United States Patent

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[45] Patented **June 29, 1971**
[73] Assignee **Sundstrand Corp.**

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[54] **PROPORTIONAL DIRECTIONAL CONTROL UNIT**
11 Claims, 20 Drawing Figs.

[52] **U.S. Cl.** **60/52,**
137/625.23, 137/625.47, 91/375
[51] **Int. Cl.** **F15b 13/04**
[50] **Field of Search** 60/525;
180/79.2; 137/625.23, 625.47; 91/375

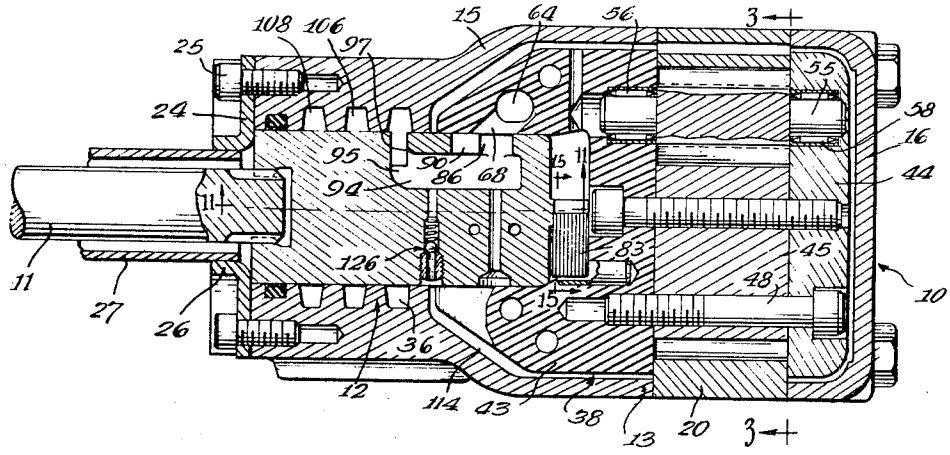
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Primary Examiner—Edgar W. Geoghegan
Attorney—Hofgren, Wegner, Allen, Stellman & McCord

ABSTRACT: A hydraulic servo device which controls the delivery of high-pressure fluid in an amount proportional to a manual input signal including a rotary valve for supplying fluid to a rotary servo metering device that follows movement of the valve about the same axis delivering fluid to a remotely disposed actuator, with the servo metering device taking the form of a stationary internally toothed ring gear, a carrier supporting a plurality of planetary gears interengaging the stationary ring gear and defining therewith a plurality of expanding and contracting fluid chambers, there being provided in the rotary valve a supply port and two adjacent load ports which continuously communicate with the remote actuator for supplying fluid to and from the same, with a plurality of pressure-balancing recesses disposed oppositely in the rotary valve with respect to each of the ports in the valve, there being also provided two ports in the carrier for supplying fluid to and from the servo device so that the latter device acts in follow up action with respect to the rotary valve through the selective communication between the ports in the carrier and the supply port in the rotary valve. The rotary valve may be either of the closed center or open center type and may also be either load reactive or nonload reactive through the proper location of the load ports in the valve with respect to the two ports in the carrier member.



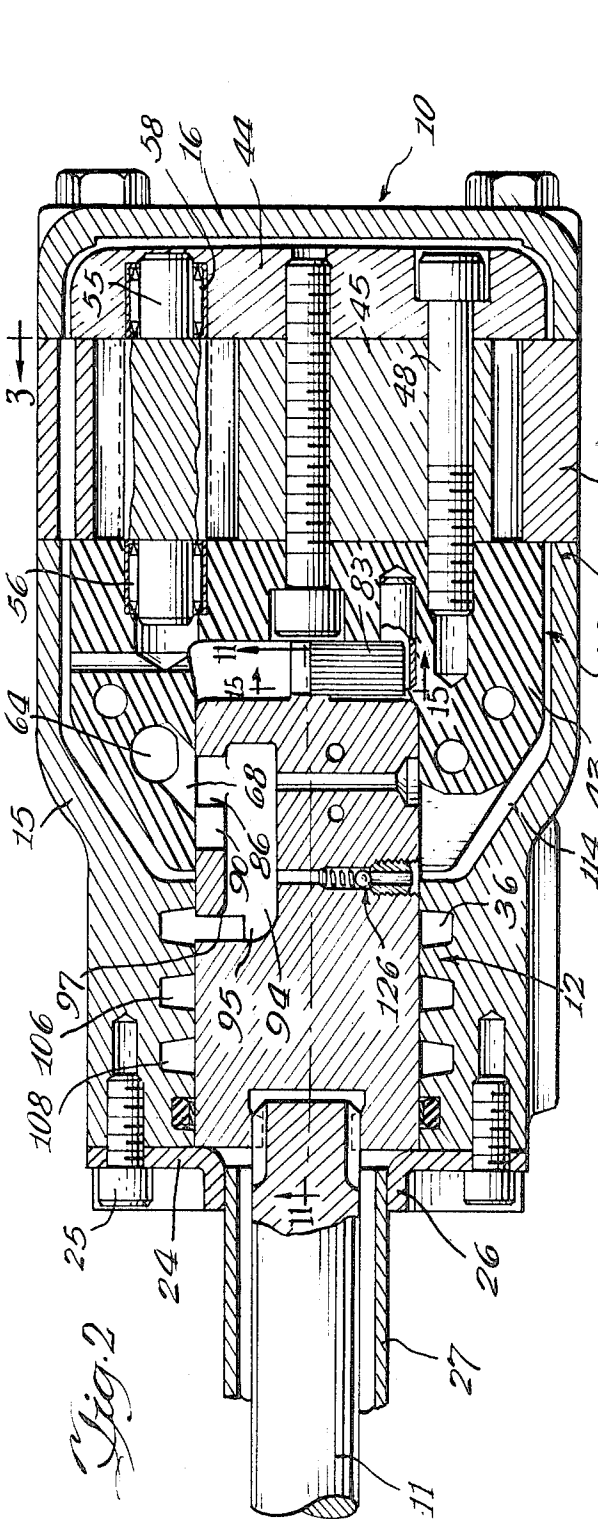


Fig. 2

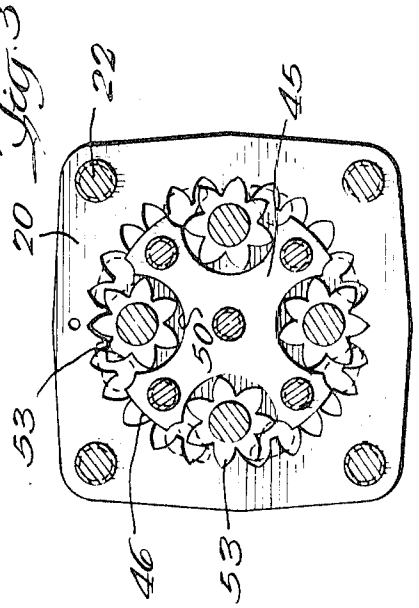


Fig. 3

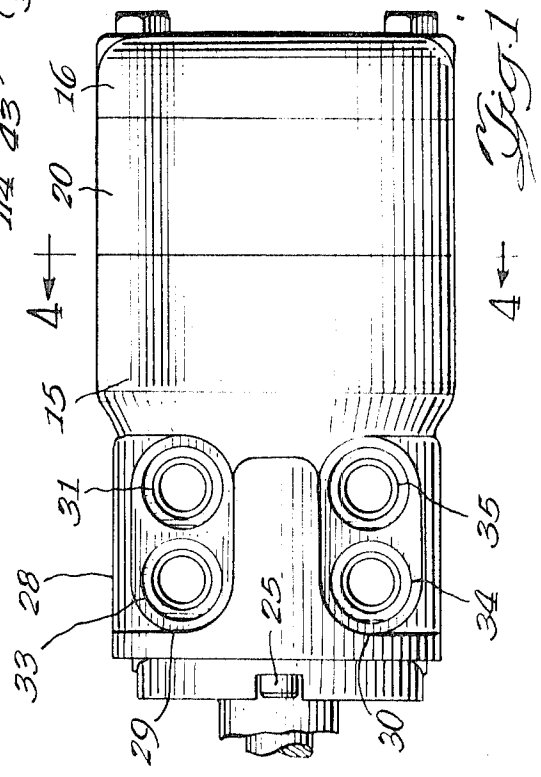


Fig. 1

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ATTORNEYS

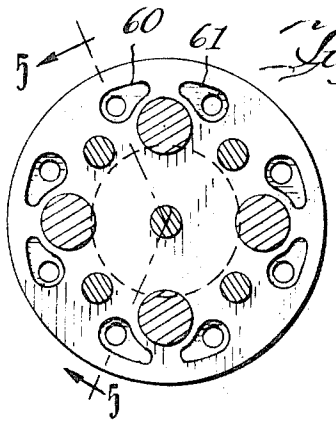


Fig. 4

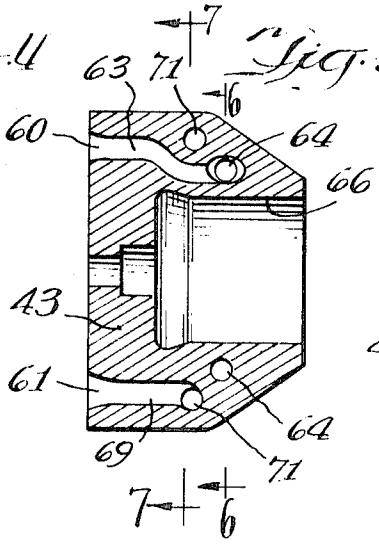


Fig. 5

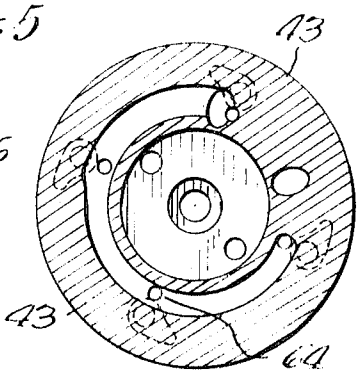


Fig. 6

Fig. 7

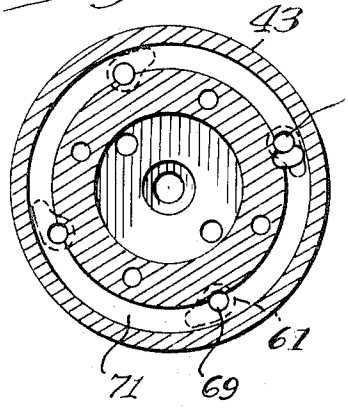


Fig. 8

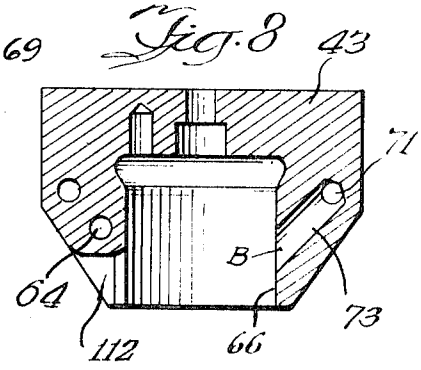


Fig. 15

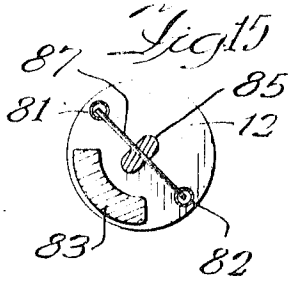
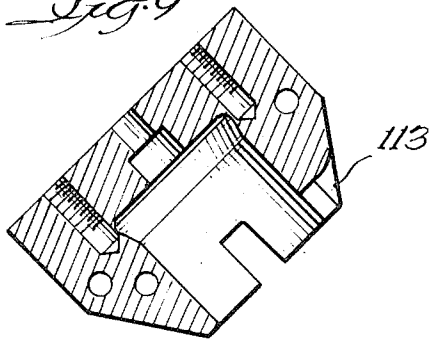


Fig. 9



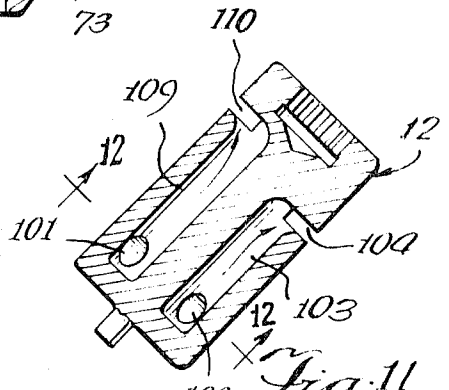


Fig. 11

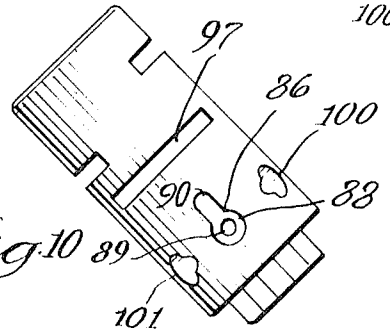


Fig. 10

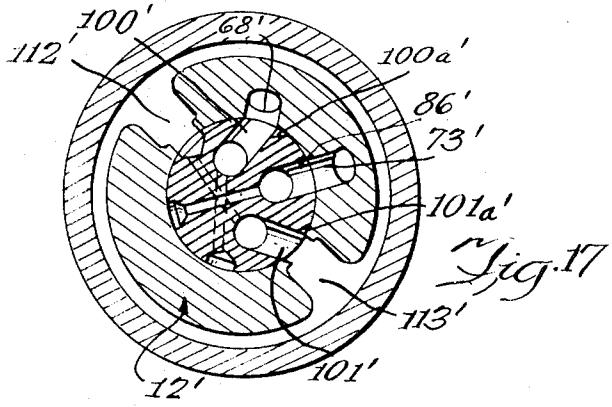


Fig. 17

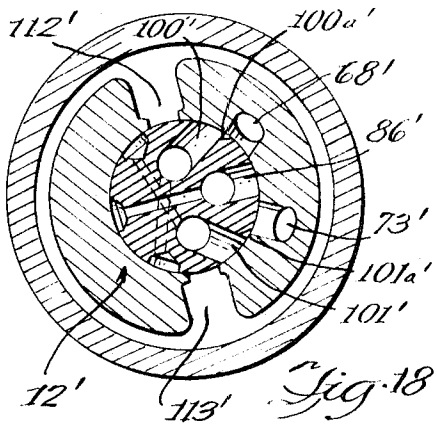


Fig. 18

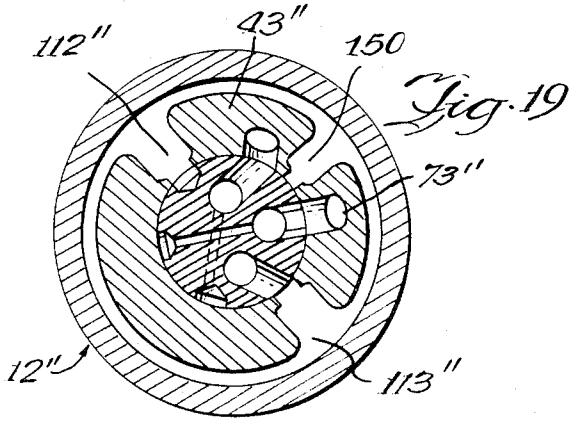


Fig. 19

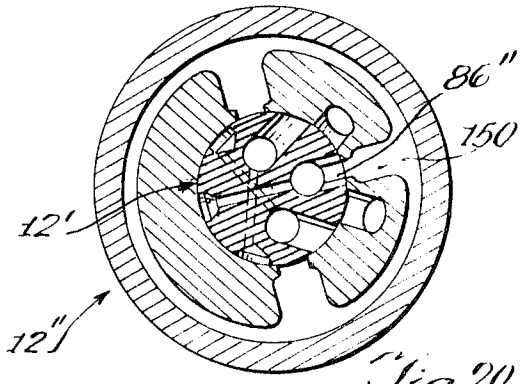


Fig. 20

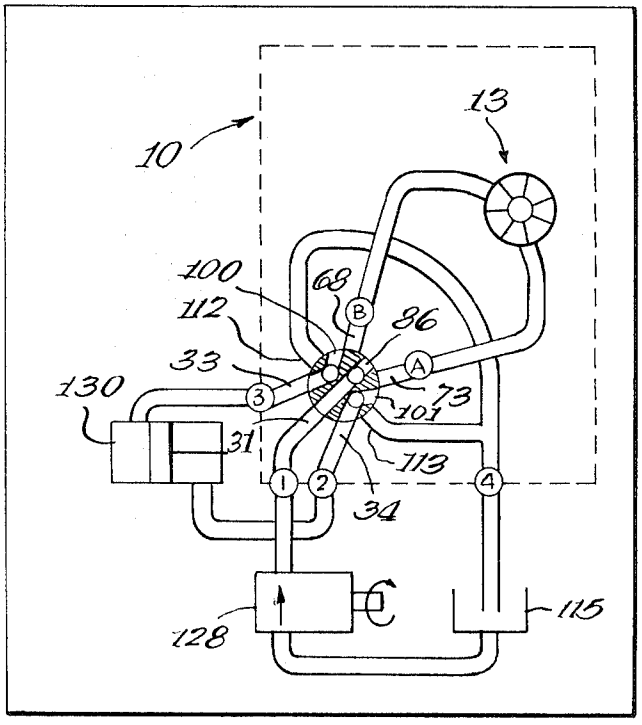


Fig. 16

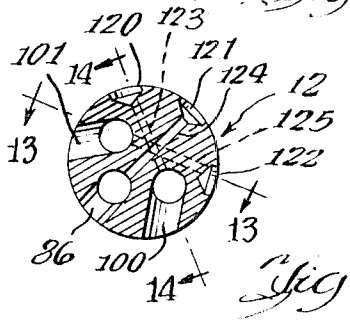


Fig. 12

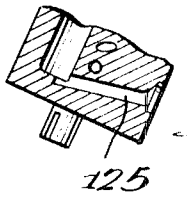


Fig. 13

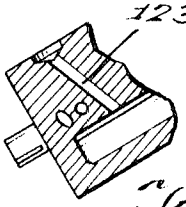


Fig. 10

PROPORTIONAL DIRECTIONAL CONTROL UNIT

BACKGROUND OF THE PRESENT INVENTION

The present invention relates to a self-contained hydraulic device which controls the delivery of high-pressure fluid in an amount proportional to a manual input. Such a device is suitable for use in hydraulic circuits such a power steering controls on heavy machinery in which a given rotation of a steering wheel or other manual operator will produce a supply of fluid to a hydraulic actuator or cylinder for steering, for example, the ground wheels. Such devices can be used for controlling flow in any hydraulic circuit in which precise adjustment in the quantity of high pressure fluid to be delivered is required.

In the past various hydraulic controls have been provided for supplying a predetermined quantity of hydraulic fluid to loads such as actuators and steering systems. One such hydraulic device is a pump in combination with a manual on-off valve. With such simple valves operator judgment and experience are required to maintain effective modulation of fluid pressure and flow. In addition an on-off valve requires two manual movements, i.e. one to open the valve in a first direction and one to close the valve in a second direction, to supply a predetermined quantity of fluid.

Another type of hydraulic circuit for supplying a predetermined quantity of fluid to systems such as steering controls is a hydraulic supply circuit which has feedbacks to terminate the supply of fluid when a predetermined movement of the associated actuator has been achieved. Hydraulic feedback systems have the disadvantage, however, of requiring significant amount of torques to operate, particularly for high displacement units. Systems having a similar purpose which operate on a pressure-differential-type control are susceptible to erratic operation from large viscosity changes such as might be encountered in cold weather startup. Also pressure differential controls are not readily adaptable to nonload reaction systems which limit their potential usefulness.

It has also been suggested in the past that actuator control may be achieved through a servo device which supplies a predetermined quantity of fluid in accordance with the extent of movement of a manually controlled member. This is the general character of the present device. However, devices that have been provided in the past for this purpose have included gear elements which have been extremely expensive to manufacture correctly and valves which also require a substantial amount of precision machining. For these reasons these prior units are susceptible to efficiency declines caused by valve component wear because of the many sealing surfaces required. Moreover, high rates of fluid flow are difficult to obtain in these units while still maintaining a small total envelope therefor.

It is, therefore, a primary object of the present invention to overcome difficulties in prior fluid circuit systems through the provision of a fluid pressure proportioner which may be easily controlled to prevent the instantaneous opening of full fluid flow at full pressure to prevent abrupt changes in movement of the associated actuator and to minimize hydraulic shock loads in the lines and components, as well as to completely hydraulically balance the associated servo valve and to provide a circuit that may be either open or closed centered, or either load reactive or nonload reactive as desired.

SUMMARY OF THE PRESENT INVENTION

In accordance with the present invention a self-contained hydraulic device is provided which controls delivery of high-pressure fluid to a load in an amount proportional to manual input. The hydraulic device is particularly suitable for hydraulic circuits such as power steering systems of heavy machinery in which a given rotation of the steering wheel will provide a predetermined supply of hydraulic fluid under pressure to a steering actuator.

The device operates functionally by responding to a predetermined amount of manual rotation of an input, plus hydraulic fluid under pressure, to deliver a definite quantity of hydraulic power which is proportional to and directionally oriented with respect to the input rotation. This is accomplished generally by a valve, a portion of which is an integral part of a separate fluid displacing or metering servo device that is hydraulically self-restoring to the off or neutral position.

The servo device according to the present invention takes the form of a manually positionable servo valve having a supply port and two adjacent load-connected ports. The supply port is connected to receive fluid under pressure from a pump and load ports are connected respectively to the opposite sides of an actuator, such as a steering cylinder. The servo valve is rotatable within a hydraulic servomotor or metering device that has first and second followup-type ports that selectively communicate with the supply and load ports in the servo valve. The metering device takes the form of a stationary ring gear with a main carrier mounted for rotation within the ring gear and has a plurality of rotatable planet gears mounted within the carrier spaced from the central axis thereof and engaging the teeth on the ring gear. The tooth spaces between the planet gears and the ring gear define the expanding and contracting fluid chambers that when pressurized effect rotation of the carrier and the ports associated with the carrier.

As the servo valve is rotated in one direction by the operator fluid under pressure from the supply port will be delivered to one of the carrier ports driving the servo device in one direction of rotation with fluid from the carrier being delivered through the other carrier port, back through the servo valve in one of the load ports and through suitable conduits to one side of an associated hydraulic actuator. The other side of the hydraulic actuator returns fluid through the other load port in the servo valve to a suitable reservoir.

The servo valve is completely hydraulically balanced through the provision of recesses or pockets diametrically opposed to each of the ports in the servo valve which are subjected to fluid under pressure whenever the associated or opposite port is pressurized and in this manner the servo valve is hydraulically balanced within its associated valve for minimizing wear on the servo valve.

Moreover, in accordance with the present invention the servo valve may be nonload reactive by providing lands of sufficient length so that when the servo valve is in a neutral position the load ports in the servo valve will be blocked by lands within the associated valve bore. In one embodiment of the present invention a load-reactive servo valve is shown in which the servo device reacts to loads applied to the associated actuator through the appropriate location of the load ports in the servo valve so that they partially communicate with the first and second ports in carrier of the metering device. In this manner the operator may resist any external load applied to the actuator by maintaining the position of the servo valve so that any movement of the actuator will cause a correcting flow of fluid from the metering portion of the hydraulic device to return the actuator to its desired position.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is an elevation view of the present proportional directional control fluid control unit;

FIG. 2 is an enlarged longitudinal cross section of the hydraulic device shown in FIG. 1;

FIG. 3 is a reduced section taken generally along line 3–3 of FIG. 2 illustrating a portion of the servo motor in the hydraulic unit;

FIG. 4 is a cross section taken generally along line 4–4 of FIG. 1 showing the ports associated with the servomotor;

FIG. 5 is a section taken generally along line 5–5 of FIG. 4 showing the gallery passages in the gear carrier associated with the servomotor;

FIG. 6 is a cross section taken generally along line 6–6 of FIG. 5 illustrating an arcuate gallery passage associated with one set of the ports associated with the hydraulic servomotor;

FIG. 7 is a cross section taken generally along line 7–7 of FIG. 5 illustrating the other arcuate gallery passage associated with the other set of ports in the servomotor;

FIG. 8 is another cross section of the carrier associated with the servomotor illustrating one of the two ports associated with the servomotor along with a tank slot;

FIG. 9 is another cross section of the carrier member shown in FIGS. 5 and 8 illustrating both of the reservoir slots;

FIG. 10 is a subassembly view of the present servo valve;

FIG. 11 is a cross section of the servo valve taken generally along line 11–11 of FIG. 2;

FIG. 12 is a cross section of the servo valve taken generally along line 12–12 of FIG. 11;

FIG. 13 is a fragmentary section of the servo valve taken generally along line 13–13 of FIG. 12;

FIG. 14 is a fragmentary section of the servo valve taken generally along line 14–14 of FIG. 12;

FIG. 15 is a cross section of the interconnection between the servo valve and the servomotor taken generally along line 15–15 of FIG. 2;

FIG. 16 is a schematic diagram of the hydraulic proportioner of FIGS. 1 to 15 as shown in circuit with a pump, actuator and reservoir;

FIGS. 17 and 18 are cross-sectional views of a modified form of the present servo valve that is load reactive; and

FIGS. 19 and 20 are cross sections of still another modified form of the present servo valve that includes provision for free flow from the pump when the valve is in a neutral position.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENTS

As best seen in FIGS. 1 to 3, the present hydraulic proportioner 10 is seen to consist generally of a manually rotatable input shaft 11, rotatable servo valve 12 driven thereby, and a rotary hydraulic servo motor or metering device 13.

Generally opposed rectangular housing members 15 and 16 are provided which sandwich a stationary internally toothed ring gear 20 shown more clearly in FIG. 3. The housing members 15 and 16 as well as the ring gear 20 are fastened together by suitable fasteners 22 shown in FIG. 3. Closing the left end of the housing 15 is an annular end plate 24 fixed to the housing member by suitable fasteners 25 and having an axial flange 26 fixed to sleeve 27 surrounding the manually operable input shaft 11.

Housing member 15 has disposed on the reduced portion 28 bosses 29 and 30. Boss 29 has an inlet port 31 which is adapted to be connected to a source of fluid under pressure such as a pump. In this regard it should be understood that the present device does not technically act as either a pump or a motor in its normal mode of operation. Its primary purpose is to receive fluid under pressure from a source of fluid pressure and to deliver a predetermined quantity of that fluid to an actuator. There is intended to be only a small pressure drop across the hydraulic device 10 during its normal operation so that it may not properly be considered a motor in the generally accepted sense of the term.

Boss 29 on the housing has a second port 33 which is connected to one side of an actuator (not shown in FIGS. 1 to 3) which may take the form of a reciprocating piston cylinder connected to steer a vehicle. Still another port 34 in boss 30 is adapted to be connected through suitable conduits to the other side of the actuator for conveying hydraulic fluid relative thereto. A fourth port 35 in boss 30 is adapted to be connected to a reservoir from which the pump connected to port 31 draws fluid.

The servomotor 13 consists generally of an annular gear carrier assembly 38 rotatably supported in the housing. Carrier assembly 38 includes a left-end cap member 43 and a right-end cap member 44. Disposed between the end members 43 and 44 is a spacer 45 also shown in FIG. 3. The end members 43 and 44 and the spacer 45 are fixed for rotation together as a unit by suitable threaded fasteners 48.

The spacer 45 has four arcuate recesses 50 in the periphery thereof spaced 90° apart which define pockets between the end members 43 and 44 that receive planet gears 53, there being provided four planet gears 53 within the present hydraulic device, although a greater or lesser number of planets may be provided. Plate 45 also has convex arcuate surfaces 46 (FIG. 3) which have a close sealing clearance with the tips of the internal gear 20 and which also serve to support the servo device 13 within the housing members 15 and 16.

Each of the planet gears 53 has teeth interengaging the ring gear 20 and each is rotatably supported in the carrier assembly 38 by a shaft 55 which is rotatably supported at one end in end cap 43 by bearing 56 and the other end in end portion 44 by bearing 58. The gears 53 have a width slightly less than the width of spacer 45 to provide a running clearance for the gears 53 within the associated pockets within the gear carrier assembly. As the gears 53 rotate with the carrier, the interengaging teeth define expanding and contracting fluid chambers with the ring gear and the present device operates by porting fluid to these expanding fluid chambers causing rotation of the gears 53 driving the gear carrier 38 in rotation.

For the purpose of porting fluid to and from the expanding and contracting tooth spaces defined by the interengaging gears 53 and 20, the end member 43 of the carrier assembly is provided with two kidney-shaped ports 60 and 61 associated with each of the gears 53. The spacer 45, gears 20 and 53, along with the sides of end members 43 and 44 seal ports 60 from ports 61 so that either may act as a high- or low-pressure port and thus the servo or metering device 13 is reversible and fluid flow may, under pressure, be delivered to either port 33 or 34 shown in FIG. 1.

Shown in FIG. 5 the ports 60 are connected together through axial passages 63 in end member 43 which communicate with a semiannular gallery passage 64. Annular passage 64 communicates with a valve bore 66 defined centrally in end member 43 through a single port 68 shown clearly in FIG. 2.

Similarly, the ports 61 (FIG. 5) communicate with each other through axial passages 69 in end member 43 and an annular gallery passage 71. Gallery passage 71 communicates with the valve bore 66 through a diagonal passage or port 73 shown clearly in FIG. 8. Actually, port 73 is angularly adjacent to and in the same transverse plane as port 68. The relative location of port 68 and 73 may be more clearly seen in FIG. 16.

The servo valve 12 performs the function of supplying fluid under pressure to one of the ports 68 and 73, and hence the servo device 13, and for returning fluid under pressure from the servo device 13 to one of the actuator connected ports 33, 34 until the carrier assembly 38 passes through a given angle of movement following movement of the valve 12. Servo device 13 continues to rotate until the valve 12 is manually stopped at which time the carrier assembly 38 will cease rotation and interrupt flow through one of the actuator connected ports 33, 34.

The control valve 12 is generally cylindrical in construction as shown in FIGS. 2, 10, 11, 12, 13, 14 and 15, and is rotatably mounted within the valve bore 66 in end member 43.

As shown in FIGS. 2 and 15 means are provided for permitting approximately 30° of rotation between the valve member 12 and the carrier 38 in either direction from the neutral position of the elements shown in FIG. 15. More specifically, the end of the valve bore has diametrically opposed axial pins 81 and 82 which will be engaged by an arcuate projection 83 extending from the end of the valve 12 and positioned to interfere with the pins 81 and 82 upon rotation of the valve member 12 relative to bore 66. In the neutral position shown in FIG. 15 it is apparent that the projection 83 along with the valve member 12 may be rotated in either direction of rotation approximately 30° before engaging the stops 81 and 82. Further movement of the control valve 12 in that direction will cause the manual driving of the carrier 38. Thus, upon a failure of the pump associated with the hydraulic unit 10 the operator may manually rotate the shaft 13 causing manual rotation of the servomotor 13 through the projection

83 which causes the servomotor 13 to act as a pump delivering fluid to the associated actuator.

For the purpose of resiliently urging the control valve 12 to a neutral position with respect to the carrier assembly 38 and the ports 68 and 73, a central projection 85 extends from the end of the control valve 12. Through this projection extends a spring 87 which is fixed to the pins 81 and 82 at its ends. Spring 87 does not prevent relative movement between the control valve 12 and the carrier assembly 38 but urges the elements to their relative neutral position shown in FIGS. 15 and 16. This is useful since the followup action of servomotor 13 may stop somewhat short of its neutral position with respect to the servo valve 12 because of the relative location between the ports in the servo valve 12 and the carrier 38. In the neutral position of the embodiment shown in FIGS. 1 to 16 both ports 68 and 73 are blocked by the lands of control valve 12 stopping rotation of the carrier 38 and flow through one of the ports 33, 34.

A keyhole-shaped high-pressure supply port 86 FIGS. 2 and 10) is provided in the valve member 12 having a circular porting portion or portions 88 and 89. An axially extending portion 90 is provided for the purpose of effecting freer communication with the return passage when open center valving is provided as shown in the embodiment of FIGS. 19 and 20.

For the purpose of supplying high-pressure fluid to the high-pressure port 68 from port 31 an axially extending passage 94 is provided in the valve member 12 shown in FIG. 2. Passage 94 has a radially extending portion 95 which communicates with a peripheral slot 97 shown also in FIG. 10, which in turn is in communication with the high-pressure inlet recess 36 in housing member 15. Recess 36 continuously communicates with port 31.

For the purpose of supplying hydraulic fluid to and from the servomotor 13 relative to an actuator associated with ports 33, 34, load ports 100 and 101 of generally circular configuration are provided on the opposite sides of and transversely coplanar with the main supply port 86 in valve member 12. Port 100 continuously communicates with main actuator port 33 through an axially extending passage 103 and a slot 104 in valve member 12 which in turn communicates with an annular recess 106 in housing member 15 (FIG. 2) that is connected with port 33.

Similarly, port 101 communicates continuously with actuator port 34 through axial passage 109 and slot 110 in valve member 12, which continuously communicates with a recess 108 in housing member 15 (FIG. 2) that in turn communicates with port 34.

The three valve ports 86, 100 and 101 are separated by an equal angular amount with supply port 86 being located between the two others. The servomotor ports 68 and 73 in bore 66 are positioned such that port 68 is midway between supply port 86 and load port 100 and port 73 is midway between supply port 86 and load port 101 when the valve spool 12 is in its neutral position shown in FIG. 16.

Two slots 112 and 113 (see FIGS. 2, 8, 9 and 16) extend radially outward from the circular bore 66 of the servo device 13 to connect with the annular space 114 between the carrier 38 and the housing members 15, 16. These slots are positioned such that load passage 100 is midway between slot 112 and passage 68 and load passage 101 is midway between slot 113 and port 73 when the servo valve 12 is in the neutral position.

The interior of the housing communicates directly with return port 35 so that fluid flows therefrom to a suitable reservoir 115 shown in FIG. 16. Slots 112 and 113 serve the purpose of conveying fluid from the retracting side of the associated actuator that passes into the device through one of the ports 33, 34, to the reservoir 115.

The control valve 12 is pressure balanced in the valve bore 66 by recesses 120, 121 and 122 disposed coplanar with and diametrically opposite the ports 100, 86 and 101, respectively, as shown in FIG. 12. The recesses 120, 121 and 122 are equal in area to the ports 100, 86 and 101, respectively, and are supplied high-pressure fluid through generally radially extending passages 123, 124 and 125 which communicate with the opposed ports. Passages 123, and 125 are shown more clearly in FIGS. 14 and 13, respectively, and are seen to be somewhat diagonal so that none of the passages 123, 124, 125 intersect each other. Thus, equal and opposite hydraulic reactions are imposed on the control valve 12 within the bore 66 hydraulically balancing the valve member.

Generally, each of the ports in the valve member 12 and the ports in the end member 43 are circumferentially separated from each other by small seal lands. To achieve various types of control certain modifications shown in FIGS. 17 to 20 will be described hereinbelow.

For the purpose of permitting fluid recirculation from space 114 to the supply passage 94 when using the servo device 13 as an emergency pump a nonreturn check valve assembly 126 is provided in the valve spool 12 as shown in FIG. 2. Check valve 126 prevents flow, however, from the supply passage 94 to the space 114 when pressure is higher in supply passage 94 which is generally the case.

While the operation of the embodiment of FIGS. 1 to 15 is believed apparent from the above description, a summary of the operation of the device will be described as follows with respect to the schematic illustration shown in FIG. 16. Assuming a pump 128 is supplying fluid under pressure to port 31 and that a reservoir 115 is connected to port 35, ports 33 and 34 are connected to a hydraulic actuator 130 such as a double-acting cylinder. It is further assumed that when port 34 is pressurized that some action will result which may be defined as left handed and similarly when port 33 is pressurized a right-handed function will occur.

If by rotation of the valve 12 either port 68 or port 73 is connected to the source of fluid pressure and the other port was connected to the hydraulic load 130, equal pressure would be developed in each line except for a very small loss caused by the friction in the servo device 13. Assuming the hydraulic pressure is sufficient to overcome the load imposed on actuator 130, fluid is displaced through the servo device 13 causing it to rotate. Fluid then flows out the opposite passage at the same pressure it enters, or substantially the same pressure. If the supply pressure and the hydraulic load are connected to the reverse openings or ports 68, 73 flow will occur in the opposite direction causing the servo device 13 to rotate in a reverse direction.

When the valve member 12 is in the neutral position shown in FIG. 16 all the ports are blocked and the high-pressure fluid entering port 31 and flowing to supply port 86 in the valve is blocked by the circular bore 66 of the servo device. Hydraulic fluid under pressure also passes into the pressure pocket 121 which is radially opposed to port 86 hydraulically balancing the spool.

At the same time, ports 100 and 101 are blocked by the circular bore so that any load which might be present at the actuator 130 will be resisted by an adequate amount of pressure in the lines connected to ports 33 and 34. Moreover, the servo device cannot move because ports 68 and 73 are blocked by the spool 12. For these reasons the hydraulic circuit described with respect to FIGS. 1 to 16 is closed-center, nonload reaction.

Assuming that the spool 12 is manually rotated in a clockwise direction a small amount as viewed in FIG. 16 so that supply port 86 becomes connected with port 73, port 68 will at the same time communicate with load port 100. High-pressure fluid will thus drive the servo device 13 in a clockwise direction so that servo valve member 43 follows manual valve member 12, and at the same time pressurizing port 68 and delivering fluid under pressure through load port 100, out main port 33 to the left side of actuator 130 driving it to the right. Since port 100 is pressurized at this time the pressure balancing recess 120 is also pressurized maintaining the spool in hydraulic balance.

As the servo device 13 rotates, the right-handed movement of actuator 130 will continue and fluid from the low pressure side of the actuator will be returned through port 34 in hous-

ing 15, port 101 in valve member 12, slot 113 in end cap 43, to the reservoir through port 35 in the housing.

If movement of the control input member 11 is stopped, the rotation of the servo device 13 eventually causes servo member 43 to catch manual member 12 and all of the ports to return to their original neutral position and to thereby cease displacement of hydraulic fluid out port 33. However, the actual position of the manual control 11 of servo device 13 in the hydraulic actuator has changed by some specified amount. It can be thus understood that further manual rotation of the input member 11 will cause a further rotation of the servo device and more fluid will be delivered under pressure to the actuator 130.

Conversely, rotation of the spool 12 in the opposite direction will cause a quantity of high-pressure fluid to be supplied to port 34 to the right side of the actuator 130 to effect a left-handed action. At this time low pressure fluid from the left side of the actuator will be returned to port 33 and the reservoir through port 100 and slot 112.

Since there are small seal lands between the various ports in the servo valve 12 and the ports in the carrier member 38, the seal lands will prevent the total rotation of the servo device 13 necessary to completely return the servo device 13 to a neutral position with respect to the servo valve 12. The leaf spring 87 shown in FIG. 15 provides for a mechanical restoration of the small final distance to the neutral position.

The above described circuit is a nonload reaction system since loads imposed on the actuator 130 have no effect on the operation of the hydraulic device 10. In accordance with the servo valve shown in FIGS. 17 and 18, the hydraulic circuit of FIGS. 1 to 16 may be rendered a load-reaction system in that a load externally imposed is "felt" by the hydraulic unit 10 thereby achieving a "feedback" to the operator. As seen in FIGS. 17 and 18 the provision of port extensions 100*a'* and 101*a'* render the hydraulic device 10 a load-reaction device, if desired. The port extensions 100*a'* and 101*a'* are of sufficient length so that when the valve 12' is in the neutral position shown in FIG. 18 the ports 100' and 101' communicate with the servomotor ports 68** and 73', respectively. This effectively eliminates the neutral seal between these ports. Since ports 100' and 101' are connected directly with both sides of the associated actuator, and openings 68' and 73' are the inlet-outlet for the servo device 13, if the actuator is displaced by an external load the servo device 13 will be driven. The servo device 13 is mechanically connected with the mechanical control through the leaf spring 87 providing operator "feel."

If no resistance to movement of the input member 11 is effected by the operator any external load on the actuator will cause fluid to flow into one of the ports 33, 34 through the servo device 13 and back into the opposite side of the actuator. The leaf spring will maintain proper alignment between the spool 12 and the servo device 13 during this rotation and at the same time cause corresponding rotation of the input shaft 11. The speed of rotation, however, is restricted because of the small fluid passages which exist between the various ports. The operation of the pump has no effect at this time since port 86' continues to be blocked.

Under the same conditions, however, if the operator provides resistance to the rotation of the input shaft 12 caused by the "feedback" action, a slight movement of the actuator will cause the servo device 13 to become hydraulically disconnected with the actuator and connected with the pump through port 86', which results from relative rotation between the cap 43 and spool member 12. Fluid is thus ported to the actuator in a direction to resist movement by the external load.

A further modification of the servo valve is shown in FIGS. 19 and 20. In this embodiment a valve 12'' is provided identical in size and construction to the valve 12 except that it is of an open center type through the provision of a slot 150 disposed midway between the slots 112'' and 113''. With the valve member 12'' in the neutral position shown in FIG. 20 the main supply port 86'' communicates freely with the slot 150. In this manner fluid from the pump will be returned directly to the reservoir when the valve 12'' is in the neutral position. With this type of circuit variable volume pumps are not necessary and fixed volume pumps may be employed since they should be allowed to pump continuously at low pressure. Through the provision of the open center valving this is possible and the expenditure of power under no-load conditions is held to a minimum.

It should be noted that the slot 150 should be positioned so that when the valve is rotated from its neutral position to an operative position such as shown in FIG. 19 slot 150 will be sealed by the lands on the valve member 12''.

I claim:

1. A hydraulic servo device for proportioning fluid under pressure, comprising: manually operably means movable to positions representing the desired quantity of fluid to be delivered, servo valve means positioned by said manually operable means, said servo valve means having supply port means, first load port means and second load port means, a relatively stationary internally toothed outer gear member, a carrier member rotatable about an axis, at least one planet gear rotatably supported on said carrier having an axis spaced from the axis of said carrier member, said planet gear having external teeth interengaging the internal teeth on said outer gear and defining therewith expanding and contracting tooth spaces upon relative rotation of said gears, first port means in said carrier member continuously communicating with one of said spaces, and second port means in said carrier member continuously communicating with the other of said spaces, said supply port means being selectively connectable with said first and second port means in said carrier member so that as the servo valve means ports fluid to one of the carrier port means the carrier will follow movement of the servo valve means, said first load port means being selectively connectable with one of said port means in said carrier member, said second load port means being selectively connectable with the other port means in said carrier member.

2. A hydraulic servo device for proportioning fluid under pressure, comprising: manually operable means movable to positions representing the desired quantity of fluid to be delivered, servo valve means positioned by said manually operable means, said servo valve means having supply port means, first load port means and second load port means, a relatively stationary internally toothed outer gear member, a carrier member rotatable about an axis, at least one planet gear rotatably supported on said carrier having an axis spaced from the axis of said carrier member, said planet gear having external teeth interengaging the internal teeth on said outer gear and defining therewith expanding and contracting tooth spaces upon relative rotation of said gears, first port means in said carrier member continuously communicating with one of said spaces, and second port means in said carrier member continuously communicating with the other of said supply port means being selectively connectable with said first and second port means in said carrier member so that as the servo valve means ports fluid to one of the carrier port means the carrier will follow movement of the servo valve means, said first load port means being selectively connectable with one of said port means in said carrier member, said second load port means being selectively connectable with the other port means in said carrier member, said carrier member having a centrally disposed bore therein, said bore having a first port opening communicating with said first port means and a second port opening communicating with said second port means, said servo valve means including a cylindrical member rotatably supported for limited rotating movement in said bore, said supply port means and said first and second load port means being defined in said cylindrical member and being selectively communicable with said first and second openings.

3. A hydraulic servo device as defined in claim 2, wherein said supply port means and said first and second load port

means lie in a plane transverse to said cylindrical servo valve member.

4. A hydraulic servo device as defined in claim 1, including a plurality of gears rotatably supported on said carrier member and having external teeth interengaging the teeth on said outer gear.

5. A hydraulic servo device as defined in claim 1, wherein said first and second load port means are adapted to be connected to drive a load in either direction of movement, said servo valve member being movable to a neutral position where said supply port means is out of communication with said first and second port means in said carrier member, said first and second load port means in said valve means being positioned so that they at least partially communicate with said first and second port means in the carrier member when the valve means is in the neutral position.

6. A hydraulic servo device as defined in claim 1, wherein each of said supply port means, said first load port means and said second load port means has a port opening at the periphery of the valve means, said valve means including a cylindrical valve member, and means for balancing the force of fluid pressure on the valve member in each of said port openings.

7. A hydraulic servo device as defined in claim 6, wherein said balancing means includes a pocket in the periphery of the valve member opposite each of said port openings, and means providing communication between the port openings and the oppositely disposed pocket.

8. A fluid-proportioning device, comprising: manually operable means movable to positions representing the desired quantity of fluid to be delivered, servo valve means positioned by said manually operable means, said servo valve means having supply port means, first load port means and second load port means, a servo device supplied fluid by said servo valve including first and second fluid pressure chambers, said servo device having first and second ports when pressurized causing rotation of said servo device, said supply port means being selectively connectable with said servo device first and second port means so that as the servo valve means ports fluid to one of the carrier port means the carrier will follow movement of the servo valve means, said servo device first and second port means also being selectively connectable with the first and second load port means, said servo valve being movable to a neutral position when said supply port means is out of communication with said first and second port means of the servo device, said first and second load port means having at least partial communication with said first and second servo device port means in said neutral position whereby the servo device is responsive to load variations when the servo valve means is in the neutral position.

9. A fluid-proportioning device as defined in claim 8, including a pump for supplying fluid under pressure to said supply port means, a fluid actuator, first conduit means connecting one of said load port means with one side of said actuator, second conduit means connecting the other of said load port means to the other side of said actuator.

10. A hydraulic servo device as defined in claim 1, wherein said carrier member has a return port between said first and second port means, said supply port being communicable with said return port when the servo valve means is in a neutral position.

11. A fluid-proportioning device, comprising: manually operable means movable to positions representing the desired quantity of fluid to be delivered, servo valve means positioned by said manually operable means, said servo valve means having supply port means, first load port means and second load port means, a servo device supplied fluid by said servo valve including first and second fluid pressure chambers, said servo device having first and second ports when pressurized causing rotation of said servo device, said supply port means being selectively connectable with said servo device first and second port means so that as the servo valve means ports fluid to one of the carrier port means the carrier will follow movement of the servo valve means, said servo device first and second port means also being selectively connectable with the first and second load port means, said servo valve being movable to a neutral position when said supply port means is out of communication with said first and second port means of the servo device, return port means in said servo device between said first and second ports, said supply port means communicating with said return port means when the servo valve means is in a neutral position so that fluid under pressure in said supply port means may flow through said return port means.