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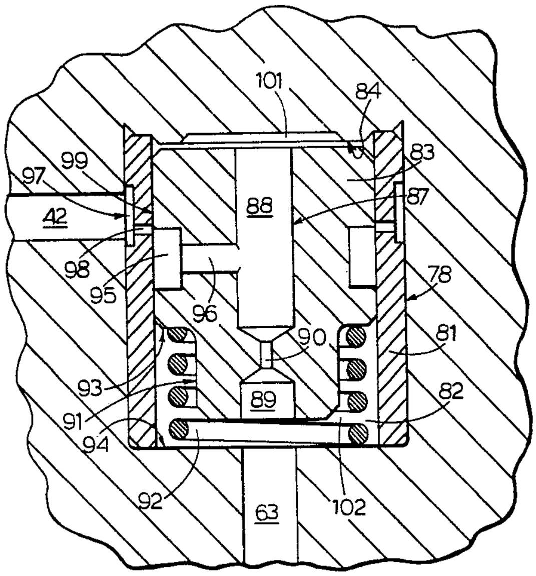
(57) Abstract

Anti-skid brake control systems which sense the presence of skid conditions at a hydraulically braked wheel and then relieve the break pressure at that wheel by displacing from the brake. A regulator (80) is connected in a high pressure fluid supply line so as to control the rate at which the brakes are re-applied on cessation of the skid conditions. The regulator preferably comprises a cup-shaped piston (83) reciprocable within a cylinder bore (82). A first restriction (90) of predetermined size is defined by the end wall of the piston and a second restriction of variable size is formed between a number of radial ports (98) in the bore wall and an annular recess (95) in the outside of the piston, this recess being permanently connected to the inner space (88) of the piston. Fluid flows from the variable restriction to the predetermined restriction and a pressure difference is produced across the predetermined restriction. Any increase in this pressure difference causes the piston to move against the action of a compression spring (92) to restrict the size of the variable restriction, thereby regulating the flow of fluid to a predetermined value. The regulator is suitable for use with systems of the kind which include a brake pressure modulator and with those which include a control valve to dump fluid from the brakes on occurrence of skid conditions.



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ANTI-SKID BRAKE CONTROL SYSTEMS

This invention relates to anti-skid brake control systems which sense the presence of skid conditions at a hydraulically braked wheel and then relieve the brake pressure at that wheel by displacing fluid from the brake, so as to overcome the skid conditions.

In systems of this kind it is desirable to restrict the rate at which the hydraulic pressure in the brake is re-developed when the skid conditions have been overcome, to enable the system to monitor what is occuring during re-application of the brakes. A orifice has been used to control the re-development of brake pressure but when a relatively high pressure fluid supply is used, the diameter of the orifice can become unacceptably small. A very small orifice can be difficult to machine with accuracy and can easily become blocked with foreign matter, thereby preventing re-application of the brake.

According to the invention, an anti-skid brake control system comprises a wheel having a hydraulic brake applicable by supplying thereto fluid under pressure, means to sense the presence of skid conditions at the wheel during braking and thereupon to produce a skid signal, and a regulator arranged to regulate flow of fluid to the brake on cessation of the skid signal so as to re-apply the brake pressure at a controlled rate, the regulator comprising a first restriction of predetermined size connected in series with a second restriction defined between two relatively movable members, relative movement of the members being controlled by the difference in fluid pressure across the first restriction such that an increase in this pressure difference causes a reduction in the size of the second restriction.

The flow of fluid through the regulator tends to be substantially independent of supply pressure.

Since the first restriction is not the sole means of restricting the fluid flow, the opening defined by the first restriction can be made of greater size than would otherwise be possible.

Preferably the first restriction is located downstream of the second restriction. Thus, any foreign matter present in the fluid supply to the regulator tends to lodge in the second restriction, resulting in a drop in the pressure difference across the first restriction which in turn causes an increase in the size of the second restriction so that the obstruction tends to be released.

Preferably the first restriction is defined by one of the relatively movable members and that member is resiliently biassed against the action of the pressure difference across the first restriction.

The characteristics of the means by which the member is biassed may be chosen such that the majority of the pressure drop across the regulator occurs across the second restriction so that the opening defined by the first restriction can be made relatively large.

Conveniently, the relatively movable member defining the first restriction is a piston which is movable within a bore, and the bore wall is the other relatively movable member. In a particularly preferred arrangement, the piston is cup-shaped, the first restriction is defined by the end wall of the piston and the second restriction is defined between at least one radial passageway in the wall of the bore and an annular recess in the outside of the piston, the recess being in permanent communication with the inner space of the piston. Alternatively, the radial passageway or passageways may be formed on the wall of the cup-shaped piston and the annular recess may be formed in the wall of the bore. The annular recess eliminates the need for circumferential alignment of passageways in the two walls.

The regulator may be arranged such that the regulated flow of fluid controls re-application of the brake either indirectly or directly.

Where the regulated flow of fluid indirectly controls re-application of the brake, the system is preferably of the kind which includes a movable wall separating a pair of chambers, one of the chambers being connected to the hydraulic brake and the movable wall being arranged to increase the volume of that chamber in response to the skid signal so as to relieve the brake pressure at the braked wheel, the regulator being arranged to regulate the flow of fluid to the second of the chambers on cessation of the skid signal such that the movable wall is thereby moved progressively to reduce the volume of the first chamber so as to re-apply the brake pressure at a controlled rate. An assembly which incorporates such a movable wall and pair of chambers is called a brake pressure modulator.

Preferably a three-way valve is interposed between the second chamber and the regulator; the valve being responsive to the skid signal such that the second chamber normally is connected to the regulator but on production of the skid signal the valve is switched to connect the second chamber to an outlet for fluid. This arrangement allows the second chamber to be connected via the regulator to its fluid supply under normal (non-skid) conditions so that the pressure of that supply acting on the movable wall tends to maintain the volume of the first chamber (which is connected to the hydraulic brake) at a minimum value. On the occurrence of skid conditions the second chamber is isolated from its supply so that fluid may pass from the second chamber through the outlet under the pressure of fluid in the hydraulic brake circuit acting on the movable wall from within the first chamber.

A two-way valve is preferably connected in the fluid supply line to the hydraulic brake, the valve being arranged to close when the movable wall moves to increase the volume of the chamber connected to the hydraulic brake. Thus, the hydraulic brake is isolated from its fluid supply during skid conditions to assist in reducing the brake pressure.

In a much preferred embodiment of the invention, the movable wall and the first and second chambers, the regulator, the three-way valve and the two-way valve are all contained within a single housing which may be formed in one part or as many parts as is necessary for manufacturing convenience. Thus, a major proportion of the anti-skid brake control system here described may be provided in a single low-cost unit which may easily be installed or replaced.

A common source of hydraulic fluid may be used to supply the regulator and the hydraulic brake.

Where the regulated flow of fluid directly supplies the brake during re-application, the system is preferably of the kind which includes an electromagnetically actuated valve arrangement connected to the fluid supply for applying the brakes, the hydraulic brake and a reservoir, and being responsive to the skid signal such that the valve arrangement normally connects the brake to the fluid supply but on production of the skid signal the valve arrangement is switched to connect the brake to the reservoir, the regulator being arranged to regulate the flow of fluid from the fluid supply to the brakes on cessation of the skid signal. An assembly which includes such a valve arrangement is called a control valve assembly. This kind of system is said to dump fluid from the brakes on production of the skid signal, that is, the fluid displaced from the brakes is passed to a reservoir and is not necessarily returned to the brakes on cessation of the skid signal as in systems which incorporate a modulator assembly.

The valve arrangement may comprise, for example, a solenoid-controlled spool valve, a ball valve arrangement, or a combination of both.

The regulator may be connected in various arrangements which each have particular advantages.

In one preferred arrangement, the regulator is connected in parallel with the valve arrangement between the fluid supply and the hydraulic brake, and the valve arrangement is arranged not to connect the fluid supply to the brake on cessation of the skid signal until the brake pressure has been re-applied by fluid supplied via the regulator, the valve arrangement being adapted to cut off flow of fluid through the regulator during production of the skid signal.

In another arrangement the regulator is connected in parallel with the valve arrangement between the fluid supply and the hydraulic brake, and the relatively movable members of the regulator are adapted to cut off communication between the fluid supply and the valve arrangement until the brake pressure has been re-applied by fluid supplied via the regulator.

The regulator may also be connected between the fluid supply and the valve arrangement with a normally open further valve connected in parallel with the regulator, the further valve being arranged to close on production of the skid signal and to stay closed until the brake pressure is re-applied by fluid supplied via the regulator. The further valve is preferably a latch valve comprising a piston responsive to the pressure differential between the fluid supply and the brake such that an increase in this pressure differential above a predetermined valve results in closure of the latch valve.

In a further preferred arrangement, the regulator is connected between the fluid supply and the valve arrangement and the first restriction is defined by one of the relatively movable members, that member being responsive to the pressure differential between the fluid supply and the brake such that an increase in this pressure differential above predetermined valve causes relative movement of the members by an amount which is sufficient to close a by-pass connected across the regulator, the by-pass being arranged to stay closed until the brake pressure is re-applied by fluid supplied via the regulator. This arrangement avoids the need for a normally open valve in parallel with the regulator.

The regulator is preferably included in a control valve assembly comprising a housing, an inlet for connection to the fluid supply, a first outlet for connection to the brake and a second outlet for connection to the reservoir.

The reservoir to which fluid is dumped by the control valve assembly may be in the form of an accumulator from which fluid is pumped back to the master cylinder or to the brake by a suitable pump.

The fluid source which supplies the hydraulic brake with fluid during normal braking may be a hydrostatic master cylinder or a power valve type master cylinder which controls a high pressure hydraulic fluid supply.

When the fluid source is a hydrostatic master cylinder, fluid expelled from the second chamber of the modulator or from the second outlet of the control valve on the occurrence of a skid signal may be directed to a scavenge pump which returns the fluid to the master cylinder. This ensures that there is no overall loss of fluid from the master cylinder during a skid control cycle. A reservoir may be connected between the modulator or control valve and the scavenge pump.

The invention will now be further described, by way of example only, with reference to the accompanying drawings, in which:-

Figure 1 is a schematic circuit diagram of an anti-skid brake control system incorporating a modulator assembly,

Figure 2 is a vertical cross-section of one of the modulator assemblies of the system of Figure 1,

Figure 3 is an enlarged view of the upper right-hand portion of Figure 2 showing the regulator piston in a different position,

Figure 4 is a schematic circuit diagram of another such system which incorporates a modulator assembly,

Figure 5 is a longitudinal section of a control valve assembly for use in an anti-skid brake control system according to the invention,

Figures 6 and 7 show modifications of the assembly of Figure 5,

Figure 8 is a circuit diagram of a system incorporating the assembly of Figure 5 and a power valve,

Figure 9 is similar to Figure 8 but incorporates a hydrostatic master cylinder,

Figure 10 is a vertical cross-section of another control valve assembly for use in a system according to the invention,

Figure 11 is a vertical cross-section of a further control valve assembly which incorporates a latch valve, and

Figure 12 is a vertical cross-section of a further control valve assembly in which the need for a latch valve has been eliminated.

With reference to Figure 1, a pump and unloader valve assembly 1 which constitutes a high pressure hydraulic fluid source, supplies hydraulic fluid from a reservoir 2 to a main supply line 3. The main supply line 3 feeds independent hydraulic accumulators 4 and 5 which are respectively connected to two braking circuits controlled by a pedal-operated dual control valve 6. A suitable valve is described in United Kingdom Patent Specification No. 1 509 187. The rear wheel 7 of the vehicle have wheel brakes 8 which, during normal braking, are fed with hydraulic fluid from the accumulator 4 via a supply line 9. The front wheels 10 have wheel brakes 11 which, during normal braking, are fed with hydraulic fluid from accumulator 5 via a supply line 12.

Three identical modulator assemblies 13, 14 and 15 are incorporated in the system for reducing the fluid pressure at the wheel brakes, 8, 11, in the event of a skid condition. Each of the front wheel brakes 11 has a separate modulator assembly 13, 14, whereas a single modulator assembly 15 controls both rear wheel brakes 8.

Each of the four wheels has a speed sensor, not shown, for sensing the speed of the associated wheel. The sensors are each connected to a respective electronic module, also not shown, which ascertains whether or not a skid conditon is imminent at the respective wheel and, if it is, sends an electrical control signal to the respective modulator assembly 13, 14 or 15. Suitable sensors and electronic modules are well known in the art and so will not be described here in detail.

Lines 16 and 17 respectively connect the front wheel brake modulator assemblies 13, 14, and the rear wheel brake modulator assembly 15 to the reservoir 2 to return fluid to the reservoir when the associated modulator assembly is actuated. However, as will become apparent from the following description, fluid is not returned from the wheel brakes themselves to the reservoir during a skid condition.

Referring now to Figure 2, each modulator assembly 13, 14, 15, comprises a housing 18 having two parallel stepped bores 19 and 20. Bore 19 comprises upper and lower bore portions 25 and 26 respectively, connected by an intermediate portion 27 of reduced diameter. A one-way ball valve assembly 28 is located at the junction of bore portions 25 and 27 and comprises a ball 35 which is urged against a seat 31 at the step in the bore, by a compression spring 32. A push-rod 34, which is not connected to the ball 35, is located in the intermediate bore portion 27.

An inlet port 41 in housing 18 is connected to the appropriate high pressure supply line 9 or 12, and communicates permanently with the upper bore portion 25 and with an aligned drilling 42 leading to the upper portion of bore 20.

A modulator piston 44 is located in the lower bore portion 26. This piston constitutes the movable wall referred to above. A first modulator chamber 43 is defined between the upper end of the modulator piston 44 and the upper end-wall of bore portion 26. This chamber 43 communicates permanently with a port 45 which is connected directly to the appropriate wheelbrake 8 or 11. A sealing ring 46 is located in an annular groove in the piston 44. A second modulator chamber 47 is defined between the lower end of the piston 44 and the lower end-wall of bore portion 26, and this chamber communicates permanently with the lower end of bore 20 by a drilling 48.

Piston 44 is urged upwards in bore portion 26 by means of a compression spring 50 which is located over a spigot 51 on the lower end of piston 44. The preloading of spring 50 is greater than that of spring 32 so that the ball 35 is normally held clear of seat 31 by rod 34 thus allowing communication between ports 41 and 45 during normal braking. In this position, the piston 44 abuts a shoulder 49 in bore portion 26.

Bore 20 comprises an enlarged portion 52 in which the solenoid 54 of a solenoid-controlled ball valve assembly 55 is located. A spigot 58, which is integral with the housing 18, extends downwardly into the bore of solenoid 54. This spigot has an axial through-bore 63, part of the bore 20, with a valve seat 64 at its lower end.

An armature 65 of circular cross-section is located with clearance in the bore 20 with its upper portion located within the bore of solenoid 54, and its lower portion projecting into a cavity 67 which communicates with enlarged bore portion 52 and, via drilling 48, with second modulator chamber 47. The armature 65 is resiliently biassed downwards by a compression spring 69 located over the spigot 58. Armature 65 has at opposite ends, axial recesses 70 in which are located respective balls 71 of the ball valve assembly 55. The upper ball 71 is positioned to seat against the valve seat 64, and the lower ball against a valve seat 72 formed on a boss 74 disposed centrally of cavity 67. An outlet port 73 opens into the centre of valve seat 72.

When solenoid 54 is not energised, the armature 65 holds the lower ball 71 against seat 72 under the action of spring 69, to close outlet port 73. Upper ball 71 is free to move away from seat 64 to permit free communication between bore 63 and second modulator chamber 47 via cavity 67 and drilling 48.

Bore 20 also comprises an upper portion 78 in which is located a regulator assembly 80 which will now be described with reference to Figure 3.

The bore portion 78 is lined by a sleeve 81 which defines a cylinder bore 82 in which a regulator piston 83 is slidable. This piston 83 has an axial bore 87 comprising upper and lower portions 88 and 89 respectively, communicating permanently with each other via an orifice 90. This orifice constitutes the first restriction of predetermined size referred to above. A spigot 91 is formed at the lower end of the piston 83. A compression spring 92 is located over the spigot 91 with its upper end abutting a step 93 formed by the junction of spigot 91 with the body of the piston and its lower end abutting a step 94 formed by the junction of bore portions 78 and 63. Thus, as shown in Figure 2, the piston 83 is urged upwards by the spring 92 to abut upper end wall 84. In Figure 3, the piston 83 is shown displaced downwards from wall 84 under pressure of fluid, as will be explained below. Piston 83 has at its mid-length an external annular recess 95 which communicates permanently with bore portion 88 by a radial passage 96.

The sleeve 81 has an external annular recess 97 which is arranged to communicate permanently with the drilling 42. The sleeve 81 also has a series of circumferentially spaced radial ports 98, the outer ends of which open into the recess 97, and the inner ends of which communicate in a restricted manner with the annular recess 95. This restriction is caused by the outer surface 99 of the regulator piston 83, immediately adjacent to the recess 95. The degree of restriction depends upon the position of the piston 83 in the bore 82 relative to the ports 98. This restriction constitutes the second restriction referred to above.

Pressure spaces 101 and 102 respectively are defined between the upper end of regulator piston 83 and the upper end wall 84 of bore portion 78, and between the lower end of regulator piston 83 and the lower end wall 94 of bore portion 78. The areas of the upper and lower ends of regulator piston 83 exposed to pressure spaces 101 and 102 respectively, are equal.

The operation of the modulator assembly of Figure 2 and 3 will now be described.

During normal braking, port 45, which is connected to the appropriate wheelbrake 8 or 11, communicates freely with inlet port 41 through bore portion 27 and valve assembly 28, the ball 35 being held off seat 31 by push-rod 34 since piston 44 is urged upwards by spring 50. Second modulator chamber 47 contains hydraulic fluid at the pressure which exists at inlet port 41, this port communicating with chamber 47 via drilling 42, ports 98, recess 95, passage 96, orifice 90, bore 63, cavity 67 and drilling 48. Regulator piston 83 is held against wall 84 by spring 92 to provide unrestricted flow between the radially inner ends of ports 98 and recess 95. During normal braking, solenoid 54 is not energised, so that lower valve member 71 is held against seat 72 to isolate outlet port 73.

If an electronic module detects an imminent skid, it sends a skid signal to the solenoid 54 of the associated modulator assembly to energise the solenoid.

Energisation of solenoid 54 causes an upward movement of armature 65 to close valve seat 64 and open valve seat 72 and thereby connect second chamber 47 to outlet port 73. Since the pressure of fluid in the reservoir 2 is less than that in the line connected to the port 41, the pressure of fluid in the first chamber 43 results in downward movement of piston 44 and closure of valve seat 31 to isolate outlet port 45 from inlet port 41. Further movement of piston 44 results in expansion of the first chamber 43 and relieves the brake pressure at the wheel brake connected to port 45. Note that the regulator valve 80 plays no part in the relief of brake pressure and thus the first and second restrictions of regulator valve 80 are not operative during this period.

When the associated electronic module detects that the tendency to skid has been overcome, the solenoid 54 is deactivated.

Armature 65 then moves downwards to close valve seat 72 and open seat 64, so that the second chamber 47 begins to receive fluid from the inlet port 41 via regulator valve 80, the function of regulator valve 80 being to control the rate at which second chamber 47 is re-filled and thereby re-pressurised on cessation of the skid signal. When valve seat 64 is opened, second chamber 47 is initially at a reduced pressure so that the pressure in pressure space 102 of regulator valve 80 initially drops substantially to the reduced pressure. Regulator piston 83 then moves downwards due to the fluid pressure in pressure space 101, so that the radially inner ends of the ports 98 become restricted by piston wall 99.

Regulator piston 83 then adopts an equilibrium position in which the downward force on the piston due to the pressure difference between pressure spaces 101 and 102 is equal to the upward force of spring 92. Since the rate of flow through orifice 90 is determined by the pressure difference between pressure spaces 101 and 102, a suitable choice of the characteristics of spring 92 will provide a predetermined rate of flow through orifice 90 to second chamber 47.

In the equilibrium position of piston 83, the total pressure difference acting across the regulator valve, between drilling 42 and bore portion 63, is divided between the pressure difference acting across the orifice 90 and that acting across the second restriction at the inner ends of port 98. Suitable choice of the spring 92 allows a substantial part of the total pressure drop to act across the second restriction to enable the diameter of orifice 90 to be made of substantial size.

Since the second restriction comprises a number of ports 98 it is very unlikely that this restriction could become totally blocked by foreign matter. If any blockage did occur it would automatically be relieved by upward movement of the regulator piston.

Thus second chamber 47 is re-filled at a predetermined rate controlled by a regulator valve 80 so that the first chamber 43 and the associated wheel brake are re-pressurised at a corresponding rate. If the re-pressurisation of the wheel brake does not result in a further skid signal, piston 44 moves to its normal position to re-open valve seat 31. If a new skid condition occurs when the wheel gains speed, second chamber 47 is again connected to exhaust port 73 for as long as necessary until the control module signals that a skid is no longer imminent.

Note that no hydraulic fluid is dumped from first chamber 43 during the sequence of operations initiated by a skid signal.

In Figure 4, parts corresponding to those of the system of Figure 1 have been given corresponding reference numerals. In this case a hydrostatic tandem master cylinder assembly 103 is used to generate pressure in the wheel brakes 8 and 11 and to provide the energy required to operate modulator piston 44 of the modulator assemblies 13, 14 and 15. The primary pressure space of the master cylinder assembly 103 is connected by a line 104 and line 9 to the inlet port 41 of modulator assembly 15 associated with the wheel brakes 8 of the rear wheels 7, and the secondary pressure space is connected by a line 105 and lines 12 to the inlet ports 41 of the modulator assemblies 13 and 14 associated with the front wheel brakes 11. The outlet ports 73 of the modulator assemblies 13 and 14 are not connected to the reservoir in this case, but instead are connected by lines 106a, 106b and 107 to a first pump chamber 108 of a two-cylinder scavenge pump 109 of which the opposed pistons 110, 111 are driven by a common cam 112. A one-way valve 133 is connected in line 107 to prevent flow of fluid from pump chamber 108 to lines 106a and b, and a further one-way valve 114 is connected in an outlet line 115 from pump chamber 108 to brake supply lines 12. The action of piston 110 is thus to return at brake-line pressure fluid expelled from outlet ports 73 of modulators 13 and 14 to the brake line 105 and thereby to the secondary pressure space of master culinder assembly 103. A reservoir 116 is connected to line 107 on the ineot side of valve 113, to enable the use of a smaller pump 109 than would otherwise be necessary.

Modulator assembly 15 associated with both rear wheel-brakes 8, similarly has exhaust port 73 connected by way of valves 113', 114', reservoir 116' and second pump chamber 108' to the brake supply line 104 connected to the primary pressure space of the master cylinder, so that hydraulic fluid exhausted from the second chamber 47 of the modulator assembly 15 is returned at brake line pressure to the primary brake circuit.

The scavenge pump 109 is thus arranged to return to the master cylinder pressure spaces all the fluid exhausted from the modulators 13, 14 and 15 during actuation of the modulators on the occurence of a skid signal. This fluid is then available for repressurising the second chambers 47 of the modulators by way of regulator valves 80 on cessation of the skid signals.

The modulator assembly described thus enables a common pressure source to be used for actuating the wheel brakes and for controlling movement of the modulator piston. Previously, it has usually been necessary to provide separate fluid pressure sources operating at different pressures to perform these two functions.

As an alternative to the arrangement described above with reference to Figure 3, the radial ports 98 may lead at their inner ends into an annular recess formed in the sleeve 81, and the wall of the piston 83 may have a number of circumferentially spaced radial ports in lieu of recess 95 and passage 96. In such a case the inner surface of the sleeve 81 adjacent to the annular recess serves to restrict flow of fluid through the ports of the piston, depending upon the position of the piston in the bore 82.

The modulator assemblies described above with reference to Figures 1-4 are intended for use in systems where fluid is not dumped from the brake circuit in response to a skid signal, but is temporarily diverted to a chamber until the skid conditions have been overcome whereupon the same fluid is returned to the brakes at a controlled rate. The remaining Figures are all concerned with systems incorporating a control valve which, in response to a skid signal, isolates the brakes from their fluid supply, and then connects the brakes to a reservoir of some kind. On cessation of the skid signal the brakes are re-applied at a controlled rate from their fluid supply, but not necessarily with the same fluid which was removed from the brakes in response to the skid signal.

Referring to Figure 5, the control valve assembly comprises a housing 121 containing a solenoid 122 and a co-axial armature/spool 123 slidable in a stepped bore 124 having bore portions 125 and 126 of larger and smaller diameter respectively. Spool 123 is biassed to the left as shown, by a coiled compression spring 127 which bears at one end against the step between bore portions 125 and 126 and at the other end against an outwardly directed flange 128 on the spool 123. A through-passage 129, provides permanent fluid communication between chambers 130 and 131 at opposite ends of spool 123 for pressure-balancing purposes. An inlet 132 for connection to the outlet of a master cylinder, communicates with a chamber 133 in which is housed the ball 134 of a ball valve which controls communication of inlet 132 with an axial passage 135 connected to chamber 130 from which a first outlet port 136 leads for connection to a wheel brake.

A second outlet port 137 for connection to a reservoir leads from bore portion 126 and is spaced axially from one end 138 of a by-pass which contains a regulator 144, and the other end 140 of this by-pass connects with chamber 133. An annular recess 141 in the spool 123 connects passage 129 with one end 138 of the by-pass. Ball 134 is normally held off its seat against the action of a closure spring 142 by a push rod 143 carried by spool 123. In the unactuated condition of the valve shown in Figure 5, the second outlet port 137 is closed by the spool 123.

Regulator 144 comprises a regulator piston 145 which is slidable against the action of a compression spring 146 which urges piston 145 to the left as shown. Piston 145 has a longitudinal passage 147 which communicates with an annular recess 148 by way of a radial orifice 149. A chamber 150 at the left-hand end of the piston 145 is pressure-balanced with the pressure in recess 148 by way of axial passages 151. A chamber 139 is formed at the right hand end of the piston.

On the occurence of a skid condition, solenoid 122 is energised by a skid signal produced by a skid detection means, not shown, and spool 123 moves to the right allowing ball 134 to close against its seat to isolate the inlet 132 from first outlet 136, and then on further movement of spool 123 to the right, to connect the first and second outlets 136 and 137 by way of passage 129 and annular recess 141 which is moved into register with port 137. The connection of ports 136 and 137 results in fluid being dumped from the brakes to reservoir causing the wheel to recover from the skid condition with the result that solenoid 122 is de-energised. On the initial leftward return movement of spool 123, second outlet port 137 is closed, and on further movement recess 141 comes into register with end 138 of the by-pass to place the inlet 132 in communication with the brakes via regulator 144.

The regulator 144 produces a predetermined fluid flow rate to the brakes substantially independent of the inlet pressure. Orifice 149 constitutes a first restriction, and a second restriction of variable size is defined between the left-hand end of recess 148 and the wall of passage 140. Any reduction the the pressure in chamber 139 tends to bring about an increase in the pressure difference across orifice 149, but that increase causes piston 145 to move to the right to reduce the second restriction, so that the second restriction supports the pressure difference, and the flow through orifice 149 remains substantially the same.

The brakes are thus re-pressurised in a controlled manner, and the spool is held in this position by the ball 134 which is held in the closed condition by the pressure differential between chambers 133 and 130, providing that the inlet pressure has not been reduced in the meantime.

If, due to re-application of the brakes, a further skid condition is produced, the solenoid 122 is re-energised and spool 123 moves to the right to commence a new cycle of brake release/controlled re-application by regulator 144.

Should the re-application process continue until the inlet and brake pressures equalise, the spring 127 causes re-opening of the ball valve.

Figure 6 shows a modification of the construction of Figure 5, in which corresponding parts have been given corresponding reference numerals. In this embodiment the ball valve controls both the inlet and exhaust functions since ball 134 is adapted to close both coaxial valve seats 155 and 156 formed on the housing 121 and on the end of a tubular projection 157 on the left-hand end of the spool 123, respectively. The bore 158 of projection 157 communicates permanently with recess 141. Chamber 130 communicates with chamber 131 via by-pass portion 138, chamber 139, and a bore 159 in housing 121, for pressure-balancing purposes. The main advantage of this arrangement is that the working stroke of the spool 123 is substantially reduced compared with the embodiment of Figure 5. This reduced working stroke allows a faster response to be achieved, and a higher force can be applied to the spool by the solenoid across chamber 131 which constitutes the air gap of the solenoid. It will be noticed, however, that in this arrangement the by-pass connection containing regulator 144 is not cut-off on energisation of the solenoid 122 but provides a restricted connection between the inlet 132 and first outlet 136. This permanent connection is acceptable in most applications.

Figure 7 shows a further modification in which a yet smaller movement of the armature 123 is facilitated by the use of a lever arm 165 pivotally connected at 160 to armature 123 and at 161 and 162 respectively to independent inlet and exhaust valve members 163 and 164. The pivotal connections 161 and 162 are not equidistant from connection 160 so as to provide a higher velocity ratio for the inlet valve 163, which is desirable because valve 163 is always operated in the near balanced state. For reasons of clarity the by-pass and regulator between inlet chamber 133 and outlet 136 is not shown in Figure 7. Passages 166 and 167 are present for pressure-balancing purposes.

In the circuit of Figure 8, a pedal-operated dual power valve master cylinder assembly 174 has separate hydraulic accumulators 175 and 176. The accumulators 175 and 176 are supplied under pressure with fluid by a pump, not shown, which draws the fluid from an hydraulic reservoir 177. Those skilled in the art will appreciate that each power valve of the assembly 174 comprises an inlet and an exhaust valve which control the metering of hydraulic fluid to the supply lines 178 and 179 to apply the brakes, and the return of fluid from the lines 178 and 179 to lines 180 and 181 respectively leading to reservoir 177. Supply line 178 leads to both the wheel brakes 182 on both front wheels 183 by way of dual control valve assembly 184 which comprises two independent control valve assemblies 185 and 186 of the type shown in Figure 5 but sharing a common housing. The solenoids of valves 185 and 186 are connected by respective leads 187 and 188 to respective electronic skid-detection units 189 and 190 which receive wheel speed signals from wheel speed sensors 191 and 192 on the respective front wheels 183 by way of leads 193 and 194 respectively. The two outlet ports 137 of the valve assembly 184 are connected to a common line 195 leading to the right hand side as shown of the reservoir 177 which has a central partition 196.

Supply line 179 leads to the inlet 132 of a further control valve assembly 197 of the type shown in Figure 5, of which the first outlet 136 leads by way of lines 198 to wheel brakes 199 on both rear wheels 200, and the second outlet 137 leads to the left-hand part of reservoir 177 by way of line 201. The solenoid 122 of control valve assembly 197 is connected by a lead 202 to elctronic skid detection unit 203 which is connected by way of lead 206 to a speed sensor 204 on the rear wheel transmission 205.

On application of the brake pedal, fluid is fed under pressure by power valve 174 to the brakes 182 and 199 by way of the open, unrestricted passages controlled by the ball valves 134 but on detection of a skid condition by one of the units 189, 190, 203, the solenoid of the associated control valve assembly 185, 186, 197 is energised to dump fluid from the appropriate brake to reservoir 177.

This results in acceleration of the particular front wheel or, in the case of rear-wheel skid, of both rear wheels, and the solenoid is eventually de-energised. The brake pressure is then increased at a controlled rate, as explained earlier, and further cycles of brake pressure reduction/regulated brake re-application will take place until the skid conditions are no longer present.

Figure 9 shows a circuit similar to that of Figure 8 but employing a boosted, hydrostatic, tandem master cylinder assembly 207. Corresponding reference numerals have been applied to parts corresponding to those of Figure 8. As is conventional, tandem master cylinder 208 of assembly 207 has its own reservoir 209, and in this system fluid is not dumped to reservoir 209 in the event of a skid signal, but is dumped instead to respective accumulators 210 and 211. Each accumulator houses a piston 212 which is biassed towards an inlet 213 by a compression spring 214.

The control valve assemblies 185, 186 and 197 are again of the kind shown in Figure 5.

Accumulator 210 is connected by a line 215 to the outlet 137 of control valve assembly 197 associated with both rear wheels 200, and accumulator 211 is connected by a line 216 to the outlets 137 of both assemblies 185 and 186 associated with the front wheels 183.

A pump assembly 217 is provided for pumping dumped fluid in the accumulator 210 or 211 back into the line 179 or 178 respectively leading to the respective master cylinder pressure space. Pump assembly 217 comprises an engine or transmission driven cam 218 and pump pistons 219, 220 working in diametrically opposed bores 221, 222 to define pump chambers 223 and 224 respectively. Pump chamber 223 is connected by way of an inlet valve 225 to the line 215 connected to accumulator 210 and by way of an outlet valve 226 to line 179. Pump chamber 224 is similarly connected by way of outlet valve 227 to line 216, and by way of outlet valve 228 to supply line 178.

When the master cylinder 208 is actuated, hydraulic fluid is fed under pressure to the front and rear brakes by way of lines 178 and 179 respectively, and by way of the ball valve of the respective control valve assembly 185, 186, 197. The pump pistons 219 and 220 are normally clear of the cam 219. On detection of a skid condition by the rear wheel skid detection unit 203, for example, the solenoid 122 of the control valve assembly 197 is energised to close ball valve 134 and dump fluid from the lines 198 to the accumulator 210 to move piston 212 against the force of spring 214, and also to pressurise chamber 223 and force pump piston 219 against the cam 218. Pump piston 219 is then reciprocated by the cam 218 to pump fluid from accumulator 210 into line 179 and thereby into the master cylinder pressure space, not shown, connected to line 179. The reduction in rear wheel brake pressure causes the rear wheels 200 to accelerate. The skid signal therefore ceases, and the solenoid 122 of assembly 197 is de-energised so that the spool 123 moves until push rod 143 abuts the ball 134 of the ball valve which remains closed. Fluid now flows in a regulated manner from the master cylinder and/or chamber 223 via lines 179, inlet port 132 and regulator 144, outlet 136 and lines 198 to the wheel brakes 199. The pumping rate is arranged such that it normally exceeds the maximum regulated re-supply rate of fluid to the brakes. When all the fluid in accumulator 210 has been returned to line 179 the pump piston 219 remains clear of the cam 218. Note that an electric failure will not prevent the brakes from being re-applied.

Figure 10 shows a control valve assembly in which the regulator is arranged in parallel with the solenoid-controlled valve between parts 132 and 136. Parts which correspond to those of Figure 5 have been given corresponding reference numerals. The housing 121 comprises a main part 231 and an end cap 232 enclosing a bore 233 in which solenoid 122 is located. A valve seat 234 surrounds outlet port 137. The solenoid is held in position in bore 233 by an annular plug 236 which slidably guides armature 123 with clearance, and provides an abutment for armature bias spring 127. Balls 237 and 238 are carried one at each end of armature 123 for co-operation respectively with seat 234 and ball valve seat 239 formed on the lower end of a tubular plug 240 located within the solenoid. A passage 245 connects the space surrounding armature 123 with outlet 136.

In the un-energised position of the armature, shown in the drawing, valve seat 234 is closed to cut off communication with reservoir, and valve seat 239 is open to allow unrestricated communication between inlet 132 and outlet 136 provided the regulator piston 145 is in a certain position as described below.

Regulator 144 comprises a sleeve 248 which is sealed in a bore 243 in housing part 231 by a plug 249 and has a pair of external annular recesses 251a, 261a from which lead a series of circumferentially spaced radial ports 251b, 261b respectively. Recess 251a communicates freely with inlet port 132 via passage 264, and recess 261a communicates freely with bore 244 of tubular plug 240 via passage 262. Regulator piston 145 has a through-bore 147 having a restriction 149 of predetermined size. Bore 147 communicates with an external annular recess 148a in piston 145 by radial ports 148b, and a variable second restriction is defined between recess 148a and ports 251b. A spring 146 urges the piston against plug 249 so that ports 251b are unrestricted during normal braking. Chamber 139 of the regulator is connected to outlet port 136 by passage 247.

Bore 147 also communicates with a second external annular recess 258a in piston 145 by radial ports 258b. Recess 258a is normally in register with ports 261b such that there is free communication between ports 132 and 136 by way of passage 264, recess 251a and ports 251b, recess 148a and ports 148b, bore 147, ports 258b and recess 258a, ports 261b and recess 261a, passage 262, the solenoid controlled valve, and passage 245. A small flow of fluid is also possible through orifice 149, but this is not significant during normal braking.

When the solenoid 122 is energised by a skid signal, valve seat 234 is opened to connect together ports 136 and 137 to dump brake fluid to reservoir, and valve seat 239 is closed. A pressure differential therefore exists between ports 132 and 136 which causes piston 145 to move downwards.

This cuts off all communication between recess 258a and ports 261b and, on further downward movement, communication is restricted between ports 251b and recess 148a. When the solenoid is de-energised, piston 145 controls the rate of re-application of the brakes in the manner described above. When the pressure differential between ports 132 and 136 becomes sufficiently small, piston 145 moves upwards to restore communication between port 132 and the solenoid-controlled valve.

Figure 11 shown a control valve assembly which incorporates a pressure-responsive latch valve 230 connected in parallel with regulator 144 which is connected between inlet 132 and the solenoid-controlled valve. Parts generally corresponding to those of Figure 10 have been given corresponding reference numerals. A tubular connector 235 is shown attached to outlet port 137 for connection to reservoir. Bores 243, which contains regulator 144, and 233, which contains the solenoid controlled valve, are axially aligned, and chamber 139 of the regulator communicates with passage 244 of plug 240. Latch valve 230 is contained in a parallel bore 242.

The latch valve 230 comprises a piston 254 which is slidable in a fixed sleeve 255 and is urged upwards by a compression spring 257 to abut a plug 256 which seals bore 242. Upper chamber 246a, which is defined between the piston 254 and plug 256, communicates with inlet 132 and via a passage 262 with recess 251a of regulator 144, and is connected via bore 247 to outlet 136. Piston 254 has an external annular recess 258a which communicates with upper chamber 246a via radial passage 258b and axial bore 259 all formed in the piston. During normal braking, recess 258a is in register with radial ports 261b and recess 261a, both formed in sleeve 255, and recess 261a communicates with chamber 139 of regulator 144 via passage 241. Thus, ports 132 and 136 are normally in un-restricted communication via chamber 246a, bore 259, passages 258b, port 261b, passage 241, bore 244, the solenoid-controlled valve, and passage 245. Thus, regulator 144 is effectively by-passed by latch valve 230 during normal braking.

When a pressure differential is produced between ports 132 and 136 during energisation of solenoid 122, the corresponding pressure differential between chambers 246a and 246b causes piston 254 to move downwards and so cut off communication between recess 258a and ports 261b, and the wheel brake is isolated from inlet 132. When the solenoid 122 is de-energised on cessation of the skid signal, ball valve 237 closes and ball valve seat 239 opens, and the brake pressure is increased in a controlled manner by regulator 144, the latch valve 230 remaining closed until the pressure differential between chambers 246a and 246b is sufficiently reduced.

If the driver releases the brake pedal before the brake pressure is restored, the pressure in inlet port 132 falls and the piston 254 of latch valve 230 is returned to its normal position by brake pressure and return spring 257. Pressure in the brakes is rapidly relieved as fluid flows back through seat 239 and latch valve 230.

In the construction of Figure 12, the regulator is again connected between inlet 132 and the solenoid-controlled valve, but the regulator piston 145 has been modified to fulfil the function of the latch valve of Figure 11. Parts which generally correspond to those of

Figures 10 and 11 have been given corresponding reference numerals.

Regulator piston 145 has an axial bore 147 which communicates via restrictor orifice 149 with a radial passage 265 leading into an annular recess 266 in the outside of the piston. A first set of radial ports 148b lead from bore 147 into an external annular recess 148a and a second set of radial parts 258b, which are axially spaced from parts 148b, lead from bore 147 into a further annular recess 258a.

Piston 145 is slidably received within a sleeve 248 having four axially spaced sets of radial ports 251b, 261b, 267 and 269b. Ports 251b open into external annular recess 251a, ports 261b and 267 open into a common external annular recess 268, and ports 269b open into a further annular recess 269a. Recess 251a communicates directly with inlet 132, recess 268 communicates with valve seat 239 via passage 244, and recess 269a communicates directly with outlet 136. The piston is urged upwardly by a compression spring 146.

During normal braking, recess 148a is in unrestricted communication with ports 251b, recess 258a is in un-restricted communication with ports 261b, and recess 266 is sealed by the inside wall of sleeve 248. Chamber 270 formed at the lower end of piston 145, as shown, communicates with the space surrounding armature 123 via a passage 271 and with outlet 136 via ports 269b and recess 269a. Thus, ports 132 and 136 are normally in un-restricted communication via ports 251b, ports 148b, bore 147, ports 258b, ports 261b, recess 268, passage 244, passage 271, chamber 270 and ports 269b.

When the solenoid 122 is energised by a skid signal, a pressure differential is produced between bore 147 and chamber 270 so that piston 145 is urged downwards. This causes ports 261b to be closed and ports 267 to open into recess 266. When the skid signal ceases, a pressure differential is still present across piston 145 so that communication through ports 261b remains cut off, but the brakes are re-supplied at a controlled rate by fluid flowing from port 132, through the variable restriction defined between ports 251b and recess 148a through orifice 149 and recess 266, via ports 267 and recess 268 to the solenoid controlled valve and so to port 136. When the brake pressure builds up, the pressure differential across piston 145 is reduced so that the piston moves upwards again to provide un-restricted communication between ports 132 and 136.

Note that in the normal position of piston 145, the regulator is, in effect, by-passed through passages 261b.

CLAIMS

1. An anti-skid brake control system comprising a wheel having a hydraulic brake applicable by supplying thereto fluid under pressure, means to sense the presence of skid conditions at the wheel during braking and thereupon to produce a skid signal, means to isolate the brake from its fluid supply and displace fluid from the brake in response to a skid signal so as to relieve the brake pressure, and a restriction arranged to control flow of fluid to the brake on cessation of the skid signal so as to re-apply the brake pressure at a controlled rate, characterised in that the restriction (90, 149) is connected in series with a second restriction defined between two relatively movable members (83, 145;and 81, 248), relative movement of the members being controlled by the difference in fluid pressure across the first restriction such that an increase in this pressure difference causes a reduction in the size of the second restriction.

2. A system according to Claim 1, characterised in that the first restriction (90, 149) is connected downsteam of the second restriction.

3. A system according to Claim 2, characterised in that the first restriction (90, 149) is defined by one

of the relatively movable members (83, 145) and that member is resiliently biassed against the action of the pressure difference across the first restriction.

4. A system according to Claim 3, characterised in that the relatively movable member defining the first restriction is a piston (83, 145) which is movable within a bore, and the bore wall is the other relatively movable member.

5. A system according to Claim 4, characterised in that the piston (83, 145) is cup-shaped, the first restriction is defined by the end wall of the piston and the second restriction is defined between at least one radial passageway (98, 251b) in the wall of the bore and an annular recess (95, 148a) in the outside of the piston, the recess being in permanent communication with the inner space (88, 147) of the piston.

6. A system according to Claim 4, characterised in that the piston (83, 145) is cup-shaped, the first restriction is defined by the end wall of the piston and second restriction is defined between at least one radial passageway in the wall of the piston and an annular recess in the wall of the bore, the annular recess being connected to a source of hydraulic fluid.

7. A system according to any preceeding claim, characterised by a movable wall (44) separating a pair of chambers (43, 47), one of the chambers (43) being connected to the hydraulic brake and the movable wall being arranged to increase the volume of that chamber (43) in response to the skid signal so as to relieve the brake pressure at the braked wheel, the regulator (80) formed by the first and second restrictions being arranged to regulate flow of fluid to the second of the cham-

bers (47) on cessation of the skid signal such that the movable wall (44) is thereby moved progressively to reduce the volume of the first chamber (43) so as to re-apply the brake pressure at a controlled rate.

8. A system according to Claim 7, characterised in that a three-way valve (55) is interposed between the second chamber (47) and the regulator (80), the valve being responsive to the skid signal such that the second chamber (47) normally is connected to the regulator (80) but on production of the skid signal the valve is switched to connect the second chamber (47) to an outlet (73) for fluid.

9. A system according to Claim 8, characterised in that a two-way valve (28) is connected in the fluid supply line to the hydraulic brake, the valve being arranged to close when the movable wall (44) moves to increase the volme of the chamber (43) connected to the hydraulic brake.

10. A system according to Claim 9, characterised in that the movable wall (44) and the first and second chambers (43, 47), the regulator (80), the three-way valve (55) and the two-way valve (28) are all contained within a single housing (18).

11. A system according to Claim 7, characterised in that a common source of hydraulic fluid supplies the regulator and the hydraulic brake.

12. A system according to any of Claims 1 to 6, characterised by an electromagnetically actuated, valve arrangement connected to the fluid supply for applying the brakes, the hydraulic brake and a reservoir, and being responsive to the skid signal such that the valve

arrangement normally connects the brake to the fluid supply but on production of the skid signal the valve arrangement is switched to connect the brake to the reservoir, the regulator (144) formed by the first and second restrictions being arranged to regulate the flow of fluid from the fluid supply to the brakes on cessation of the skid signal.

13. A system according to Claim 12, characterised in that the regulator (44) is connected in parallel with the valve arrangement between the fluid supply and the hydraulic brake, and the valve arrangement is arranged not to connect the fluid supply to the brake on cessation of the skid signal until the brake pressure has been re-applied by fluid supplied via the regulator (144), the valve arrangement being adapted to cut off flow of fluid through the regulator (144) during production of the skid signal.

14. A system according to Claim 12, characterised in that the regulator (144) is connected in parallel with the valve arrangement between the fluid supply and the hydraulic brake, and the relatively movable members (145, 248) are adapted to cut off communication between the fluid supply and the valve arrangement until the brake pressure has been re-applied by fluid supplied via the regulator (144).

15. A system according to Claim 12, characterised in that the regulator (144) is connected between the fluid supply and the valve arrangement and a normally open further valve (230) is connected in parallel with the regulator, the further valve being arranged to close on production of the skid signal and to stay closed until the brake pressure is re-applying by fluid supplied via the regulator (144).

16. A system according to Claim 15, <u>characterised in that</u> the further valve (230) is a latch valve comprising a piston (254) responsive to the pressure differential between the fluid supply and the brake such that an increase in this pressure differential above a predetermined valve results in closure of the latch valve.

17. A system according to Claim 12, <u>characterised in that</u> the regulator (144) is connected between the fluid supply and the valve arrangment and the first restriction is defined by one of the relatively movable member (145), that member (145) being responsive to the pressure differential between the fluid supply and the brake such that an increase in this pressure differential above a predetermined value causes relative movement of the members by an amount which is sufficient to close a by-pass (261b) connected across the regulator (144), the by-pass (261b) connected across the regulator (144), the by-pass (261b) being arranged to stay closed until the brake pressure is re-applied by fluid supplied via the regulator (144).

18. A system according to Claim 12, <u>characterised in that</u> at least the valve arrangement and the regulator (144) are included in a single control valve assembly comprising a housing (121; 231, 232) having an inlet (132) for connection to the fluid supply, a first outlet (136) for connection to the brake and a second outlet (137) for connection to the reservoir.

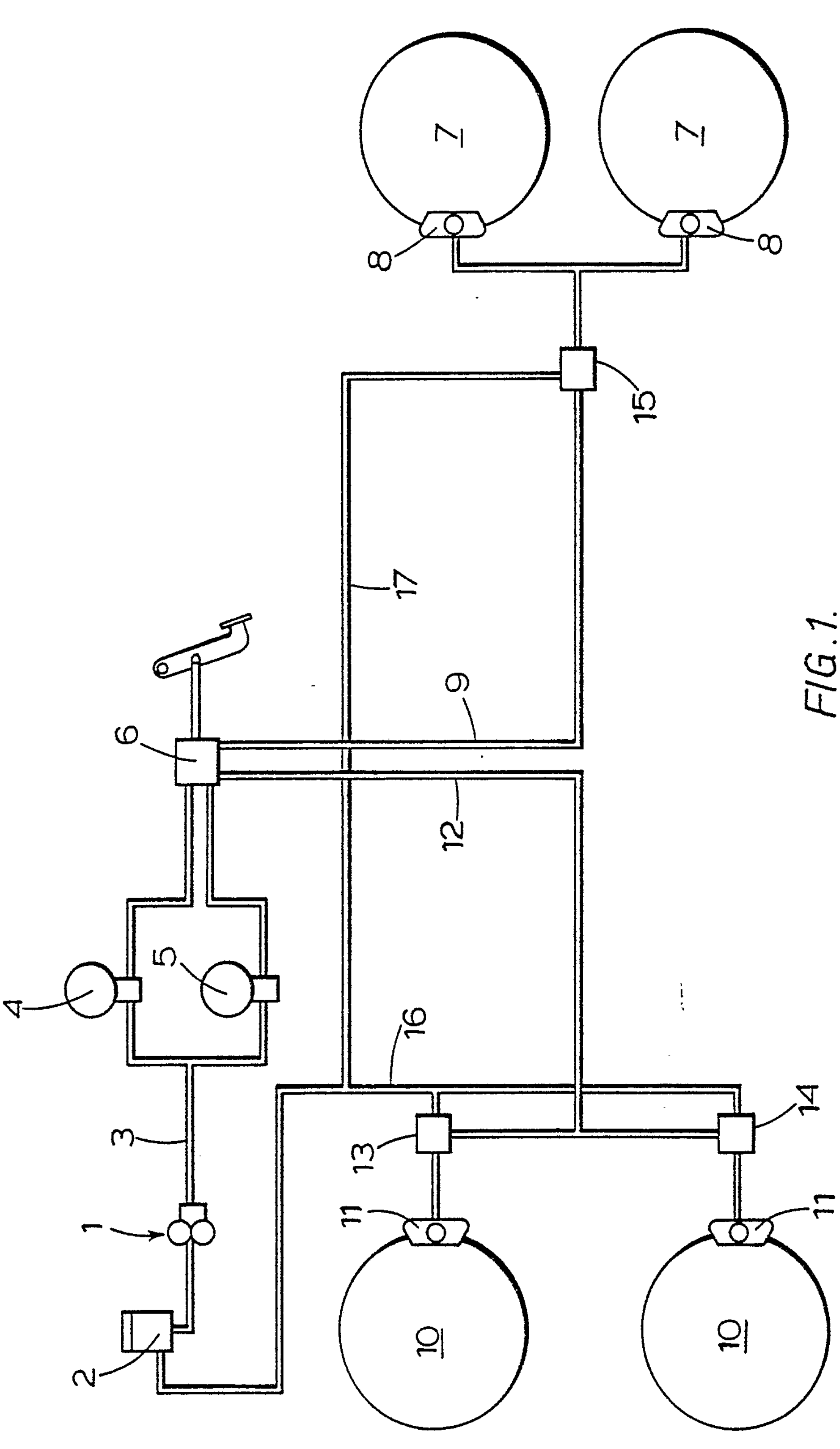

FIG.1.

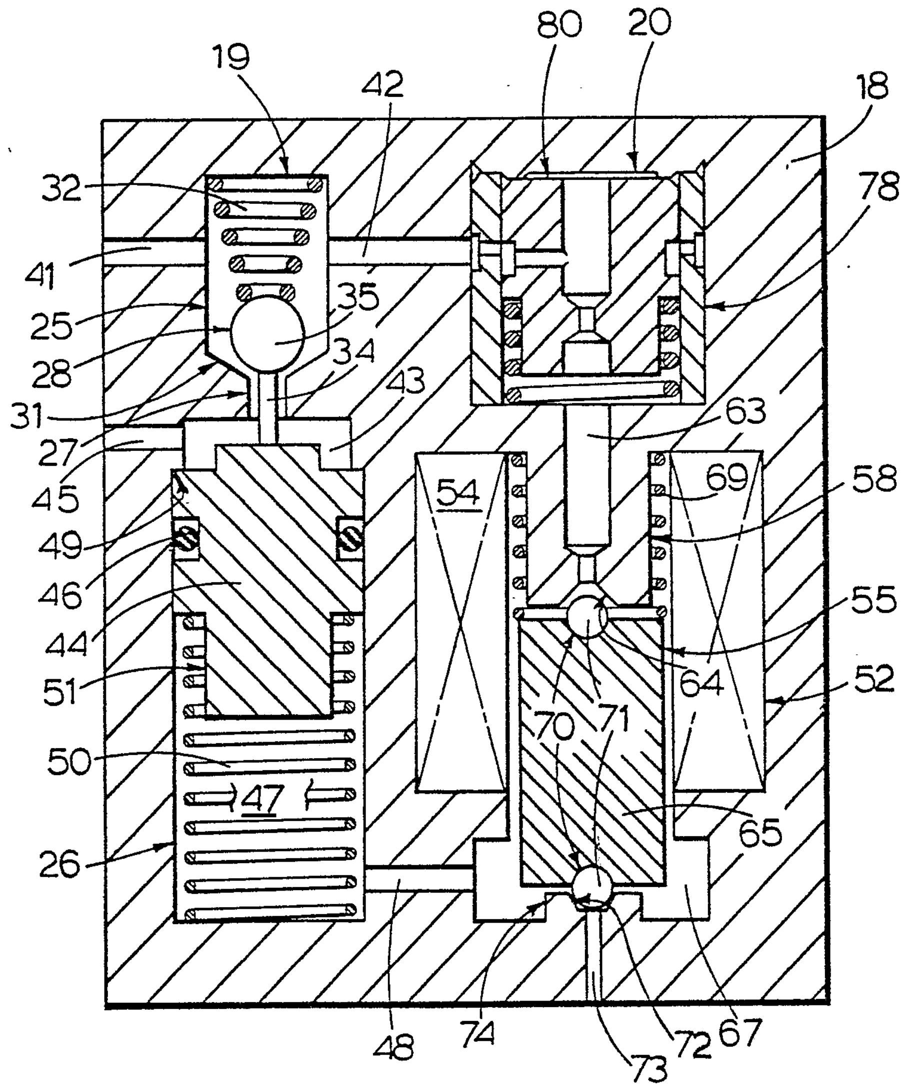
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FIG.2.

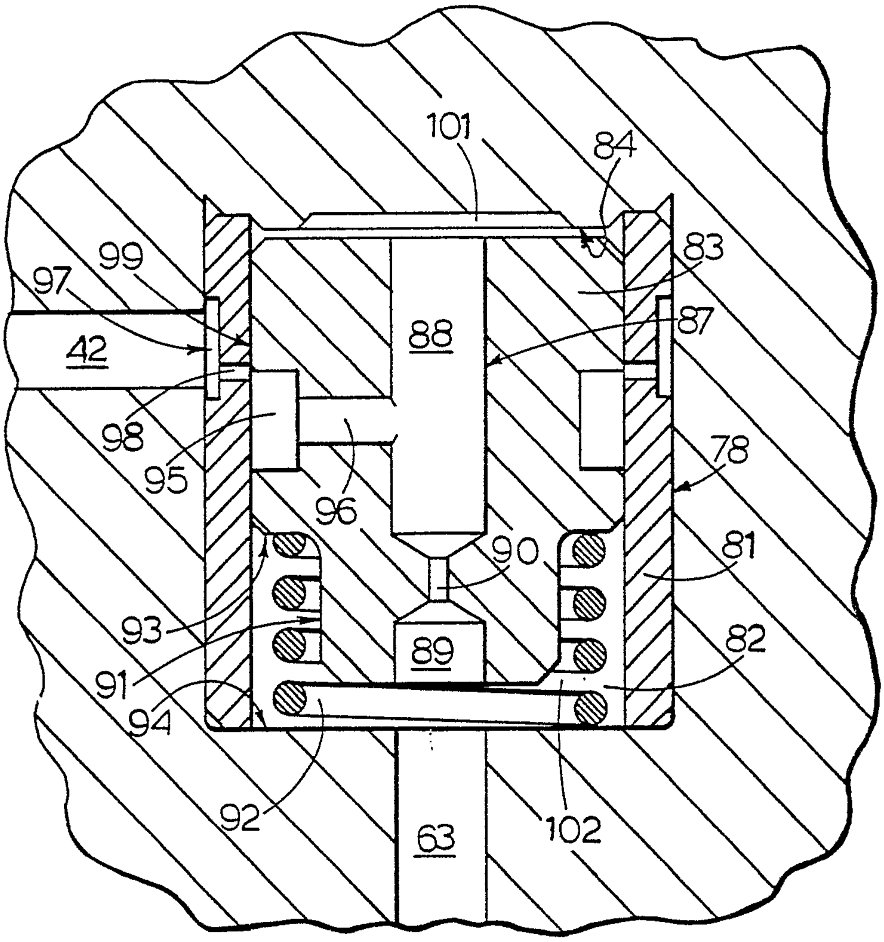

FIG.3.

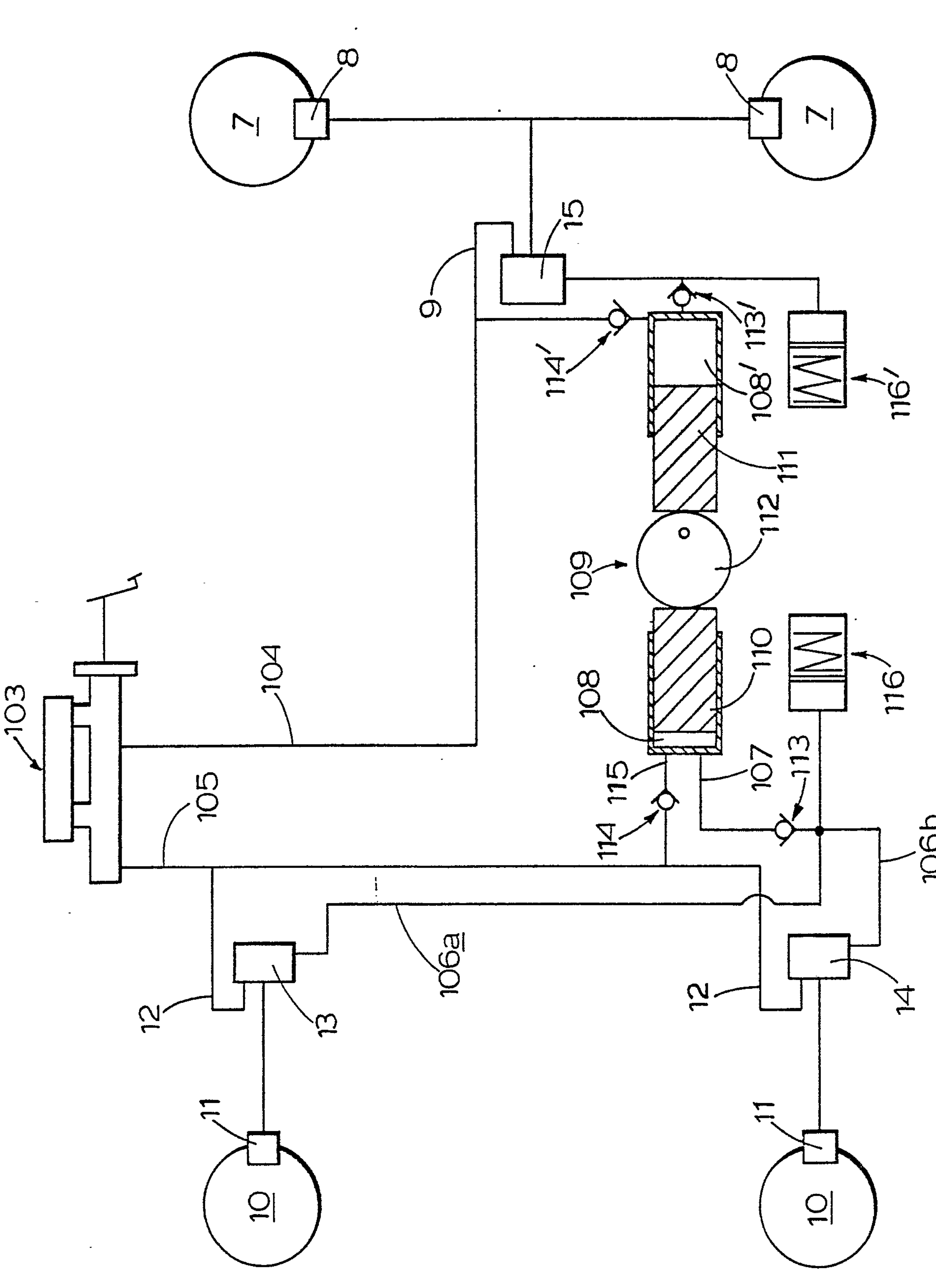
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FIG. 4.

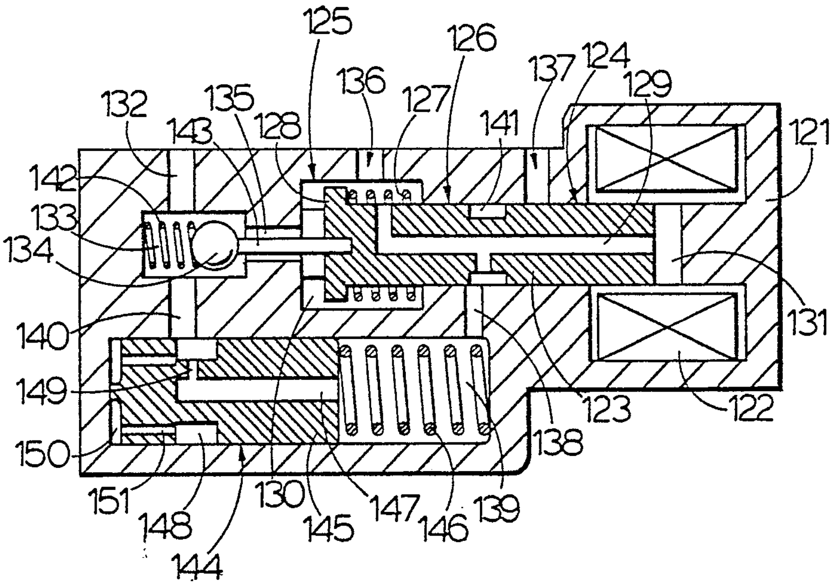
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FIG. 5.

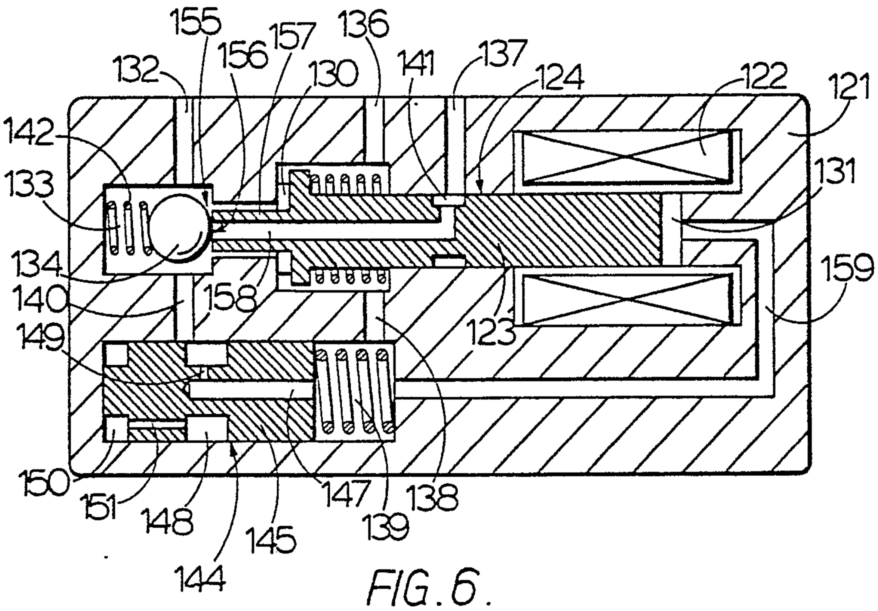
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FIG. 6.

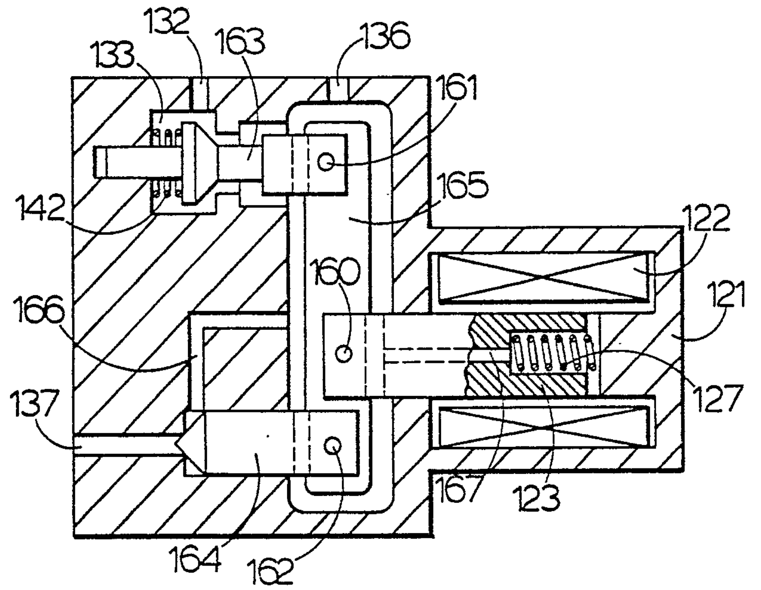
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FIG. 7.

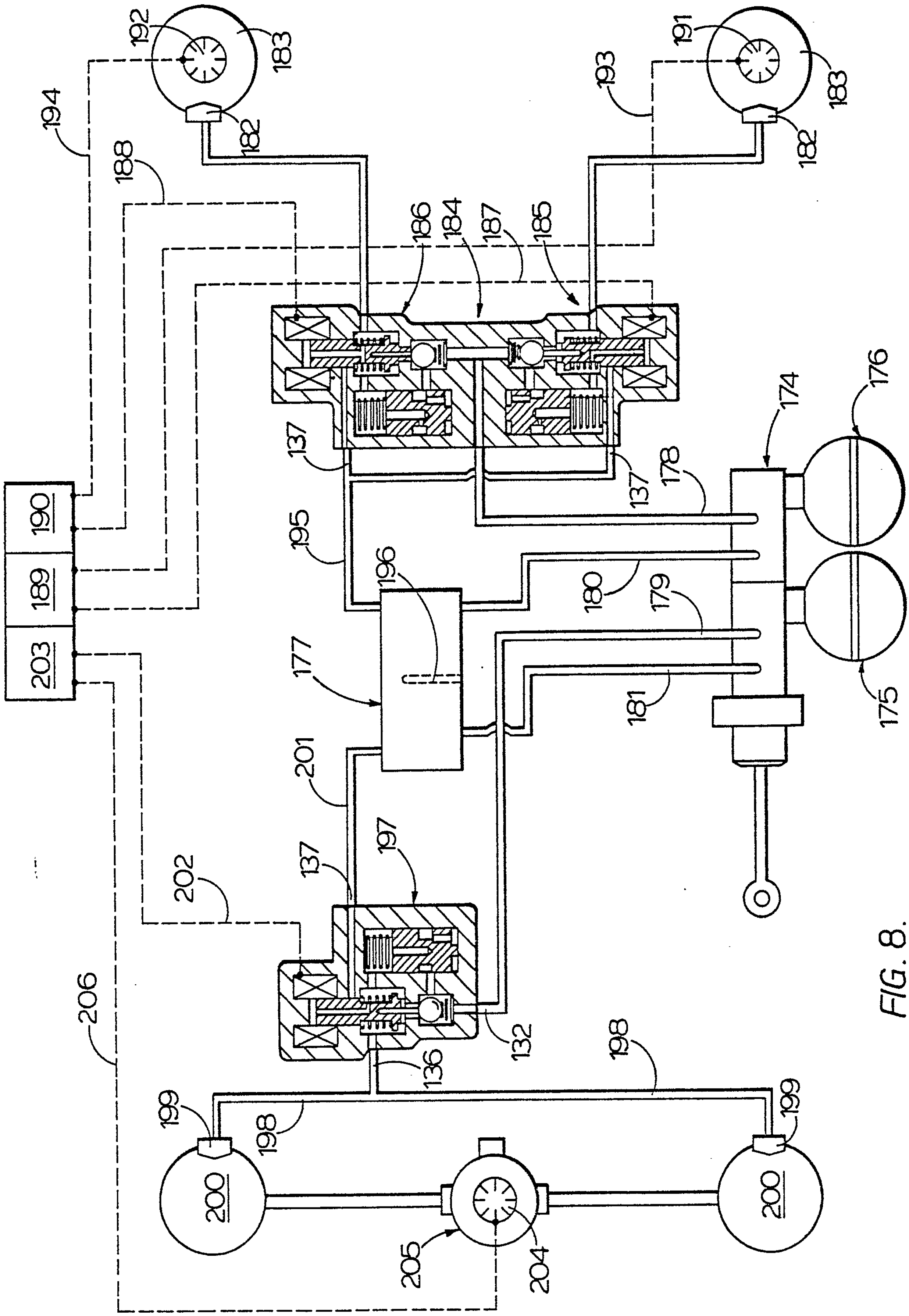
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FIG. 8.

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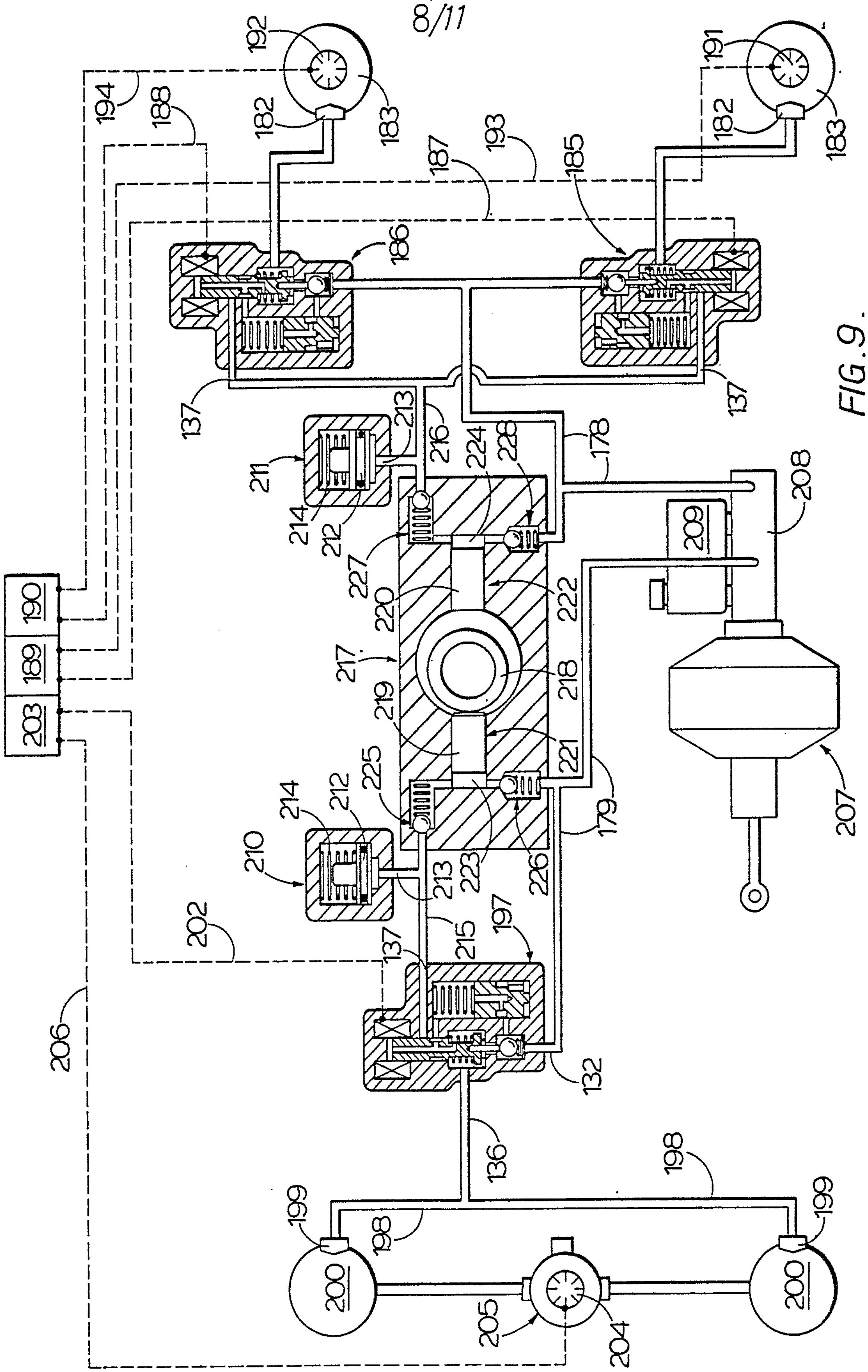
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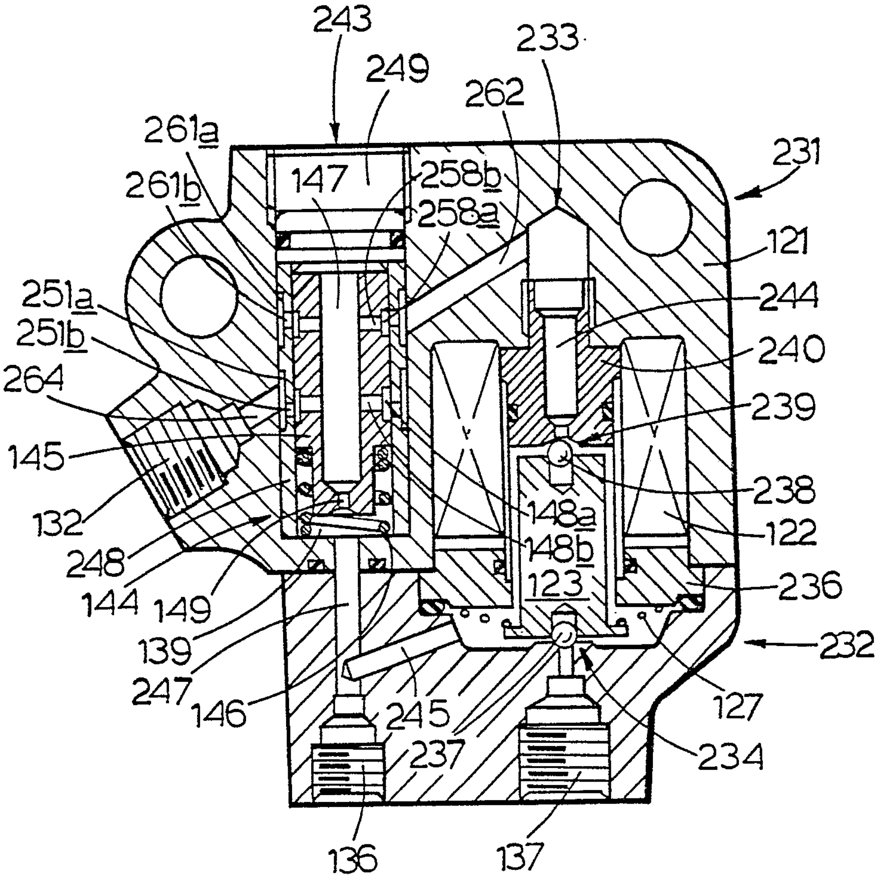
FIG. 9.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147
258b
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261b
121
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251b
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132
148a
122
248
148b
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FIG.10.

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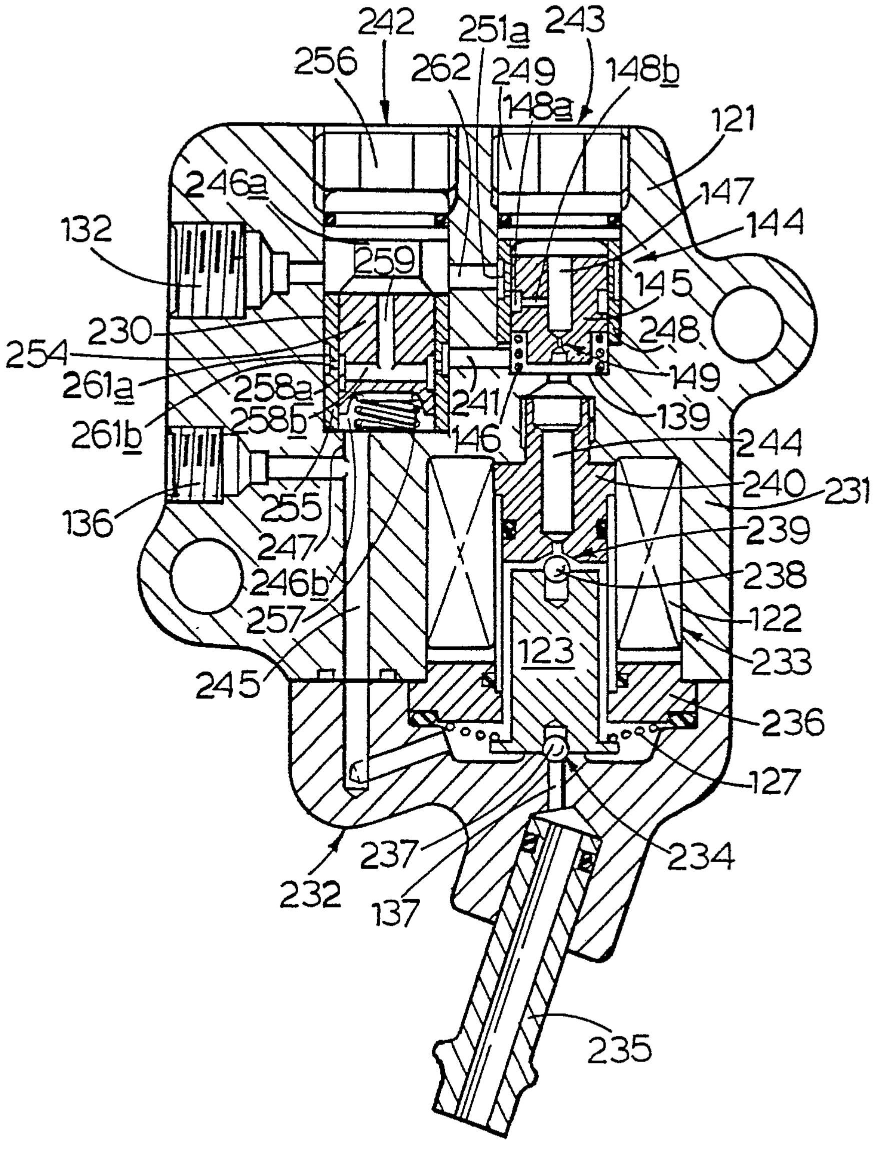
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261b
258b
146
244
136
255
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FIG.11.

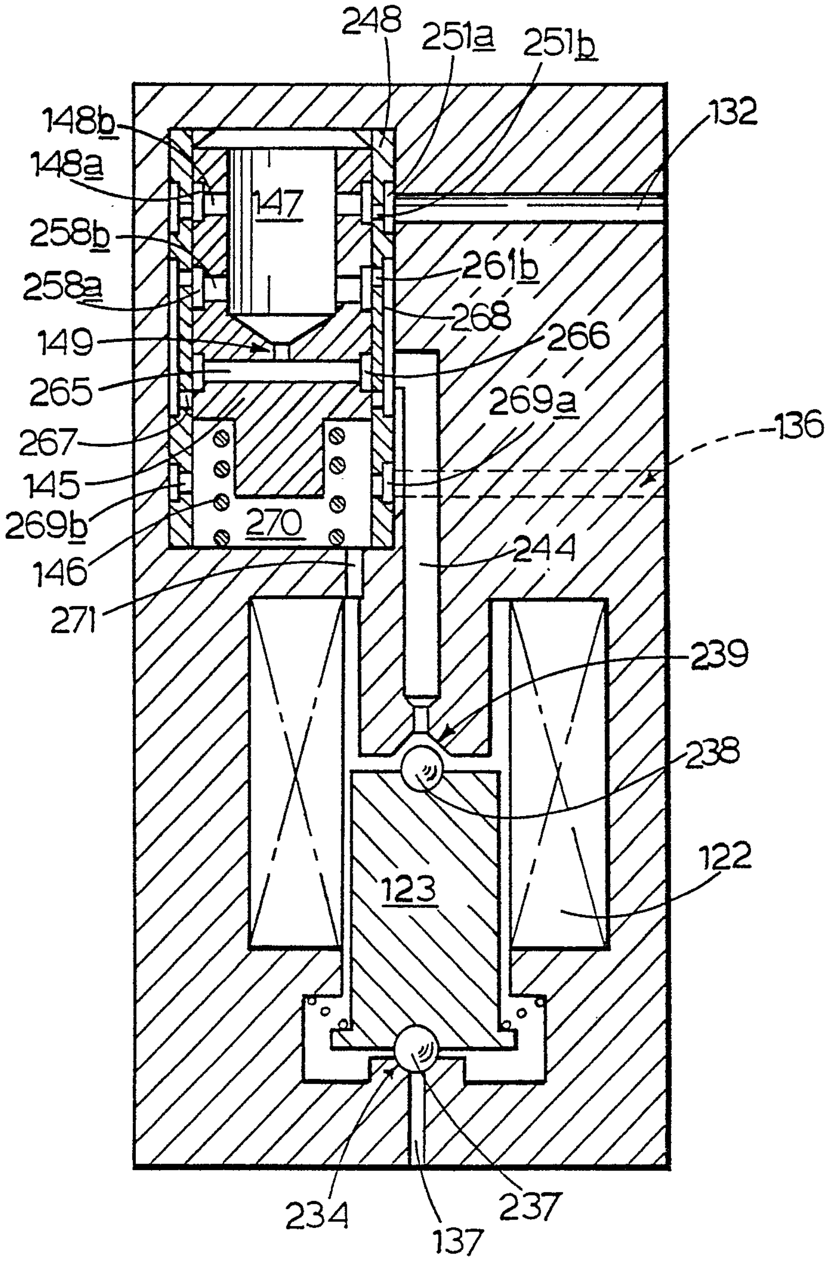
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FIG.12.

INTERNATIONAL SEARCH REPORT

International Application No PCT/GE 80/00030

I. CLASSIFICATION OF SUBJECT MATTER (if several classification symbols apply, indicate all) [3]

According to International Patent Classification (IPC) or to both National Classification and IPC

Int. Cl.^{3}: B 60 T 8/04

II. FIELDS SEARCHED

Minimum Documentation Searched [4]

Classification System	Classification Symbols
Int.Cl.^{3}	B 60 T 8/00

Documentation Searched other than Minimum Documentation to the Extent that such Documents are Included in the Fields Searched [5]

III. DOCUMENTS CONSIDERED TO BE RELEVANT [14]

Category *	Citation of Document, [16] with indication, where appropriate, of the relevant passages [17]	Relevant to Claim No. [18]
	FR, A, 2330573, published July 3, 1977, see the whole document, figures 1 to 8, Fiat	1,7,8,9, 10,12,13, 14 and 15
P	FR, A 2402562 published April 6, 1979, see the whole document, figures 1 and 2, Fiat	1,7,8,9, 10,12,13, 14 and 15
A	FR, A, 2311697, published December 17, 1976, see the whole document, figures 1 to 6, Ferodo	1
A	FR, A, 2141923, published January 26, 1973, see the whole document, figures 1 to 5 Fiat	1
A	FR, A, 2141922, published January 26, 1973, see the whole document, figures 1 to 3, Fiat	1
A	US, A, 3973806, published August 10, 1976, see the whole document, figures 1 to 5, Inada	1
		./.

* Special categories of cited documents: [15]

"A" document defining the general state of the art

"E" earlier document but published on or after the international filing date

"L" document cited for special reason other than those referred to in the other categories

"O" document referring to an oral disclosure, use, exhibition or other means

"P" document published prior to the international filing date but on or after the priority date claimed

"T" later document published on or after the international filing date or priority date and not in conflict with the application, but cited to understand the principle or theory underlying the invention

"X" document of particular relevance

IV. CERTIFICATION

Date of the Actual Completion of the International Search [2]	Date of Mailing of this International Search Report [2]
April 15, 1980	April 22, 1980
International Searching Authority [1]	**Signature of Authorized Officer [20]**
European Patent Office	G.L.M. KRUYDENBERG

Form PCT/ISA/210 (second sheet) (October 1977)

International Application No. PCT/GB 80/00030

FURTHER INFORMATION CONTINUED FROM THE SECOND SHEET

A	US, A, 3856047, published December 24, 1974, see the whole document, figures, Takayama	1

V. ☐ OBSERVATIONS WHERE CERTAIN CLAIMS WERE FOUND UNSEARCHABLE [10]

This international search report has not been established in respect of certain claims under Article 17(2) (a) for the following reasons:

1. ☐ Claim numbers ...., because they relate to subject matter [12] not required to be searched by this Authority, namely:

2. ☐ Claim numbers ...., because they relate to parts of the international application that do not comply with the prescribed requirements to such an extent that no meaningful international search can be carried out [13], specifically:

VI. ☐ OBSERVATIONS WHERE UNITY OF INVENTION IS LACKING [11]

This International Searching Authority found multiple inventions in this international application as follows:

1. ☐ As all required additional search fees were timely paid by the applicant, this international search report covers all searchable claims of the international application.
2. ☐ As only some of the required additional search fees were timely paid by the applicant, this international search report covers only those claims of the international application for which fees were paid, specifically claims:
3. ☐ No required additional search fees were timely paid by the applicant. Consequently, this international search report is restricted to the invention first mentioned in the claims; it is covered by claim numbers:

Remark on Protest

☐ The additional search fees were accompanied by applicant's protest.

☐ No protest accompanied the payment of additional search fees.