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(54) Hydraulic control system.

(57) An hydraulic control system, for example for use in moving control surfaces of aircraft, comprises first and second actuators (12A, 12B) operating in tandem. The two actuators are supplied by independent hydraulic pathways (P1, P2), the fluid in each of these pathways being controlled by a plurality of direct drive control valves (4) connected across the two pathways. The system will therefore operate even if three of the control valves, and one of the hydraulic supplies are non-operational.

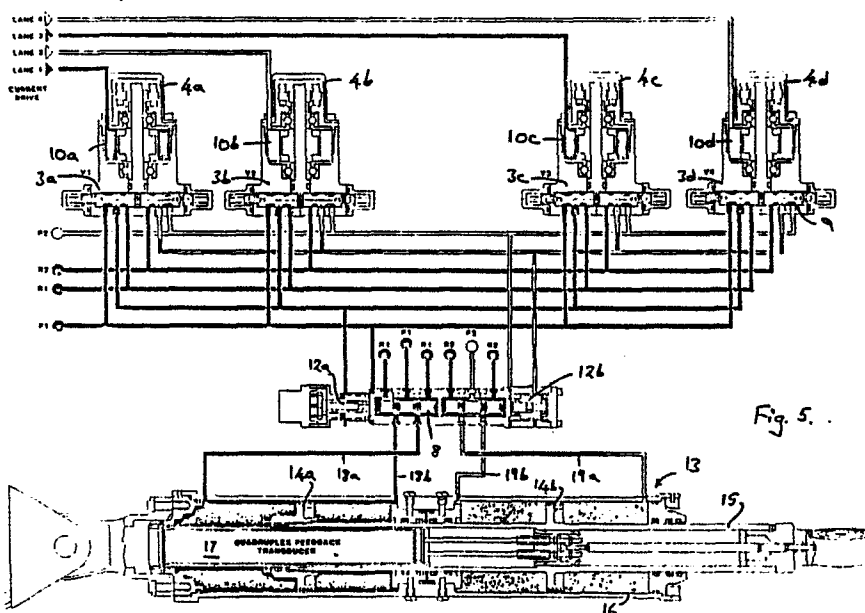


Fig. 5.

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HYDRAULIC CONTROL SYSTEM

The present invention relates to a hydraulic control system.

The invention is particularly concerned with the control of movable members provided with first and second actuators which may be arranged, for example, in parallel or in tandem. The movable member may be a movable load of substantial value which requires to be positioned and moved with a high degree of accuracy and reliability. For example, the movable member may be a hydraulic control valve. A particular but not exclusive application of such hydraulic control valves is in the control of actuators for use in moving control surfaces of aircraft.

The methods and systems according to the invention are applicable for the control of high performance aircraft and the systems may be constructed to be of high integrity and to provide multi-redundant electrohydraulic actuation.

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 in the hydraulic system or its control system. 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 is well known 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 as servo control signals to control the output position of the actuator. 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 required to transmit the full control forces, has nevertheless involved precision of operation to provide accurate manual control.

With the need to develop aircraft having ever higher performance, and the development of electronics enabling sophisticated on-board computer systems to be employed, it has become desirable to take advantage of the greater aero-dynamic efficiency which can be achieved with an aircraft which is inherently unstable. With such an aircraft the aero-dynamic 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 manual control. With the development of computer systems a computer may be used to continuously "fly" the aircraft and thus replace 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 direct mechanical linkage in the aircraft, since it would in any case be of no substantial use to the pilot as a fall-back system.

In order to provide the requisite degree of reliability and redundancy, it has become usual in such high performance aircraft to duplicate the hydraulic systems and to provide quadruplex control lanes. It would be possible in such approach to regard one hydraulic system and three electrical lanes as "back-up" and only to switch over to these upon failure of the primary hydraulic system or control lane. This approach however would require means to detect failure and to effect the necessary switch over, thus introducing further possible sources of failure. The usual approach is therefore to employ both hydraulic systems and all four electrical lanes simultaneously, although this in itself brings certain disadvantages as will be explained.

The usual design philosophy in such multi-redundant system is to provide an arrangement which can survive at least two failures, one of which may be hydraulic. This requires at least three electrical lanes and duplex hydraulic systems. Two electrical lanes are insufficient, because it is desirable to be able to identify a faulty lane by comparing it with the remaining good lanes. With a total of only two lanes, the faulty lane could not be eliminated in this way. Two hydraulic systems are

sufficient because a hydraulic failure will simply lead to loss of system pressure and no advantage is gained by comparing one hydraulic lane with another.

In the conventional approach having quadruplex electrical lanes two electrical lane failures may be survived provided they do not occur simultaneously. Following the first lane failure, the faulty lane may be identified by a comparison process and then eliminated from the system.

A potential disadvantage with multi-redundant systems of this type is the difficulty of correctly matching all the electrical and hydraulic lanes with each other to prevent "force-fighting" and parasitic loss as will be described hereinafter.

There are two types of force-fighting. The primary, potentially catastrophic type is where one electrical lane receives a large faulty signal and completely overpowers the remaining lanes.

The secondary type, more likely to occur but less serious, may arise from natural differences which will exist between the control lanes arising from tolerances of manufacture and assembly.

Parasitic loss may arise where two hydraulic control valves are connected in parallel between a source of hydraulic pressure and an actuator. If the zero or no flow positions of the valves are not exactly matched, one valve may be slightly open while the other is shut. This would lead to undesired actuator movement. In practice, because position feed-back is employed, the system sets itself so that the two valves are each slightly open in opposite senses. This results in a small flow of hydraulic fluid through the two valves to the return line. This is known as parasitic flow and represents a power loss.

A known control system is illustrated by European Patent Application No. EP-A-0092972 of the present applicants. This system proposes that, between the main valve which is to be controlled and the four electrical control lanes conventionally provided in a high performance aircraft, duplex hydraulic control systems are provided comprising first and second actuators for moving the main valve, each of which actuators is controlled by a pair of hydraulically parallel-connected electrohydraulic spool valves.

Parasitic flow between the valves of the respective pairs is avoided by providing one valve of each pair with a significant overlap at the zero point, so that no flow is provided for a significant range of spool movement either side of the zero point.

Each electrohydraulic valve comprises a so-called "flapper" or "jet pipe" which in response to an electrical input moves between a pair of orifices or receivers respectively and thus controls the flow through these orifices. This flow control is used to

vary the pressure conditions at each end of the spool and thus controls the spool movement. In order to operate, the valve therefore requires a source of hydraulic fluid pressure, and commercially available valves are arranged also to control hydraulic flow, by means of the spool, from the same source as that required for valve operation. Thus, such an electrohydraulic valve will operate on and control a single hydraulic supply. Therefore, with only two hydraulic supplies available, it clearly is best to connect each supply only to two valves. If one hydraulic supply should fail, two valves would still be operational. But, if each valve were to be connected only to one electrical lane, this would automatically eliminate the effectiveness of two lanes, even although the electrical signals on the lanes might be functioning correctly. To avoid this difficulty in EP-A-0092972 each lane is connected to two valves supplied by respective hydraulic supplies. To permit this, each valve needs two operating windings, resulting in a complex circuit arrangement. A disadvantage of this arrangement is that a fault in one electrical lane adversely affects two valves, so that, in the worst case, only two electrical lane failures could cause all four valves to malfunction.

A further disadvantage with this arrangement is that if one hydraulic supply should fail so that the system is forced to rely on the other hydraulic supply, only one pair of electro-hydraulic valves would be operational. Thus, failure of one hydraulic supply automatically eliminates one pair of valves and their corresponding driving mechanisms. Therefore, should one of the two hydraulic supplies fail, the entire burden of controlling the fluid supplied to the actuators is borne by the driving mechanisms of only one of the two pairs of valves. This is clearly disadvantageous, since it is likely to result in a substantial degradation of performance.

Another disadvantage is in the use of electrohydraulic valves which, even in their null positions, have a continuous flow which causes a power loss of about 1/2 kilowatt per valve. Also, this type of valve is not particularly reliable since it is susceptible to contaminants owing to the small size of the orifices controlled by the flapper and to the fact that a mechanical feed-back arrangement is employed using a wire. In spite of their known disadvantages, electrohydraulic valves were previously employed because they require only a small operating current of about 10 mA and because they are well-known components whose properties are well investigated.

It has been proposed however to control a main hydraulic actuator having duplex hydraulic systems and quadruplex electrical systems by means of a tandem spool valve directly driven by four high power electrical torque motors, thus elimi-

nating the first stage altogether. The servo valves may alternatively in this system be connected side by side. A disadvantage of this proposal is that any force-fighting of the first type which may occur between the torque motors may lead to a situation in which the system is unable to control the main actuator adequately.

A tandem spool valve directly driven by several high power electrical torque motors is particularly susceptible to this force-fighting problem owing to the fact that the output of each torque motor is not limited in any particular way and will increase in dependence upon the size of an input control current. Thus, if an abnormally high control current is supplied as a result of a system fault, one torque motor would, in the worst case, overpower the remaining three torque motors and lead to complete system failure. This type of failure is particularly associated with systems directly controlled by means of electrical current.

Having several torque motors acting together also incurs force-fighting of the secondary type when the system is operating normally, as a result of the unavoidable differences which will exist between the control lanes arising from manufacturing or assembly tolerances. Such secondary force-fighting either has to be accepted, in which case there will be an undesirable heating effect in the motor coils, or can be detected and neutralised by use of special circuitry which naturally adds to complexity and expense.

Thus, the above described system has serious practical short-comings.

It is an object of the present invention to provide a hydraulic control system which at least alleviates some of the problems of the prior art. It is a further object of the invention to provide a method of hydraulic control.

According to the present invention there is provided a hydraulic control system comprising first and second actuators operating in parallel or tandem and being separately supplied with fluid along first and second fluid paths, and control valve means including at least three independent electrical actuating means, the control valve means being arranged for operation, on actuation by any one or more of the electrical actuating means, for the common control of the fluid respectively in the first and second paths.

Preferably, the electrical actuating means comprise at least three electrical coils, or other windings.

The control valve means may comprise a single valve having at least three coils for operation thereof; alternatively, it may comprise at least two control valves, each having at least two coils for operation thereof; alternatively again, it may comprise at least three control valves, each having

exactly one coil for operation thereof. The control valve means may comprise one or more direct drive valves, either driven by an electrical torque motor or an electrical force motor. Alternatively, the control valve means may comprise at least one electro-hydraulic valve.

In one particularly advantageous arrangement, the control valve means comprises at least one spool valve, the or each valve having a spool extending between the first and second paths.

The actuators may comprise a further fluid valve which may be, for example, a spool valve. This further fluid valve may be arranged to control fluid within two further independent fluid paths. The fluid in these two further paths may be arranged to control a further actuator or spool.

Preferably, the control valve means is arranged for operation by exactly four electrical coils, each coil being independently energised via a separate electrical pathway.

In the present specification and claims the word "hydraulic" is used in its broad sense, and is of course not restricted to arrangements using water.

In the arrangement of the present invention, failure of one hydraulic supply does not automatically eliminate any part of the valve control means, for example any one of the individual valves which may make up the valve control means, since the two hydraulic supplies are each connected to all of the valves. In other words, the two hydraulic supplies are connected in parallel. With this arrangement, it is possible, although not essential, for the driving mechanism (for example the torque motor) of each valve to manage with only one winding. The resulting system is therefore simpler and probably more reliable than the currently known systems.

In addition, use of direct drive valves, driven by torque motors or force motors, means that the disadvantages of electro-hydraulic valves, as previously noted, are avoided. These advantages are obtained without, at the same time, reaping the disadvantages of earlier proposals to use direct drive valves (that is, that one drive motor could, in the worst case, overpower the other drive motors and lead to complete system failure, again as previously noted) by the fact that in the preferred embodiment of the present invention, the spool driven by the respective motor has only a limited possible stroke.

The present invention also avoids the use of special fault detecting equipment, which is a feature of many present arrangements.

The present invention may be carried into practice in a number of ways, and three specific embodiments will now be described, by way of example, with reference to the drawings, in which:

Figure 1 shows a perspective view of a first embodiment of a valve assembly;

Figure 2 shows a partially cut away view of the valve assembly of Figure 1;

Figures 3A and 4 are a schematic diagram of a hydraulic system incorporating the valve assembly of Figures 1 and 2, Figures 3B and 3C showing alternative arrangements to Figure 3A representing respectively second and third embodiments; and

Figure 5 is a more detailed system diagram corresponding to Figures 3A and 4.

Referring first of all to Figure 1, a valve assembly 1 comprises a main valve 2 arranged to be driven by first and second actuators 12a and 12b. Each actuator is connected to each of four control valves 3a, 3b, 3c and 3d having respective electrical torque motors 4a, 4b, 4c and 4d connected for direct drive of the respective control valves. The main valve 2 is a spool valve and connected to its spool are first and second position feed back transducer assemblies 6a and 6b each of which preferably comprises a pair of linear variable differential transformers (hereinafter LVDT). Thus, a total of four LVDT's is provided thus permitting quadruplex electrical feed back of the spool position for closed loop servo control. Electrical signals for this purpose are available at respective electrical connection sockets 5a, 5b, 5c and 5d the last of which is not visible in Figure 1.

Figure 2 is a cut-away version of Figure 1 thus enabling the spool 8 of main valve 2 to be seen and also permitting the individual LVDT's 11a, 11b, 11c and 11d to be seen. It will also be apparent from this Figure that each control valve 3a, 3b, 3c and 3d has a respective valve spool 9 and that each valve spool 9 is directly connected to the shaft of a respective one of the torque motors 4a to 4d which have respective coils 10a to 10d, and rotary feed-back transducers 20a to 20d for closed loop servo control of position.

Each torque motor operates through a limited angle in the range of 5 to 30° and thereby causes linear motion of the respective valve spool by means of a respective spherical ball joint 21a, 21b, 21c or 21d between the motor shaft and the spool which is offset from the axis of rotation of the motor. The spherical ball is not illustrated in the Figures.

As an alternative to the use of electrical torque motors, it is equally possible to operate the control valves by respective linearly moving force motors mounted on the axes of the valve spools 9.

Preferably, each valve spool 9 is provided with a return spring and in addition or as an alternative may have multi-redundant electrical positional feed-back for closed loop servo control.

Such return spring and/or positional feedback is provided in the illustrated embodiment.

Referring now to Figures 3A and 4, the inter-connection of the various components of the valve assembly may be seen schematically. In addition to the components of the valve assembly of Figures 1 and 2, these figures also illustrate a second stage or main actuator 13 provided with quadruplex feedback transducers 17, preferably LDVT's, for closed loop servo control of position.

The control connections are indicated purely - schematically and it may be seen that each of the four first stage valves 3a to 3d is connected to control the first stage actuator 12a via control lines C1 and is also connected to control the second actuator 12b via control lines C2 which are independent of control lines C1. The first stage actuators 12a and 12b directly control the second stage valves 8, which may be referred to as the main valves, which in turn control via two independent control lines C3 and C4 two independent hydraulic piston and cylinder assemblies of the second stage main actuator 13.

Alternative arrangements to those shown in Figure 3A are shown in Figures 3B and 3C. They show, respectively, an arrangement in which two first stage valves are provided, each having two coils (and, possibly, two separate corresponding force motors or torque motors), and an arrangement in which a single first stage valve is provided, this valve having four coils (and, possibly, four independent torque or force motors).

Reference will now be made to Figure 5 which corresponds to Figures 3A and 4 and which shows further detail of the construction of the torque motors 4a to 4b, further detail of the connection of the hydraulic lines and further internal detail of the first stage valves, second stage valves and first and second stage actuators.

It may be seen from the Figure that two independent high pressure hydraulic supplies P1 and P2 are provided which have respective return lines R1 and R2. The hydraulic fluid pressure is preferably 27 MN/m² (4000 psi nominal).

It will be observed from Figure 5 that each of the control valves 3a to 3d provides two independently controllable hydraulic porting arrangements on a common spool. Each porting arrangement is connected to a respective one of the hydraulic supplies P1 and P2. In addition, each of the supplies P1 and P2 is connected to one side of a respective one of the first stage actuators 12a and 12b. The other side of each of the first stage actuators 12a and 12b is connected to a respective one of the hydraulic porting arrangements of each of the valves 3a to 3d.

Thus, displacement of the spool 9 of each of the valves 3a to 3d couples the controllable side of each first stage actuator progressively either to hydraulic high pressure or to hydraulic return pressure. In fact, each hydraulic porting arrangement of each of the first stage valve 3a to 3d is such as to reduce the system pressure by approximately half and to supply this to one side of each of the first stage actuators 12a and 12b when the spool 9 is in its undisplaced or central position. Thus, each of the actuators 12a and 12b is provided with system pressure on one side and 50% of system pressure on the other side in the neutral position. The actuators are balanced by arranging for the unequal pressures to be applied to unequal areas in the ratio of approximately 2:1.

When the first stage actuators are required to move, each first stage valve moves such that the pressure to the larger area (to which it is connected) is either increased or reduced thus providing a net force to move the main valve spool 8.

For the above described system, each of the control valves 3a to 3d requires two three-port configurations. It is equally feasible to use two four-port arrangements and in this case the first stage actuators will have equal piston areas and the two active chambers will be controlled differentially.

The main valves are arranged on a common tandem spool 8 and are each arranged to control a respective hydraulic piston 14a or 14b of the main actuator 13. In the illustrated case, a conventional 4-port arrangement is employed and as the spool 8 displaces pressure on one side of each piston 14a and 14b tends to increase whilst it tends to reduce on the other side.

The pistons 14a and 14b are connected on a common hollow shaft 15 in a housing 16.

Quadruplex feedback transducers 17, preferably LDVT's, are provided within the shaft 15 for position feedback control.

The main concept of the above described system lies in the fact that each first stage valve is a duplex arrangement and a multi-redundant system is obtained by the addition of several such duplex valves by flow summation to control the first stage actuators 12a and 12b. Thus, each of the torque motors 4a to 4d drives a tandem spool arrangement.

Using this concept, force fighting between the tandem pair of first stage actuators may be virtually eliminated during manufacture of the first stage valves by accurate port matching control, since in each case the hydraulic porting arrangements for opposite senses are on a common spool. Mismatch between the electrical lanes 1 to 4 of the respective torque motors does not induce such force fighting, and no feed-back is necessary to achieve this.

It is intended that mismatch between the several first stage valves 3a to 3d will be minimized by accurate mechanical adjustment of the hydraulic and electrical datums to ensure that these are closely coincident. Furthermore, any residual electrical mismatch between the lanes may be minimized by an equalisation technique which reduces or eliminates the level of the steady state motor current.

If after these processes have been applied residual mismatch remains such as to cause parasitic leakage to occur between the several valves, such leakage will be of a secondary nature by virtue of the relatively low flow capability required for the first stage valves, because the 1st stage actuators are of relatively low power capability and hence have relatively small swept volumes.

Hydraulic integrity is provided by a duplex tandem arrangement throughout.

Integrity of the system is enhanced by the fact that the multi redundant electrical control systems are totally separated at the motors and the motors themselves are also physically separated. An electrical hardover of one motor leading to a hardover of the associated control valve cannot overpower the remaining motors because the motors are not connected to be force summing.

Mechanical integrity is provided at the first stage by a similar philosophy to that applied to the electrical integrity. If a first stage servo valve is mechanically jammed, the flow summation technique employed ensures that the remaining valves can overpower the jammed valve and that the system as a whole can continue to operate. Thus, the system can tolerate a single valve electrical or mechanical hardover without immediate corrective action being required. It is clearly a prerequisite for this advantage to be achieved at least three first stage valves are provided.

Integrity of the second stage valve is ensured by the provision of a sufficiently large first stage actuator area and force to overcome any definable jam condition.

It will thus be appreciated that direct-drive techniques have been employed in a way which leads to no loss of system integrity or reliability and no loss of redundancy. The advantages of the direct-drive technique may therefore be achieved without suffering the disadvantages previously associated with this approach. It will also be appreciated that within the scope of the invention any electrically operated drive means may be used for the first stage valves 3a to 3d. Thus not only direct-drive torque motors and electrohydraulic valves providing indirect drive may be employed, but also linear motors or solenoid type actuation systems acting directly or indirectly on the valve spools.

Finally, it will of course be clear that although the present application does not include any specific diagrams corresponding to Figures 1, 2 and 5 for the arrangements of Figures 3B and 3C, a skilled man will have no difficulty in constructing specific embodiments.

Claims

1. An hydraulic control system comprising first and second actuators operating in parallel or tandem and being separately supplied with fluid along first and second fluid paths, and control valve means including at least three independent electrical actuating means, the control valve means being arranged for operation, on actuation by any one or more of the electrical actuating means, for the common control of the fluid respectively in the first and second paths.

2. An hydraulic control system as claimed in claim 1 in which the electrical actuating means comprise at least three electrical coils.

3. An hydraulic control system as claimed in claim 2 in which the control valve means comprises a single valve having at least three coils for operation thereof.

4. An hydraulic control system as claimed in claim 2 in which the control valve means comprises at least two control valves, each having at least two coils for operation thereof.

5. An hydraulic control system as claimed in claim 1 in which the control valve means comprises at least three control valves, each having exactly one coil for operation thereof.

6. An hydraulic control system as claimed in any one of the preceding claims in which the control valve means comprises at least one direct drive valve, the valve being driven by an electrical torque motor under control of the electrical actuating means.

7. An hydraulic control system as claimed in any one of claims 1 to 5 in which the control valve means comprises at least one direct drive valve, the valve being driven by an electrical force motor under control of the electrical actuating means.

8. An hydraulic control system as claimed in any one of claims 1 to 5 in which the control valve means comprises at least one electrohydraulic valve.

9. An hydraulic control system as claimed in any one of the preceding claims in which the control valve means comprises at least one spool valve, the or each valve having a spool extending between the first and second paths.

10. An hydraulic control system as claimed in any one of the preceding claims in which the actuators comprise a further fluid valve.

11. An hydraulic control system as claimed in claim 10 in which the further fluid valve is a spool valve.

12. An hydraulic control system as claimed in claim 10 or claim 11 in which the further fluid valve is arranged to control fluid within two further independent fluid paths.

13. An hydraulic control system substantially as specifically described with reference to Figures 1, 2, 3A, 4 and 5; or Figures 1, 2, 4 and 5 modified as shown in either Figure 3B or Figure 3C.

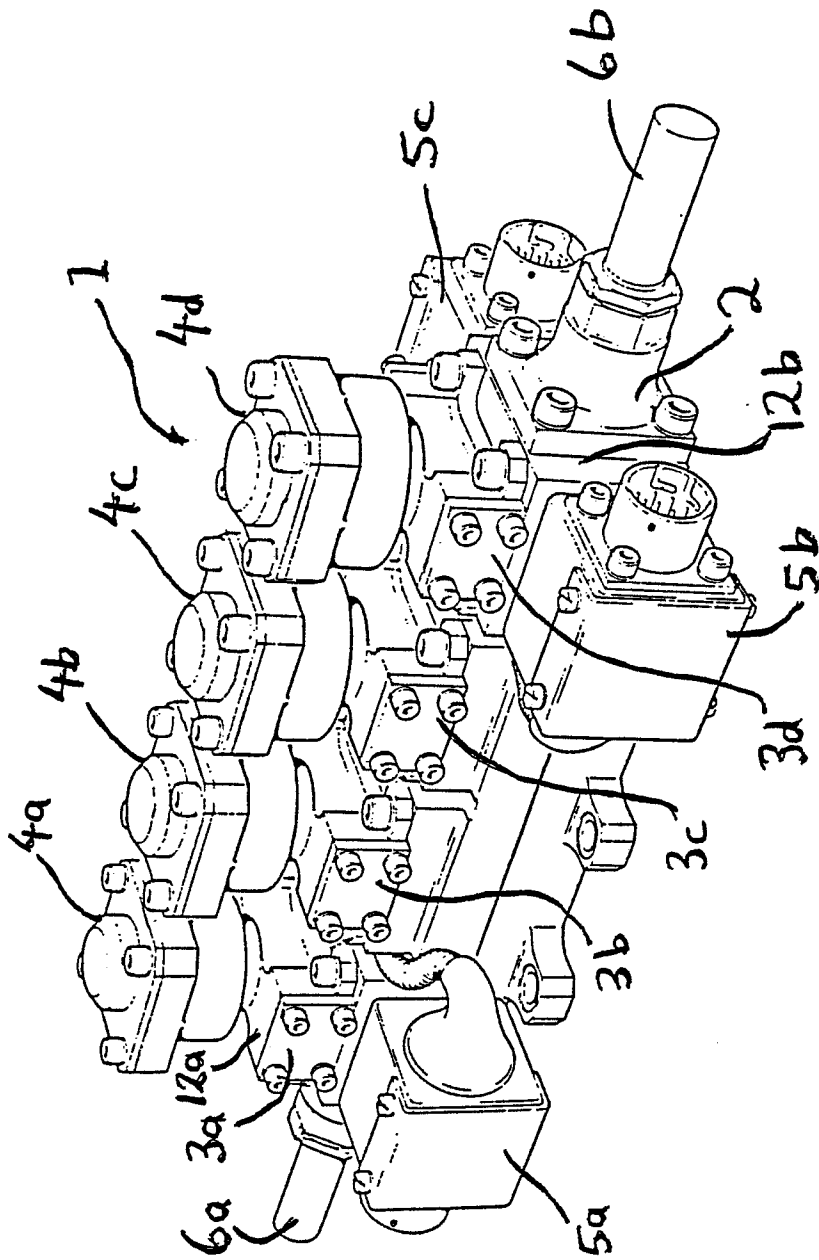


Fig 1

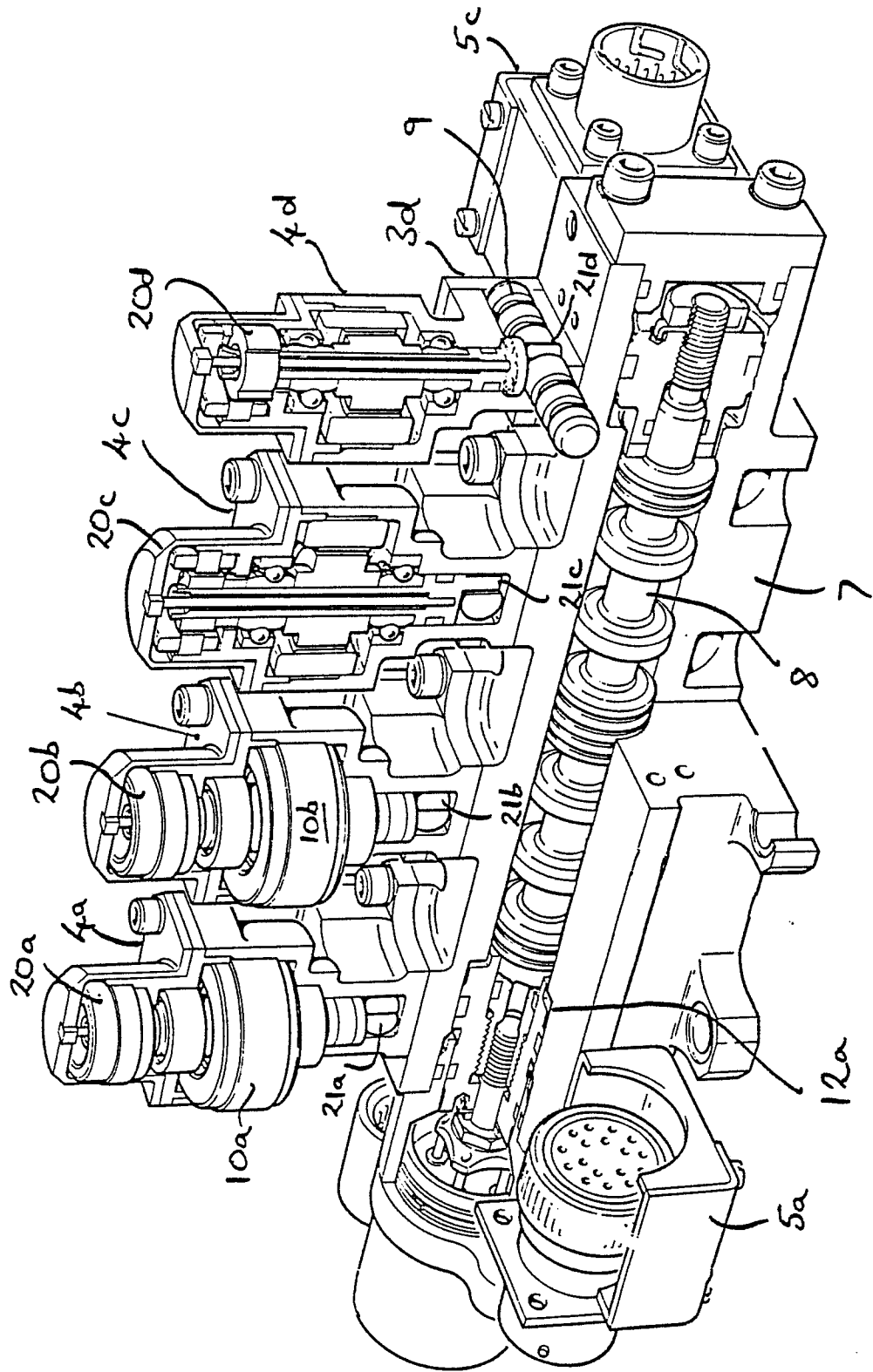
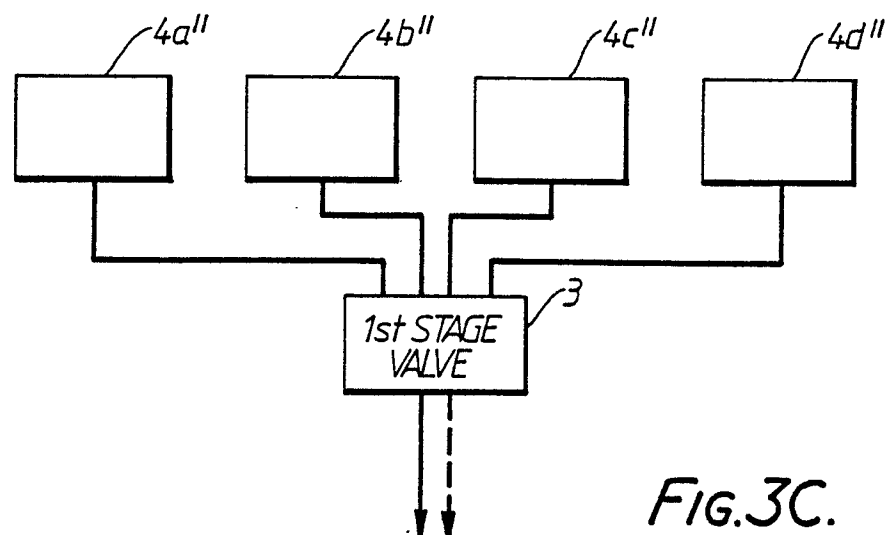
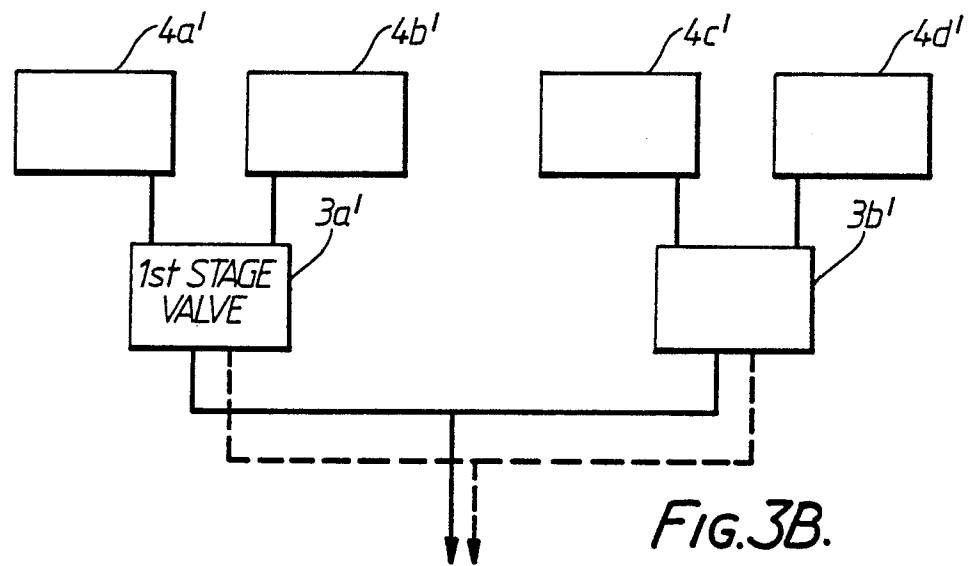
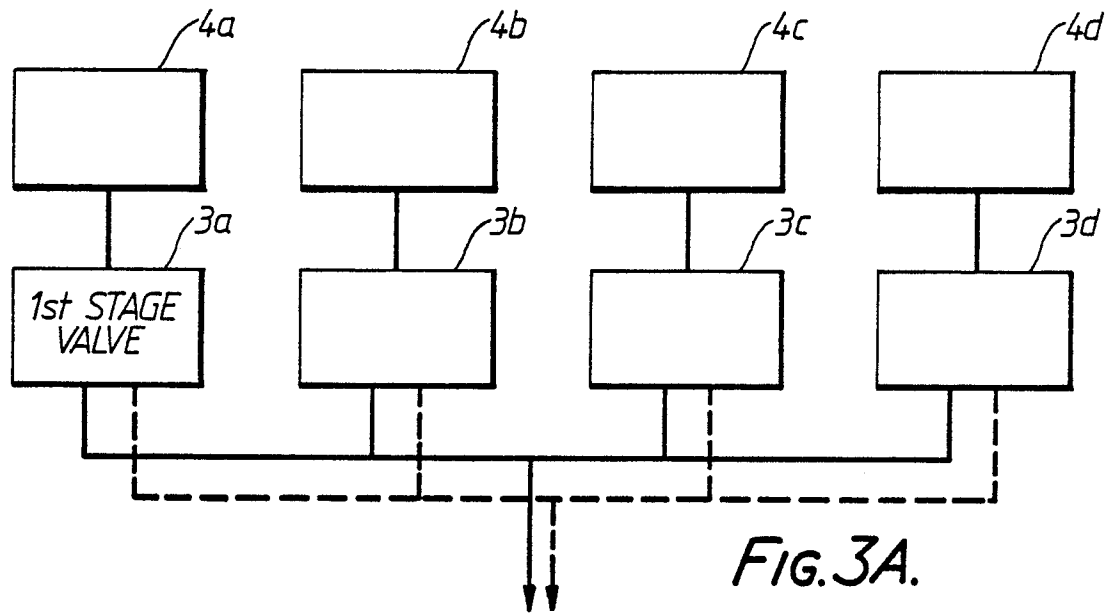
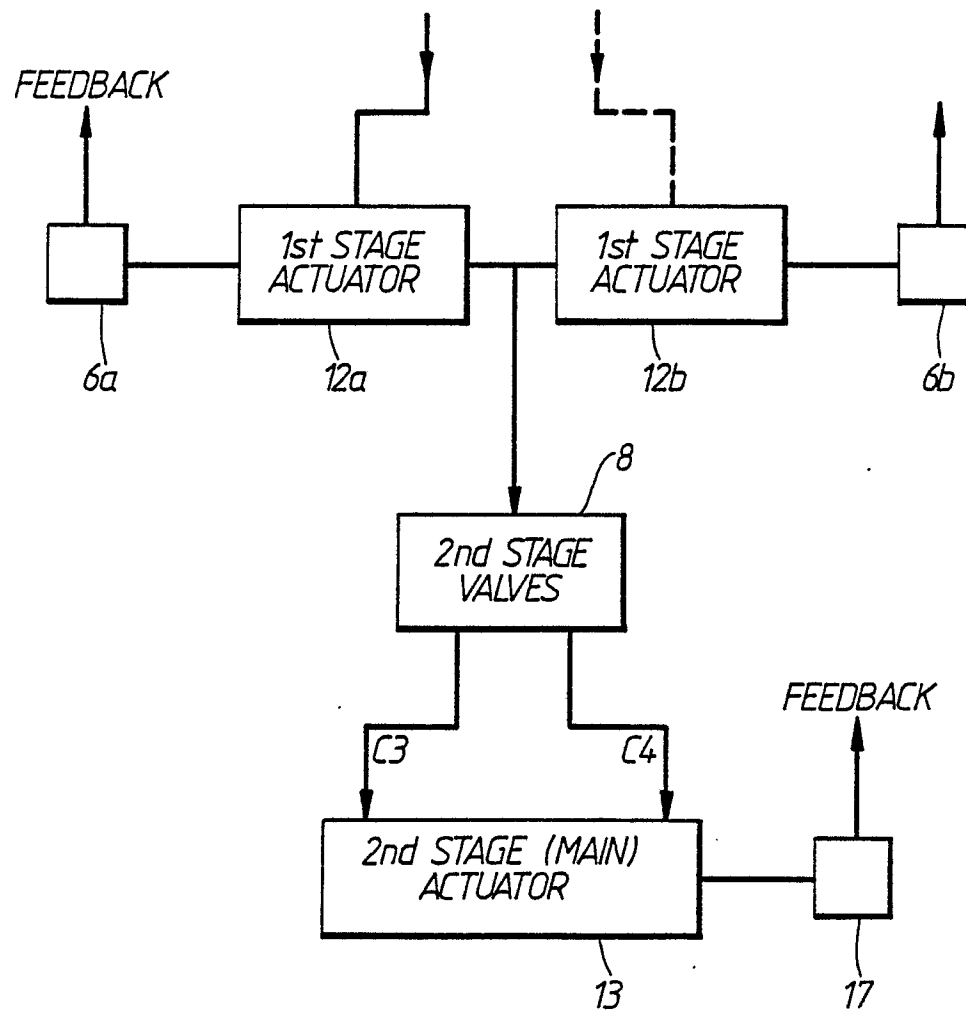


Fig 2.



**FIG.4.**

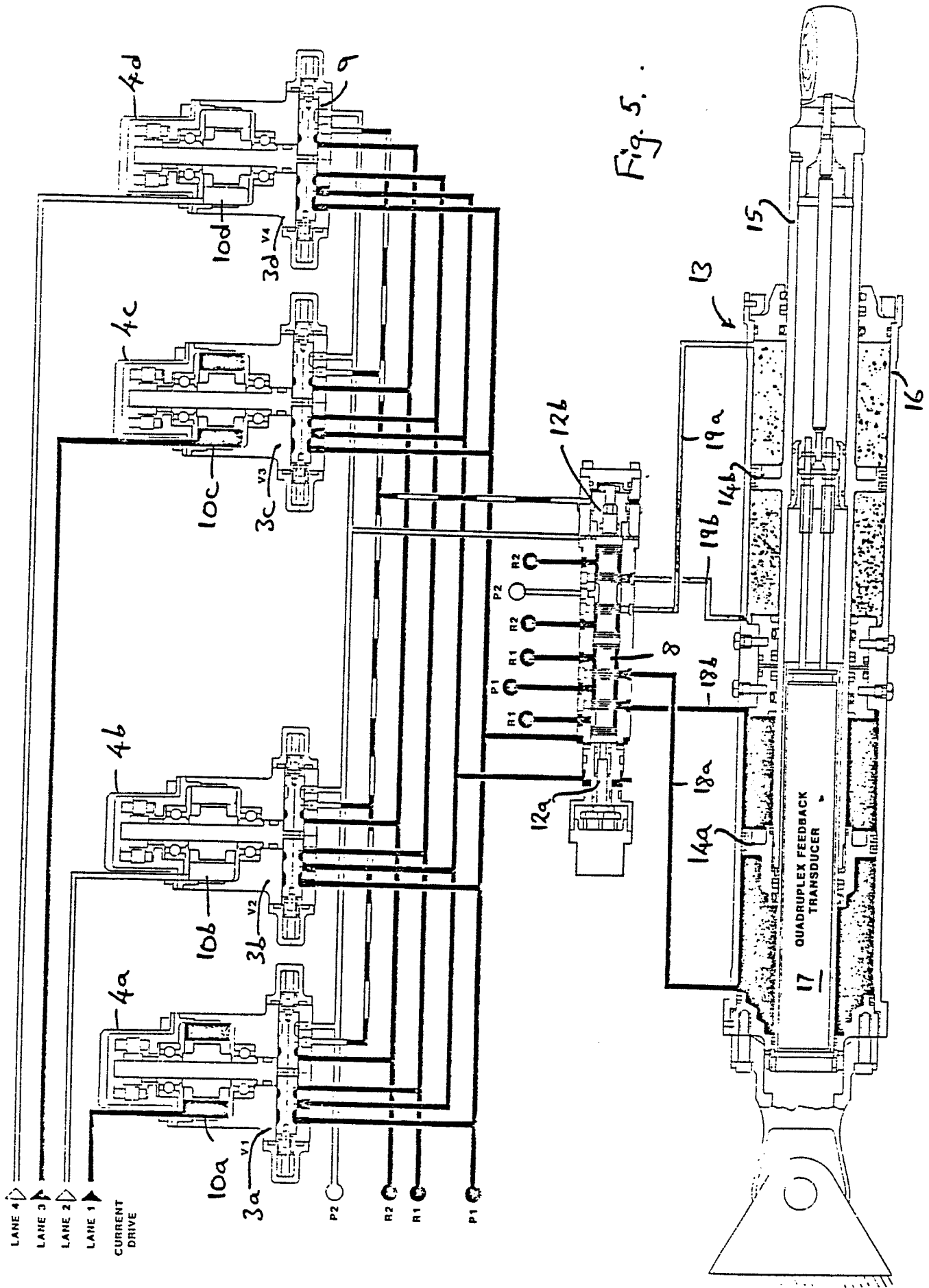


Fig. 5.