A hydraulic control system comprising two actuators (10A; 10B) operating in parallel or tandem each having a separate supply (22; 26) of hydraulic fluid and a primary (20; 25) and a secondary (21; 24) electrohydraulic servo control valve interposed between each actuator (10A; 10B) and its respective hydraulic supply (22; 26), each control valve having two operating coils for the electrical operation thereof, the said coils being connected to four electrical control lanes (15—18) in such manner that each lane is connected to one coil of a primary control valve associated with one actuator and with one coil of a secondary control valve associated with the other actuator, the primary control valves (20; 25) having a linear input current output flow characteristic passing through the zero current flow point and the secondary control valves (22; 24) have an overlap at the zero flow point such that they provide no flow output for a part of the input current range either side of the zero position.

2. A system as claimed in Claim 1 in which the secondary valve overlap at zero flow point is such as to provide no flow output for up to 15% either side of zero of the input current range.

3. A system as claimed in Claim 1 or Claim 2 wherein the normal flow gain of each of the secondary control valves (21; 24) differs from the normal flow gain of each of the primary control valves (20; 25) such that the full control signal in either sense or direction applied to both types of valve achieves substantially the same maximum flow rate to the respective actuators.

4. A system as claimed in any of Claims 1 to 3 in which means (30; 31) are provided to feed an electrical feedback signal to the said control valves (20, 21; 24, 25) in response to a pressure differential across one and/or the other actuator (10A; 10B).

5. A system as claimed in Claim 4 in which a first pressure difference feedback signal generator (30) is provided to give a feedback signal in response to a pressure differential across one actuator (10A) and a second pressure generator (31) across the other actuator (10B), each generator giving its feedback signal to the two electrical lanes (15—18) connected to the primary control valve (20; 25) of its associated actuator.

6. A system as claimed in Claim 4 or Claim 5 wherein the pressure feedback signal magnitude is arranged to be less than 10% of the valve control current.

Revendications

1. Un système de commande hydraulique comportant deux organes d'actionnement (10A, 10B) fonctionnant en parallèle ou en tandem et comportant chacun une alimentation séparée (22, 26) en fluide hydraulique et une valve primaire (20, 25) et secondaire (21, 24) d'asservissement électrohydraulique interposée entre chaque organe d'actionnement (10A, 10B) et son alimentation hydraulique correspondante (22, 26), chaque valve de commande comportant deux bobinages de commande pour son actionnement électrique, lesdits bobinages étant reliés à quatre voies ou circuits électriques de commande (15 à 18) de

telle façon que chaque voie soit reliée à un bobinage d'une valve de commande primaire associée à un organe d'actionnement et avec un bobinage d'une valve de commande secondaire associée à l'autre organe d'actionnement, les valves de commande primaires (20, 25) présentant une caractéristique linéaire de débit de sortie rapporté au courant d'entrée, passant par le point de débit nul pour un courant nul et que les valves de commande secondaires (22, 24) présentent un recouvrement au point de débit nul de telle façon qu'elles ne fournissent aucun débit de sortie pour une partie de la gamme des courants d'entrée de chaque côté de la position nulle.

2. Un système tel que revendiqué à la revendication 1, dans lequel le recouvrement de la valve secondaire au point de débit nul est tel qu'il ne fournisse aucun débit de sortie de chaque côté du zéro jusqu'à 15% de l'intervalle maximal du courant d'entrée.

3. Un système tel que revendiqué à la revendication 1 ou 2, dans lequel le gain normal de débit de chacune des valves de commande secondaires (21, 24) diffère du gain de débit normal de chacune des valves de commande primaires (20, 25) de telle façon que le plein signal de commande, appliqué aux deux types de valve dans chaque sens ou direction, provoque sensiblement le même débit maximum sur les organes d'actionnement correspondants.

4. Un système tel que revendiqué dans l'une quelconque des revendications 1 à 3, dans lequel des moyens (30, 31) sont prévus pour fournir un signal électrique de contre-réaction auxdites valves de commande (20, 25; 24, 25) en réponse à un différentiel de pression à travers l'un et/ou l'autre des organes d'actionnement (10A, 10B).

5. Un système tel que revendiqué à la revendication 4, dans lequel un premier générateur de signal de contre-réaction à la différence de pression (30) est prévu pour fournir un signal de contre-réaction en réponse à un différentiel de pression à travers l'un des organes d'actionnement (10A) et un second générateur de pression (31) à travers l'autre organe d'actionnement (10B), chaque générateur fournissant son signal de contre-réaction aux deux voies électriques (15 à 18) reliées à la valve de commande premaire (20, 25) de son organe d'actionnement associé.

6. Un système tel que revendiqué dans la revendication 4 ou la revendication 5, dans lequel l'amplitude du signal de contre-réaction de pression est choisi pour être inférieur à une valeur représentant 10% du courant de commande de la valve.

Patentansprüche

1. Hydraulisches Steuersystem mit zwei Betätigungseinrichtungen (10A; 10B), die parallel oder auf Tandemweise jeweils eine getrennte Versorgungseinrichtung (22; 26) für hydraulisches Strömungsmittel betätigen und mit einem primären (20; 25) und einem sekundären (21; 24) elektrohydraulischen Servosteuerventil, welches zwischen jeder Betätigungseinrichtung (10A; 10B) und der zugehörigen hydraulischen Versorgungseinrichtung (22; 26) angeordnet ist, wobei jedes Steuerventil zwei Betriebsspulen für deren elektrische Betätigung aufweist, welche Spulen an vier elektrische Steuerstreifen (15—18) auf solche Weise angeschlossen sind, daß jeder Streifen mit einer Spule eines mit einer Betätigungseinrichtung zusammenwirkenden primären Steuerventils und mit einer Spule eines mit der anderen Betätigungseinrichtung zusammenwirkenden sekundären Steuerventils verbunden ist, wobei die primären Steuerventile (20; 25) eine lineare Eingangsstrom-Ausgangsfluß-Charakteristik haben, die durch den Nullstrom-Flußpunkt verläuft und wobei die sekundären Steuerventile (22; 24) am Nullflußpunkt eine Überlappung haben, so daß sie keinen Flußausgang für einen Teil des Eingangsstrombereiches an beiden Seiten der Nullstelle vorsehen.

2. System nach Anspruch 1, dadurch gekennzeichnet, daß die Überlappung des sekundären Ventils am Nullflußpunkt so ist, daß kein Flußausgang für bis zu 15% an beiden Nullseiten des Eingangsstrombereiches vorgesehen wird.

3. System nach Anspruch 1 oder 2, dadurch gekennzeichnet, daß der Normalflußgewinn jedes sekundären Steuerventils (21; 24) sich vom normalen Flußgewinn jedes der primären Steuerventile (20; 25) unterscheidet, so daß das vollständige Steuersignal in beidem Sinn oder Richtung, aufgebracht auf beide Ventiltypen, im wesentlichen dieselbe maximale Flußrate zu den jeweiligen Betätigungseinrichtungen erzielt.

4. System nach einem der Ansprüche 1 bis 3, dadurch gekennzeichnet, daß Mittel (30; 31) vorgesehen sind, um ein elektrisches Rückkopplungssignal zu den Steuerventilen (20, 21; 24, 25) vorzusehen, und zwar in Erwiderung auf eine Druckdifferenz an einer und/oder der anderen Betätigungseinrichtung (10A; 10B).

5. System nach Anspruch 4, dadurch gekennzeichnet, daß ein erster Druckdifferenz-Rückkopplungssignal-Generator (30) vorgesehen ist, um ein Rückkopplungssignal aufgrund eines Druckunterschiedes an einer Betätigungseinrichtung (10A) vorzusehen, und daß ein zweiter Druckgenerator (31) an der anderen Betätigungseinrichtung (10B) vorgesehen ist, daß jeder Generator sein Rückkopplungssignal an die beiden elektrischen Streifen (15—18) abgibt, die an das primäre Steuerventil (20; 25) der zugehörigen Betätigungseinrichtung angeschlossen ist.

6. System nach Anspruch 4 oder 5, dadurch gekennzeichnet, daß die Druck-Rückkopplungssignal-Höhe eingerichtet ist auf weniger als 10% des Ventilsteuerstromes.

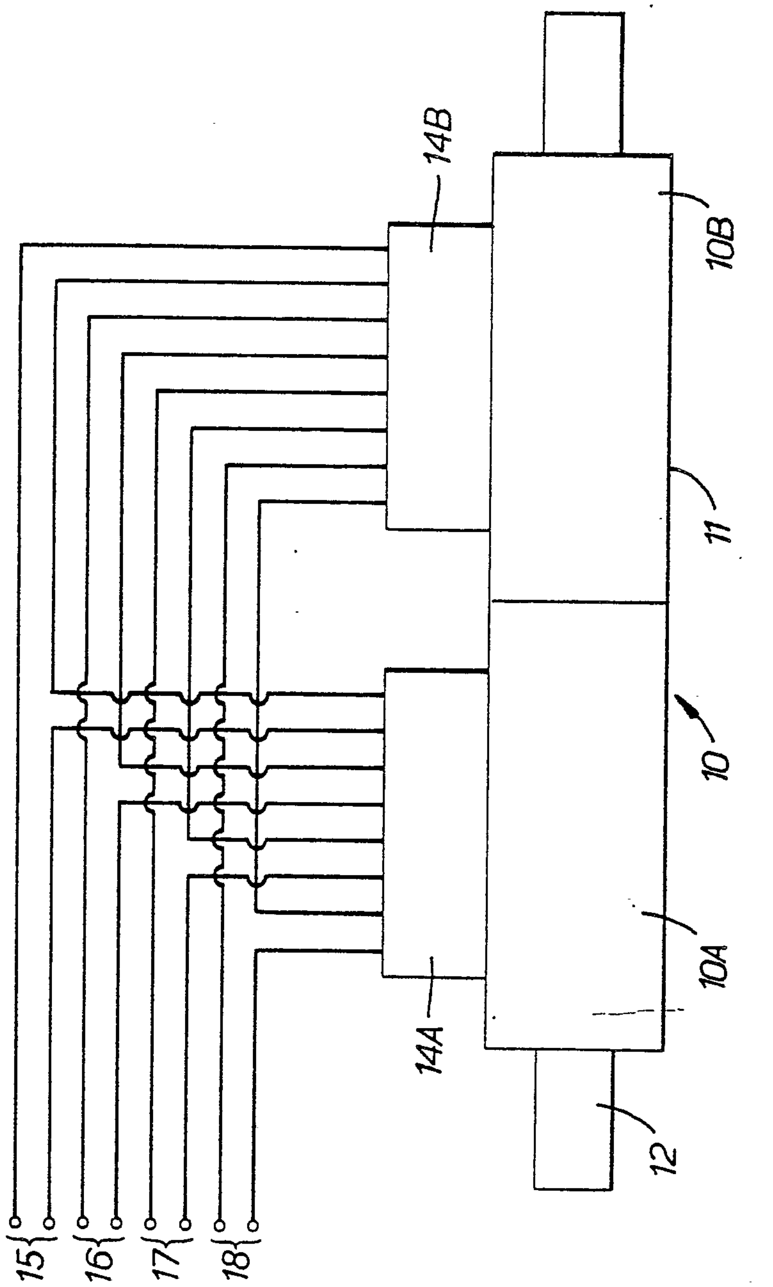

FIG. 1.

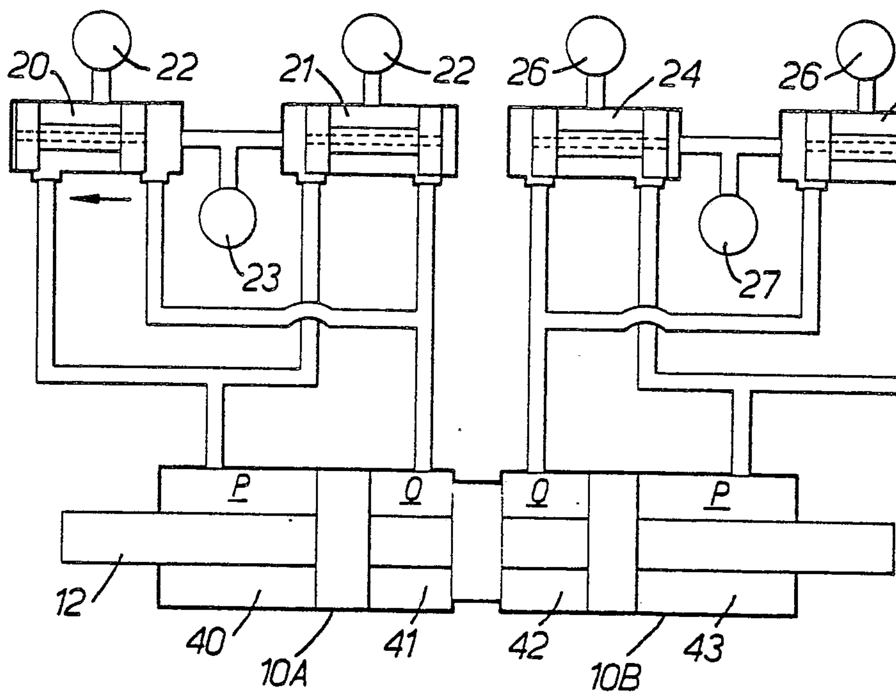
20
22
21
22
26
24
26
25
23
27
P
Q
Q
P
12
40
10A
41
42
10B
43

FIG. 2A.

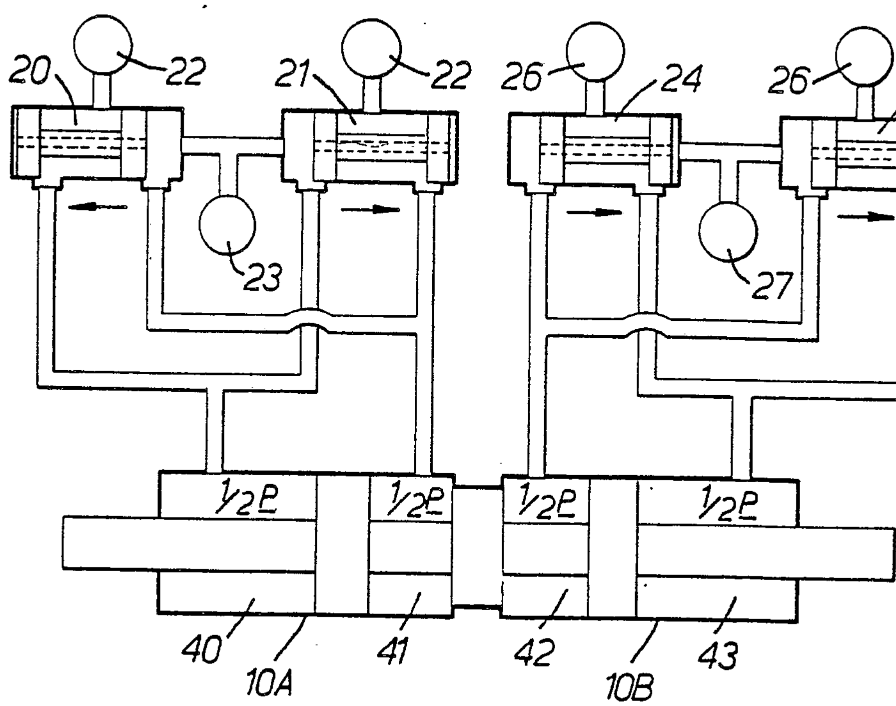
20
22
21
22
26
24
26
25
23
27
1/2P
1/2P
1/2P
1/2P
40
10A
41
42
10B
43

FIG. 2B.

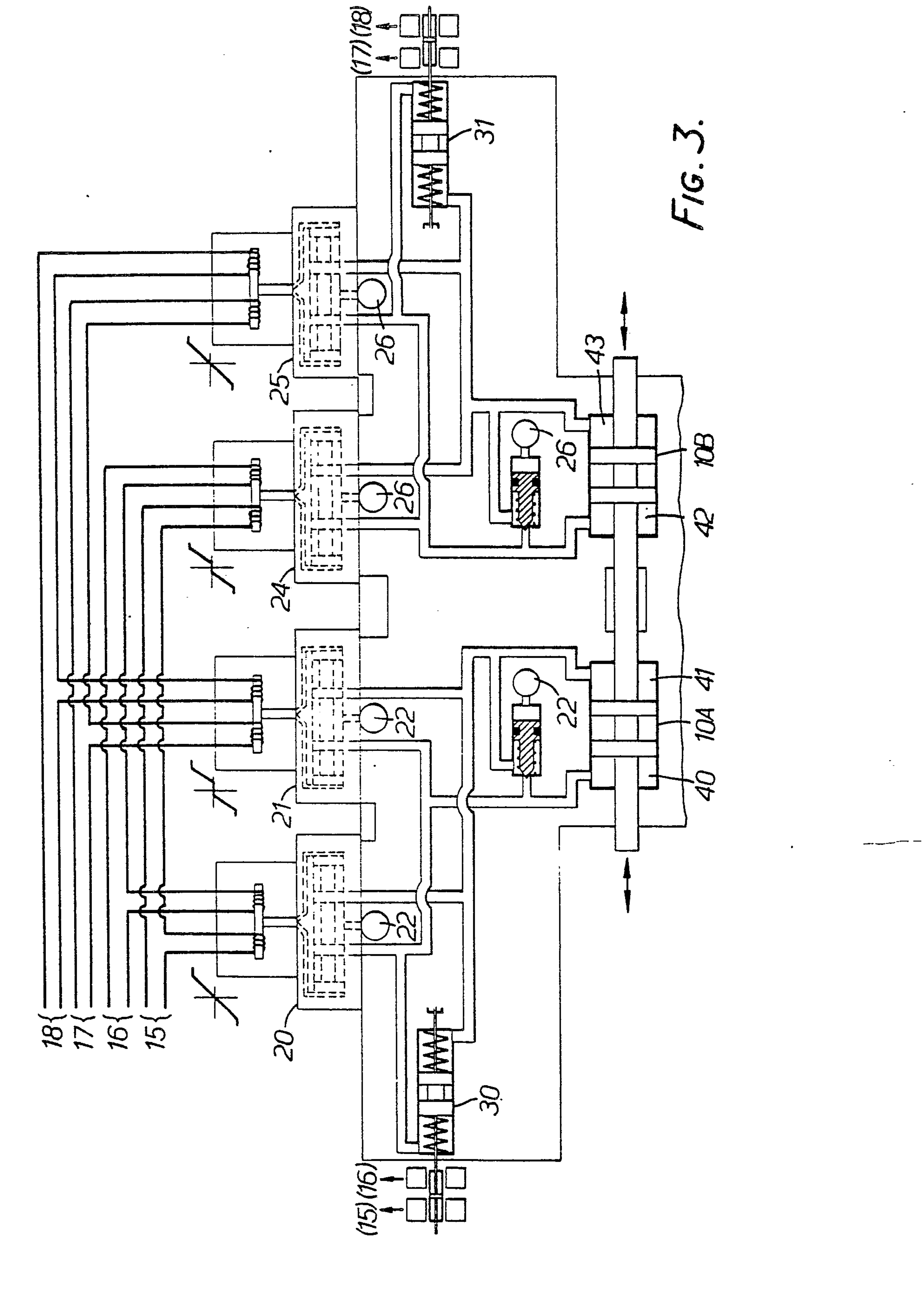
(17)(18)
31
25
26
26
26
43
10B
42
24
21
22
22
22
41
10A
40
20
18
17
16
15
30
(15)(16)

FIG. 3.