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71 Applicant: **Price, William Richards, 19 Grosvenor Close**  
**Ashley Heath, Ringwood Hampshire, BH24 2HG (GB)**

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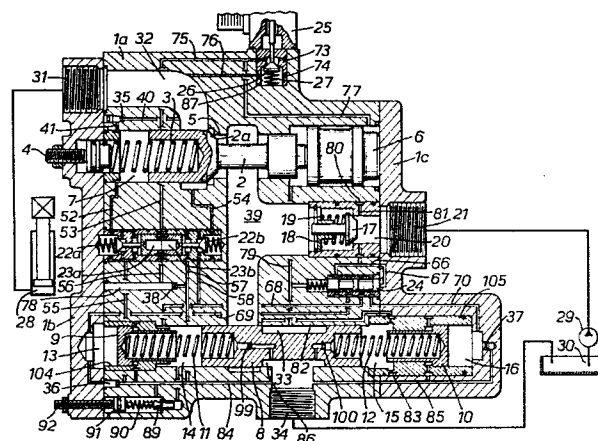
72 Inventor: **Price, William Richards, 19 Grosvenor Close**  
**Ashley Heath, Ringwood Hampshire, BH24 2HG (GB)**

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74 Representative: **Jones, Graham H., Graham Jones &**  
**Company 48 Mereworth Drive Shooters Hill, London**  
**SE18 3EE (GB)**

54 **A hydraulic control system.**

57 A hydraulic control system for regulation of fluid flow rate, in which the fluid flow rate to and from a load actuator (28) is modulated by electric current in a single electromagnetic device (25) such that the flow rate is independent of load magnitude when the system is used without load velocity feedback and such that the flow rate is independent of both load magnitude and fluid viscosity when the system is used with load velocity feedback.



**EP 0 097 007 A2**

A HYDRAULIC CONTROL SYSTEM

This invention relates to a hydraulic control system. More especially, this invention relates to a hydraulic control system for use in either open or closed loop modes, for the bi-directional regulation of speed, acceleration and  
5 deceleration of hydraulically-actuated loads, as typically encountered in hydraulic elevators and machines used in mechanical handling, construction, agricultural and machine tool industries.

Existing known hydraulic control systems exhibit the  
10 following disadvantages, either alone or in combination:-

- (1) 2 or 3 electrical control elements are necessary.
- (2) Open loop systems cannot be used in closed loop mode and vice versa.
- (3) A multiplicity of hydraulic control valves are  
15 necessary.
- (4) Sliding electrical contacts are necessary.
- (5) Relatively high pressures are created during system start-up, reflecting higher than necessary starting loads onto the prime mover.
- 20 (6) Not all hydraulic control elements are biased to the closed position by the load pressure, resulting in poor load holding capability due to oil leakage.
- (7) Relatively high minimum operating pressures are necessary, causing power wastage and limiting minimum

gravity loads.

(8) Operating fluid viscosity range is relatively limited necessitating expensive fluid cooling equipment.

5 (9) Complexity is such that both initial and maintenance costs are high.

(10) Smooth and consistent control, so essential on equipment such as elevators and fork trucks, is not consistently practicable without frequent and expensive servicing.

An aim of the present invention is to overcome or reduce  
10 these disadvantages and to provide an improved, simpler and energy-efficient hydraulic control system capable of operating in either closed or open loop systems over a wide range of loads and load speeds, in a manner which is independent of load magnitude and, in the case of operation in the closed  
15 loop mode, independent of both load magnitude and oil viscosity.

Accordingly, this invention provides a hydraulic control system for regulation of fluid flow rate, in which the fluid flow rate to and from a load actuator is modulated by electric  
20 current in a single electro-magnetic device such that the flow rate is independent of load magnitude when the system is used without load velocity feedback and such that the flow rate is independent of both load magnitude and fluid viscosity when the system is used with load velocity feedback.

25 Preferably, the control system utilises only two main hydraulic elements.

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In the open loop mode of the hydraulic control system of the invention, instantaneous load speed may be regulated in proportion to the magnitude of signal applied by an electric current function generator to a single electric control  
5 element which serves for both directions of load movement. In the closed loop mode of the hydraulic control system of the invention, instantaneous load speed may be regulated in proportion to the signal magnitude applied by an electric signal function generator to an electric summing device, which  
10 is also connected electrically with the single electric control element and a transducer of load speed.

Embodiments of the invention will now be described solely by way of example and with reference to the accompanying drawings, in which:

15 Figure 1 is a schematic diagram of a preferred hydraulic control system;

Figures 2 and 3 show on an enlarged scale parts of Figure 1;

Figures 4, 5 and 6 show alternative forms of  
20 construction; and

Figures 7, 8 and 9 show a typical electric control system external to the hydraulic control system;

Referring now to Figures 1, 2 and 3, the valve block and covers containing the various elements of the control system  
25 are indicated by reference numbers 1a, 1b and 1c.

An actuating cylinder 28 is connected to the valve assembly at a port 31, through which fluid is either directed

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to or from the cylinder. A valve 2, in addition to being a load holding valve, also performs the function of a flow-regulating valve. The valve 2 is urged on to its seat 5 by a spring 3 in addition to the force created by the fluid pressure in the cylinder acting upon the full diameter of the valve 2.

A pressure regulating valve 8 meters flow from a common flow chamber 39 across a land 86 to a reservoir 30, via an annulus 33 and an exhaust port 34, and its position is dictated by the interaction of springs 11 and 12 and fluid pressure acting on the ends of the pressure regulating valve 2 and on two pistons 9 and 10.

A poppet valve 26 is operated by a solenoid 25 which, in turn, is activated by externally applied electric current. The poppet valve 26 is connected on its upstream side with fluid under load pressure in a chamber 32. On its downstream side, the poppet valve 26 is connected to pilot pistons 23a, 23b and 6. The pilot pistons 23a and 23b operate valves 22a and 22b, the former for conditioning pressure in a flow valve spring chamber 7 and the latter for supplying load pressure from the chamber 32 to either chamber 14 or 16 at pre-determined phases in the system operation. The pilot piston 6 controls the operation of the flow regulating valve 2.

A sequence valve 24 is biased to the right hand position by a spring 59 and is connected at its other end to a supply port chamber 20. The sequence valve 24 serves to either

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isolate or communicate the supply chamber 20 with a chamber 13, or to isolate or to communicate load pressure with the chamber 16.

5 A check valve 17 operates in a guide 18 and is urged onto its seat by a spring 19. The check valve 17 isolates a common flow chamber 39 when a pump 29 is not supplying fluid to the system.

Restrictors 35, 36, 37 and 38 fulfill an important function in the operation of the system, the purpose of which will be explained later.

10 Restrictors 99 and 100 in the pressure regulating valve 8 form damping devices. A valve 89 is an overload relief valve.

To explain the function of the control system, its operation as an open loop speed regulator controlling a vertically disposed load will first be described. Figures 7 and 9 show the total control system and load travel diagram respectively.

When neither the solenoid 25 nor the pump 29 are energised, the load is held stationary. In this condition, a load 'W' acting on the actuating cylinder 28 creates a hydraulic pressure in the chamber 32, the spring chamber 7, channels 40, 41, 52, 54 and 76, and in the spring chambers of poppet valves 22a, 22b and 26. All other internal cavities of the system are vented to the reservoir. The poppet valves 22a, 22b and 26, and the flow regulating valve 2, are all of leakfree construction and they are biased to their closed

25

positions by springs 42a, 42b, 27 and 3 respectively. The combined actions of the springs and the load pressure thus act to firmly close the poppet valves 22a, 22b and 26, and valve 2, so holding the load stationary. The sequence valve 24 is  
5 biased to the right by the spring 59, so that channels 66 and 69 are isolated and channels 68 and 70 are connected by means of annulus 63 and ports 64 and 65 in the valve 24 and its bush. The valve 8 is randomly positioned. The springs 11 and 12 are relaxed.

10 To move the load up or down, appropriate trigger switches external to the hydraulic control system are operated. These trigger the current function generator to generate an electric current profile, typically as Figure 9 part 'A' for up and down load movement.

15 Current from the function generator is then applied to the single solenoid 25. The current profile may be varied to suit the particular application and need not be identical for both directions of load movement.

For up movement, only the pump 29 is energised, being  
20 started and stopped nominally simultaneously with the start and stop triggers respectively.

The poppet valve 26, when separated from its seat, allows oil under the influence of load pressure to flow, at a very low rate, from the chamber 32 via channels 76, 75, 53 and 56,  
25 orifice 38, chambers 57 and 84 and tank port 34 to the fluid reservoir 30.

As a result of this pilot oil flow, a control pressure

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will be created in all channels connected with the flow path between the poppet valve 26 and the orifice 38, the magnitude of which is varied by the flow area between the poppet valve 26 and its seat. This area varies with the axial movement of the poppet valve 26 from its seat which is related to the force exerted by the solenoid 25 on the poppet valve 26, which is a function of the electric current applied to the solenoid 25. Thus control pressure magnitude is a function of the current applied to the solenoid.

Control pressure is applied to the pilot piston 6 via a channel 77, and to pilot pistons 23a and 23b via the channels 75 and 53. The opposite sides of the pilot pistons 6, 23a and 23b are maintained at reservoir pressure by channels 79, 55 and 57 respectively, all of which connect with annular chamber 33 and thence to the fluid reservoir.

The pressure regulating valve 8 may be conditioned by the selective actuation of the pistons 9 or 10 to regulate pressure in the common flow chamber 39 to be respectively either a fixed value above load pressure or a fixed value below load pressure, the former condition being applicable to upwards load movement and the latter to downwards load movement.

For upwards load movement, the pump 29 is energised and the oil which is displaced flows via check valve 17, common flow chamber 39, chamber 33 and tank port 34 to the reservoir. The chamber 14 is at reservoir pressure, being vented to the reservoir via channel 68, sequence valve 24, channel 70,

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orifice 37, channel 85 and tank port 34. As the chamber 13 is at reservoir pressure, the piston 9 is able to move to the left, exhausting oil in the chamber 13 to the reservoir via orifice 36, channel 84 and tank port 34. Thus, the spring 11  
5 remains relaxed and so does not exert an appreciable force on the valve 8. The valve 8, if not disposed fully left when the pump is energised, is instantaneously so positioned because any pressure which tends to develop in the flow chamber 39 is communicated via channel 82 to the chamber 15, where it  
10 results in a force acting on the valve 8 which moves it left until it abuts a bushing 104.

The valve 8, in the full left position, presents an ultra-low impedance to oil flow from the chamber 39 to the chamber 33 and hence to reservoir. Pressure at the pump is determined initially by that needed to open the check valve  
15 17, which may be determined to be extremely low. Thus the pump prime mover is started in a virtually completely unloaded mode.

The pump pressure existing upstream of the check valve 17 is applied via channel 67 to the right end of the sequence  
20 valve 24, the left end of which is vented to the reservoir via channel 79, chamber 33 and tank port 34. By appropriate design of the spring 59, the sequence valve 24 may be designed to move full left virtually simultaneously with the pump starting to operate, so that the channels 66 and 69 previously  
25 isolated, become connected, and the channels 68 and 70 previously connected become isolated.

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Connections of the channels 66 and 69 cause pump pressure to be applied to the chamber 13, where it acts on the left side of the piston 9, the right side of which is at reservoir pressure by virtue of its connection with the port 34 and the  
5 channel 84.

As the control pressure, in response to the increasing electric current flowing from the function generator to the solenoid 25, gradually increases, it reaches a value which is sufficient to separate the poppet 22b from its seat. This  
10 connects the load pressure in the chamber 32, via channels 54 and 58, to the chamber 14 where it acts on the left end of the valve spool 8.

The valve spool 8 now has load pressure acting on its left hand end and chamber 39 pressure - virtually reservoir  
15 pressure - acting on its right hand end. The valve spool 8 thus moves to the right, increasing the impedance to oil flow from the chamber 39 to 33 and the reservoir port 34. The increased impedance raises chamber 39 pressure and hence pump pressure, which is applied to the piston 9.

20 The chamber 16 is at reservoir pressure, so the piston 10 gradually moves to the right as chamber 39 pressure, connected with the chamber 15 via channel 82, increases. This leaves the spring 12 in a relaxed condition.

The diameter of the piston 9 in the chamber 13 has a  
25 larger area than the diameter in the chamber 14 so, under the action of the increasing pump pressure, the piston 9 moves to the right until it registers against the abutment in sleeve

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104. In registering against this abutment, the piston 9 positions the left end of the spring 11 in a fixed axial relationship with the flow metering land 86 of the pressure regulating valve 8, which is thus conditioned to function as a pressure reducing valve, whereby it will maintain the pressure in the chamber 39 at a nominally fixed value above load pressure, variations in the pressure difference being dependent on the rate of the spring 11 which may be designed to be substantially constant over the operating stroke of the spool 8. This pressure difference will be maintained irrespective of variations in load pressure, so that oil flow rate from the chamber 39 across the flow regulating valve 2 is dependent only on the flow area created when the valve 2 is separated from its seat 5.

15       The pilot piston 23a, whose left end is vented to reservoir via channels 55 and 84 and tank port 34, moves left under the action of control pressure applied to its right end and so separates the poppet 22a from its seat. This action causes the spring in chamber 7, previously at load pressure, to be vented to reservoir pressure. Oil flow from the load actuator to reservoir is limited to a very low rate by the orifice 35.

25       The flow regulating valve 2, now biased closed only by the spring 3, is thus able to respond smoothly to control pressure variations applied to the pilot piston 6. As the current in the solenoid 25 is modulated, in this typical example according to Figure 9, the load actuator will

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accelerate, move at constant speed, then decelerate, move at slow speed and stop, as an analogue of the current in the solenoid 25.

To move the load down, only the solenoid 25 is energised,  
5 the pump remaining stationary.

Control pressure is generated as previously described and applied to all pilot pistons and chambers in an identical manner as for upward load movement. The spring chamber 7 is vented to reservoir in the same way as for upward load  
10 movement. As the pump is not energised, pressure in chamber 20, upstream of check valve 17 will be at reservoir level and hence the sequence valve 24 will remain disposed to the right under the action of the spring 59. Thus the channels 68 and 70 will be connected via the ports 64 and 65 and annulus 63 in  
15 the sequence valve 24.

When the poppet valve 22b is separated from its seat under the influence of increasing control pressure, load pressure from the chamber 32 will now, in addition to being connected to the chamber 14 as for upward load movement also  
20 be applied to the right side of the piston 10 in the chamber 16.

As the left side of the piston 10 is at reservoir pressure by virtue of its connection with the reservoir port 34 via the channel 85, the piston 10 will move to the left  
25 until it abuts the step in the bush 105. Continued increase in control pressure will be communicated through the control valve 26 to the right hand end of the control piston 6 and thus separate the flow regulating valve 2 from its seat 5. As

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this occurs, load pressure will be communicated to the chamber 39 and thence via channel 82 to the chamber 15, where it will act on the spool 8.

As piston 10 area in the chamber 16 is larger than that  
5 in the chamber 15, load pressure acting in the chamber 16 on the piston 10 will displace the piston 10 left until it registers against an abutment in the sleeve 105. In so registering, the piston 10 positions the right end of the spring 12 in a fixed axial relationship with the flow metering  
10 land 86 of the pressure regulating valve 8, which is thus conditioned to function as a pressure reducing valve whereby it will maintain the pressure in the chamber 39 at a nominally fixed value below load pressure, variations in the pressure difference being dependent on the force of the spring 12 which  
15 may be designed to be substantially constant within very small limits, over the operating stroke of the spool 8. This pressure difference will be maintained irrespective of variation in load pressure, so that oil flow rate from the chamber 32 across the flow regulating valve 2 is dependent  
20 only on the flow area created when the valve 2 is separated from its seat.

The piston 9 is able to move fully left, as the chamber 13 is at reservoir pressure, so ensuring that the spring 11 is relaxed.

25 The flow regulating valve 2 is thus able to respond to control pressure variations applied to the pilot piston 6, in principle the same as for upward load movement.

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The orifices 36 and 37 serve the dual functions of limiting flow rate to the reservoir when the chambers 13 and 16 are pressurised, and allowing both chambers to vent when the pistons 9 or 10 move away from their respective  
5 abutments.

The relief valve 89 provides an overload relief function for the system when the pump 29 is activated. Pressure from the common flow chamber 39 is transmitted via channel 82 to the chamber 15. If the pressure exceeds the predetermined  
10 limit, the pressure regulating valve 8 is urged to the left. The same pressure is transmitted to the chamber 13 but since the head of the piston 9 is larger than its second diameter in the chamber 14, the piston 9 will be urged to the right.

Pressure in the chamber 14 then becomes equal to the  
15 excess pressure in the chamber 15 and is exhausted through the relief valve 89, thus allowing the valve 8 to move to the left into the space created by the displaced fluid and enabling excess pressure in the chamber 39 to exhaust across the annulus 33 back to the reservoir.

20 Figure 4 illustrates a simple alternative construction for the pistons 9 and 10 in which a single piece form has been replaced by a two piece design consisting of a large piston 71 and a smaller piston and spring guide 72.

It will be appreciated that by modifying the force  
25 exerted by the pressure regulating valve springs 11 and 12, either individually or both at the same time, the characteristics of this valve and hence the control system can be altered to suit requirements. To this end, alternative constructions of this valve sub-assembly are illustrated.

30 In Figure 5, shims 101 are added inside the spring recess between the valve springs 11 and 12 and their respective

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pistons 9 and 10 to adjust the spring tension.

Figure 6 indicates yet another construction in which the pistons 9 and 10 are encapsulated by a sliding bush 93 and an end cap 95 and a sliding bush 96 and an end cap 97 respectively. The position of the sliding bushes and end caps can thus be modified externally to change the influence of the springs 11 and 12 by means of adjusting screws 94 and 98 without dismantling the pressure regulating valve.

Operations in the closed loop mode will now be described and reference will be made to Figures 8 and 9.

Instantaneous load speed is electrically transduced at 8.1 and its signal is applied to one element of a differential comparator system 8.2, the second element of which is connected with a signal function generator 8.3, and the third element of which is connected, via an amplifier, with the solenoid 25.

To move the load up or down, appropriate trigger switches external to the control system are operated. These trigger the function generator 8.3 to generate electric signal profiles typically as Figure 9 part 'A' and 'B' for up and down load movement respectively.

The differential comparator connected with the solenoid 25 receives a signal which is the arithmetic sum of the function generator signal and the load speed transducer signal.

Operation of the system is otherwise identical with that described for the open loop mode, with the added advantage

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that fluid viscosity, which will cause inaccuracies in the open loop mode due to the relationship Flow Rate Area/Fluid Viscosity, will not affect the performance of the system in the closed loop mode.

5        It is to be appreciated that the embodiments of the invention described above have been given by way of example only and that modifications may be effected. Thus, for example, an alternative construction for a control pressure pilot valve to that described above with reference to the  
10       poppet valve 26 is a 3-way valve comprising a spool located in a closed housing, with three lands spaced longitudinally along the spool and arranged to create two hydraulically separate chambers with the housing.

      One of the chambers (referred to as the "T" chamber) is  
15       connected to the hydraulic reservoir via port 34 and the other chamber (referred to as the "P" chamber) is connected to the load port 31 via a poppet-type flow check valve (non-return valve), the arrangement of which in relation to the 3-way valve spool is subsequently described.

20       The end of the spool nearest the "T" chamber is acted upon by a spring which gives a biasing force which tends to move the spool towards an electro-magnetic device located at the opposite end of the spool and so arranged that the armature of the electro-magnetic device is in contact with the  
25       end of the spool nearest the "P" chamber.

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The centre land of the spool overlaps a hole in the housing disposed perpendicularly to the spool, the hole being hydraulically connected to a chamber at the end of the spool to which the biasing force is applied and hydraulically  
5 connected to channels 53,56,75,77.

The spool between the centre land and the land at the electro-magnetic device end of the spool is tapered such that its diameter where it joins the centre land is smaller than that where it joins the land at the electro-magnetic device  
10 end of the spool.

The poppet-type check valve is arranged so that its polar axis is nominally perpendicular to and in the same plane as the axis of the 3-way valve spool. The poppet is arranged so that hydraulic pressure from port 31 presses the poppet onto  
15 its seat, thus effectively sealing the load pressure from the "P" chamber.

The poppet is lightly spring biased onto its seat in the housing and the stem of the poppet is so arranged in relation to the tapered portion of the 3-way valve spool that a slight  
20 longitudinal movement of the spool towards the biasing force end will lift the poppet from its seat, thus admitting pressure from port 31 to the "P" chamber.

The functional operation of the alternative control pressure pilot valve is as follows. Electric current applied  
25 to the electro-magnetic device creates a force which displaces

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the spool which in turn lifts the poppet, so admitting pressure (i.e. load) to the "P" chamber. Continued displacement of the spool as a result of increasing electric current eventually eliminates the overlap of the centre land  
5 of the spool with the hole in the housing which is connected to channels 53,56,75,77 and the end of the spool to which the biasing force is applied. The pressure acting on the end of the spool creates a force which acts in opposition to that exerted at the other end of the spool by the electro-magnetic  
10 device. The spool adopts a position of equilibrium determined by the opposing hydraulic, electro-magnetic and biasing forces, such that the hydraulic pressure existing at the force-biased end of the spool will be a function of the electric current in the electro-magnetic device.

CLAIMS:

1. A hydraulic control system for regulation of fluid flow rate, in which the fluid flow rate to and from a load actuator is modulated by electric current in a single electro-magnetic device such that the flow rate is independent of load  
5 magnitude when the system is used without load velocity feedback and such that the flow rate is independent of both load magnitude and fluid viscosity when the system is used with load velocity feedback.
2. A control system according to claim 1 and including a  
10 flow regulating valve which is biased to a closed position and which varies the impedance to fluid flow between a common flow chamber and a system load port in both directions of fluid flow.
3. A control system according to claim 1 or claim 2 and  
15 including a pressure regulating valve one end of which is permanently connected with a common flow chamber and the other end of which is connectable with a load port, which pressure regulating valve when appropriately conditioned by logic devices in the control system, varies the impedance to fluid  
20 flow between the common flow chamber and a fluid reservoir port of the control system and so maintains fluid pressure in the common flow chamber at a fixed value above that in the load port when fluid flow is from the common flow chamber to

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the load port, and at a fixed value below that in the load port when fluid flow is from the load port to the common flow chamber.

4. A control system according to any one of the  
5 preceding claims and including a control pressure pilot valve biased to a closed position, the flow area of the control pressure pilot valve being modulatable by means of an electric current in the electro-magnetic device, the electro-magnetic device being in contact with the control  
10 pressure pilot valve, which control pressure pilot valve when open permits fluid to flow from a load port to a fluid reservoir port via a fixed area orifice and thus enables channels between the control pressure pilot valve and the fixed area orifice to receive a control pressure,  
15 the magnitude of which is related, for a given load port pressure, to the magnitude of electric current in the electro-magnetic device.

5. A control system according to any one of the preceding claims and including a first conditioning piston  
20 located at one end of a pressure regulating valve bore, so situated to form a chamber between itself and a pressure regulating valve in the pressure regulating valve bore, which chamber is connectable with a load port, the first conditioning piston having a second larger diameter  
25 portion the underside of which forms a space between the step of the two diameters which is permanently connected and vented to a fluid reservoir port, and the other

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side of the first conditioning piston forms a chamber at the end of the valve bore connectable with a source of fluid pressure, there being a spring interposed between the first conditioning piston and the pressure regulating valve so that  
5 when the first conditioning piston is subjected to the action of differential pressure resulting from pressure from the control system fluid energy source, the first conditioning piston moves against the resistance of the spring until it abuts the step between the first and second diameters of the  
10 valve bore which provides a pre-arranged axial relationship with the pressure regulating valve.

6. A control system according to claim 5 and including a second conditioning piston located at the other end of the pressure regulating valve bore to the first conditioning  
15 piston and situated in a similar manner to form a chamber between itself and the pressure regulating valve and permanently connected with a common chamber, there being an intermediate space formed by the stepped bores and the first and second diameters of the second conditioning piston  
20 permanently connected and vented to the fluid reservoir port, and the other side of the second conditioning piston forming a chamber at the end of the valve bore connectable with pressure existing in the load port so that when the second conditioning piston is subjected to differential pressure transmitted from  
25 this source, the second conditioning piston moves against the resistance of the spring until it abuts the step between the

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first and second diameters of the valve bore which provides a pre-arranged axial relationship with the pressure regulating valve.

7. A control system according to claim 5 or claim 6 and  
5 including a pressure regulator conditioning valve biased to a closed position directly by load pressure via a bore permanently connected with a load port, and opened directly or indirectly by control pressure, with the other side of the pressure regulator conditioning valve connected permanently by  
10 means of a bore with the chamber formed between the end of the pressure regulating valve and the first conditioning piston.

8. A control system according to claim 1 and including a flow regulator conditioning valve biased to a closed position by pressure transmitted from a load chamber via a channel, an  
15 orifice, a spring chamber and a bore and opened directly or indirectly by control pressure, with the other side of the flow regulator conditioning valve being permanently connected through a bore with a fluid reservoir.

9. A control system according to claim 1 and including a  
20 flow check valve which permits flow to a common flow chamber from a fluid energy port of the control system but not vice versa.

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10. A control system as claimed in claim 2 and including an orifice located in a channel connecting the load port, with a chamber situated at that end of the flow regulating valve to which the closing bias is applied.
- 5 11. A control system according to claim 2 or claim 10 and including a control piston located at the opposite end of the flow regulating valve to which the closing bias is applied and able, when subjected to a pre-conditioned control pressure, to modulate the position of the flow regulating valve, with the  
10 side of the control piston nearest the flow regulating valve permanently connected with a fluid reservoir port and the opposite side of the control piston permanently connected with a channel which receives the control pressure.
12. A control system according to claim 7 and including a  
15 pilot piston which operates the pressure regulator conditioning valve, the pilot piston having one side permanently connected with the fluid reservoir port and the opposite side remote from the pressure regulator conditioning valve connected with a channel which receives the control  
20 pressure.
13. A control system according to claim 8 and including a pilot piston which operates the flow regulator conditioning valve, the pilot piston having one side permanently connected with the fluid reservoir port and the opposite side remote

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from the flow regulator conditioning valve connected with a channel which receives the control pressure.

14. A control system according to claim 7 and including a sequence valve having four ports permanently connected by individual channels, the first port being connected through a channel with a fluid energy port, the second port being connected by a channel with that side of the first conditioning piston which receives the fluid pressure, the third port being connected with a channel which receives the load port pressure when the pressure regulator conditioning valve is open, and the fourth port being connected by a channel with that side of the second conditioning piston which receives the load pressure, and the sequence valve being arranged so that when positioned only by the biasing means, the sequence valve isolates the first and second ports from each other and connects the third and fourth ports together, and when operated by pressure from the fluid energy port the sequence valve connects the first and second ports and isolates the third and fourth ports from each other.

15. A control system according to claim 14 and including a permanent connection channel between the chamber upstream of the flow check valve and the end of the sequence valve opposite to that to which the bias is applied.

16. A control system according to claim 6 and in which piston

bores of the first and second conditioning pistons which are on that side remote from the pressure regulating valve and which are subjective to the application of pressure, are each connected with the reservoir port via fixed area orifices.

5 17. A control system according to claim 6 or claim 16 in which the influence exerted by the two springs on the pressure regulating valve is modified by shims located between the ends of the springs and their respective pistons, and hence the characteristics of the pressure regulating valve.

10 18. A control system according to claim 6, claim 16 or claim 17 in which the pressure regulating valve conditioning pistons are encapsulated by sliding bushes and end caps so that the effective force of the pressure regulating springs and hence the characteristics of the pressure regulating valve, can be  
15 adjusted by modifying the positions of abutment screws which are in direct contact with the end caps.

19. A control system according to any one of the preceding claims and including a three-element electric summing device connected electrically with the single electro-magnetic device  
20 and a load velocity transducer in such a way that electric current in the electro-magnetic device regulates fluid flow rate between the control system and the load actuator to be in a particular relationship with the magnitude of an electric demand signal applied to the third element of the summing

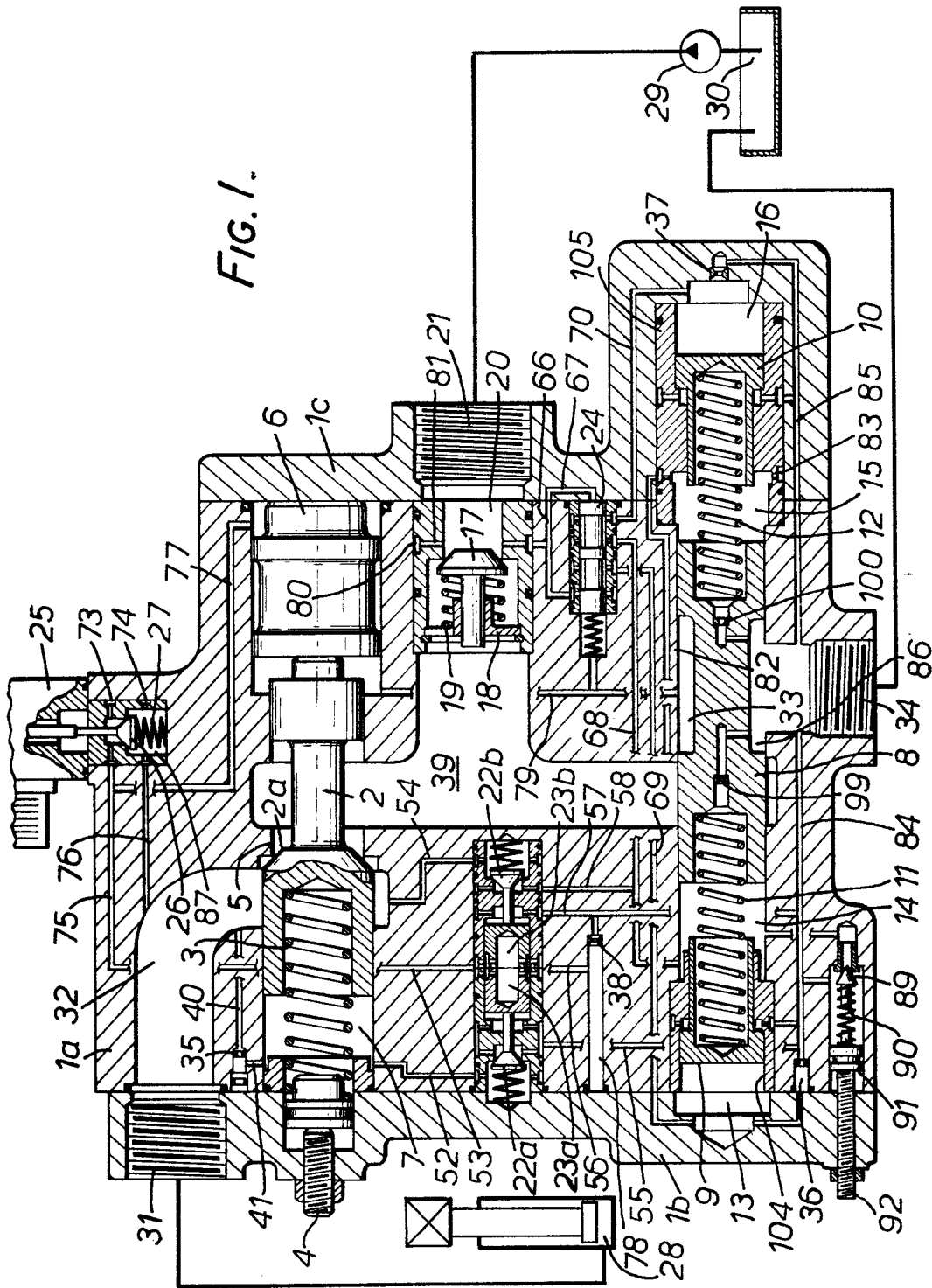
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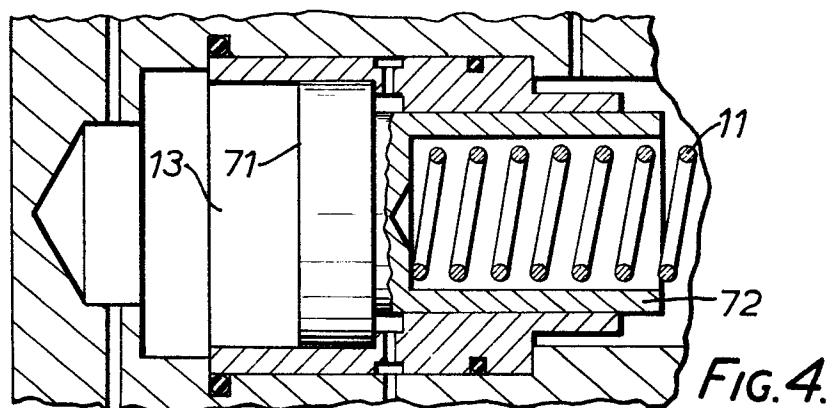
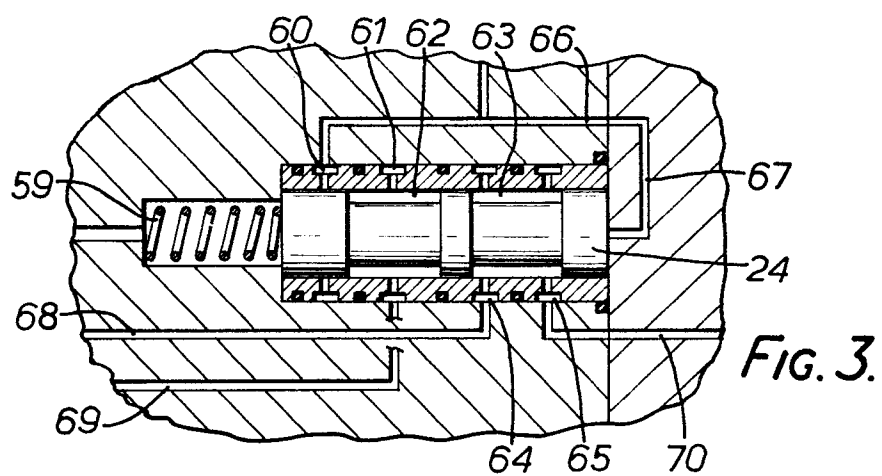
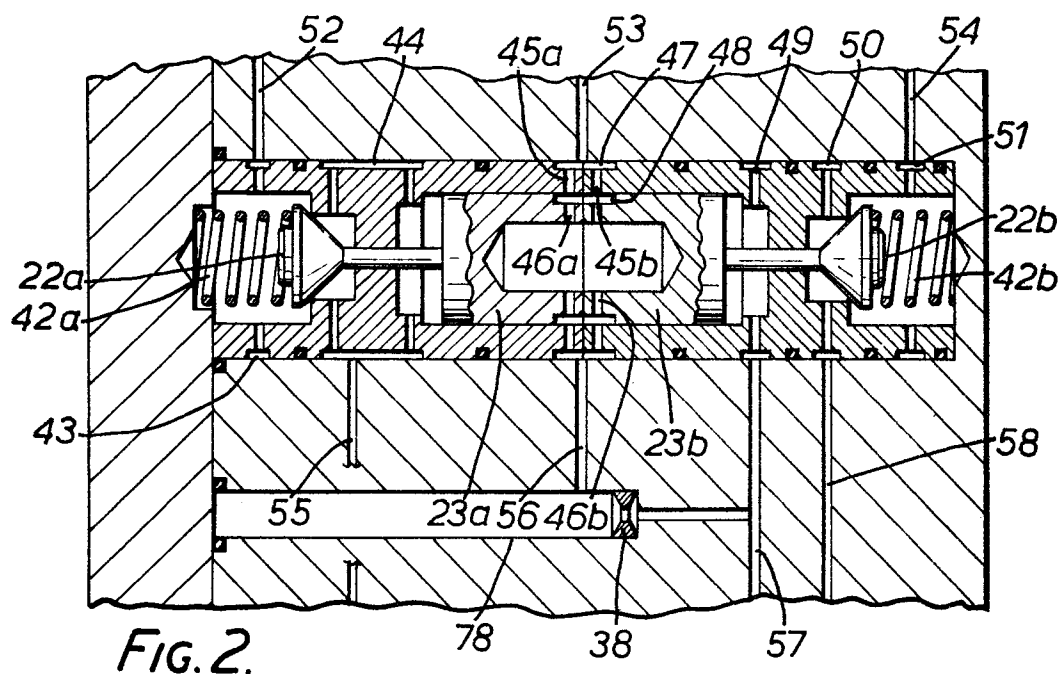
device.

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FIG. 1.



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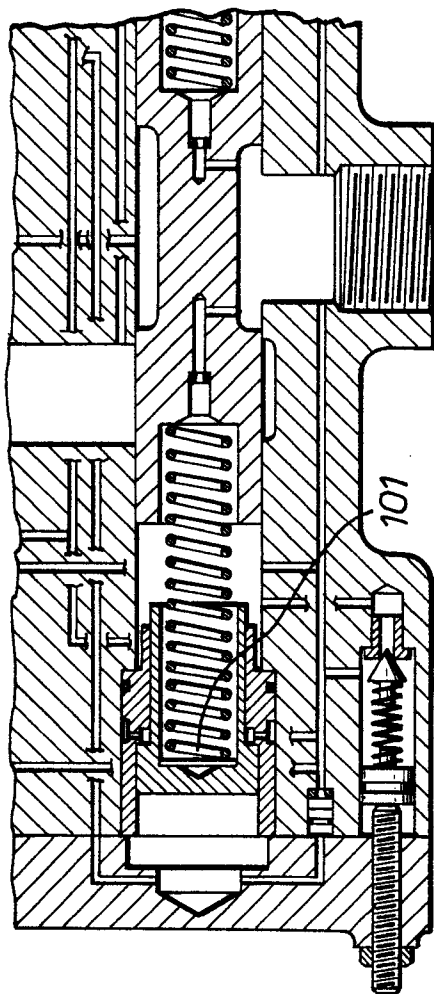


FIG. 5.

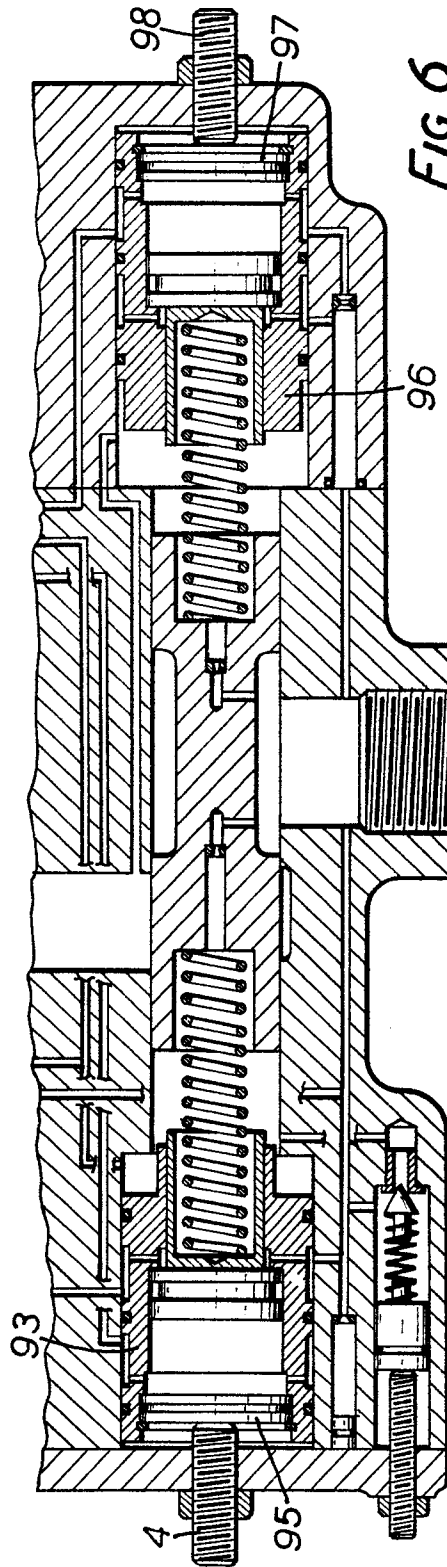


FIG. 6.

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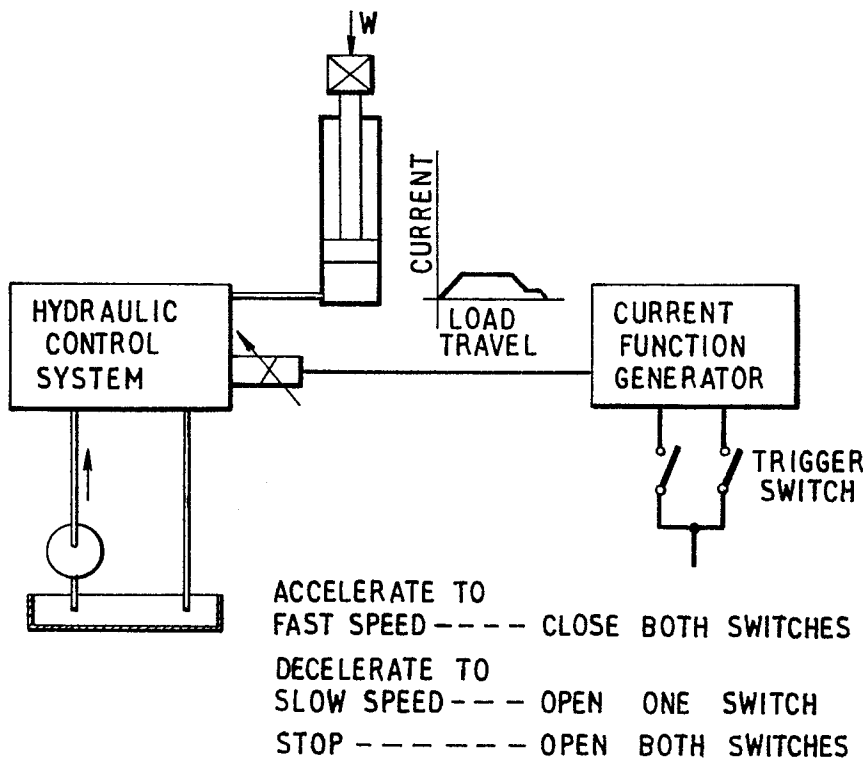


Fig. 7.

— OPEN LOOP SYSTEM —

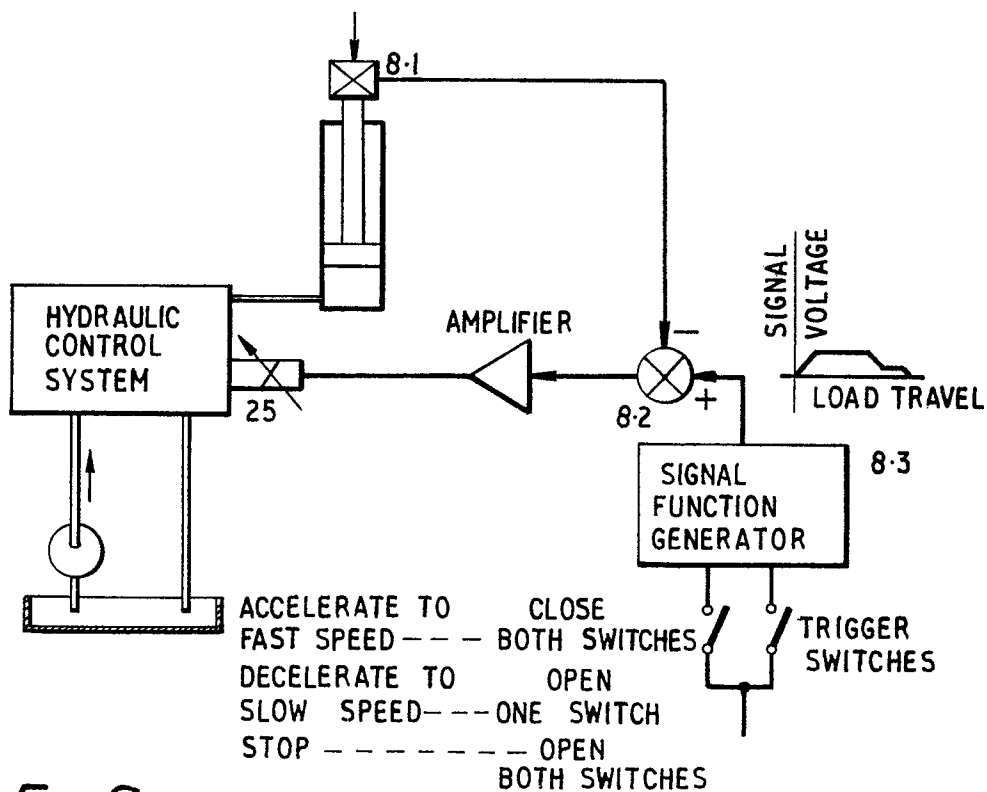
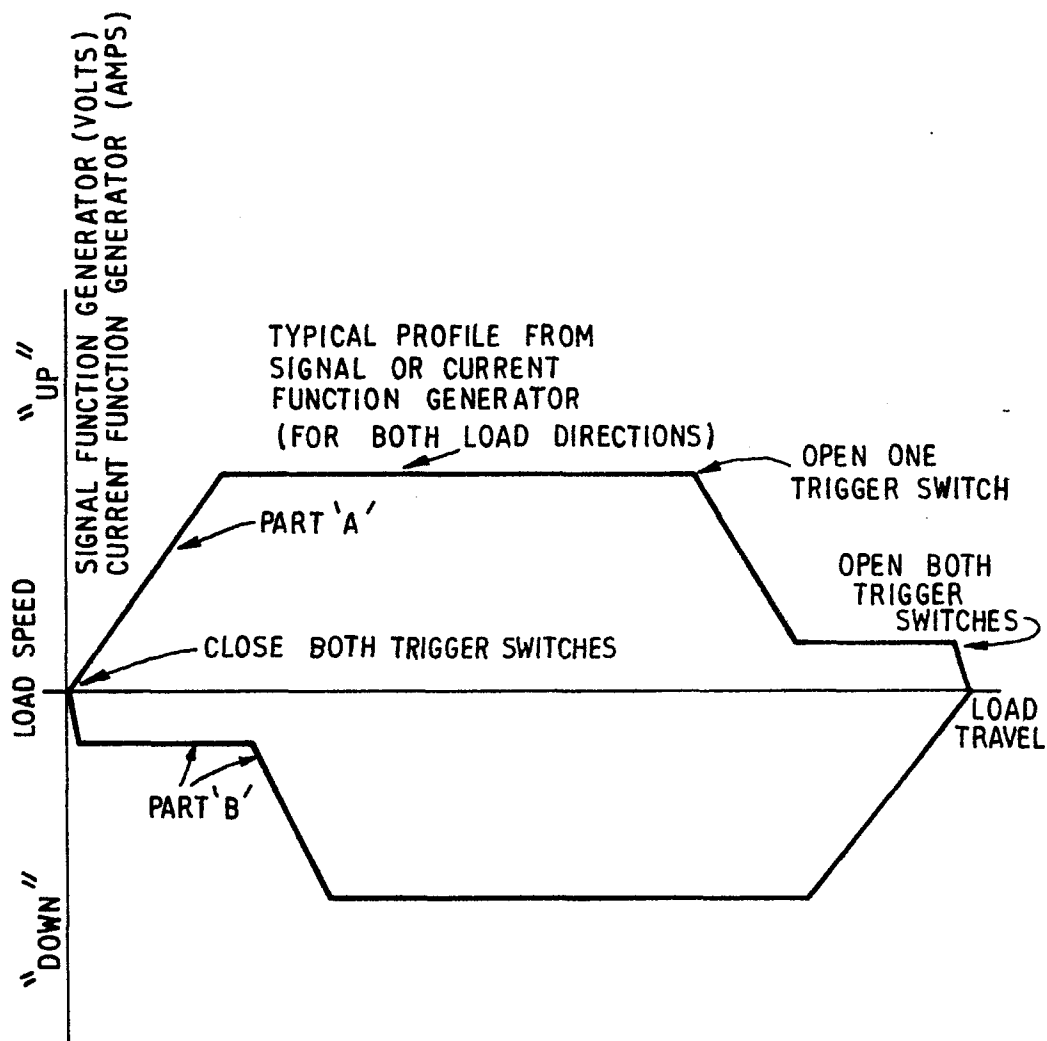


Fig. 8.

— CLOSED LOOP SYSTEM —

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— LOAD SPEED — TRAVEL DIAGRAM —

Fig. 9.