Feb. 24, 1959 H. A. VANDER KAAY 2,874,682

POSITION-RESPONSIVE SERVO-CONTROL UNIT

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Fig. 1

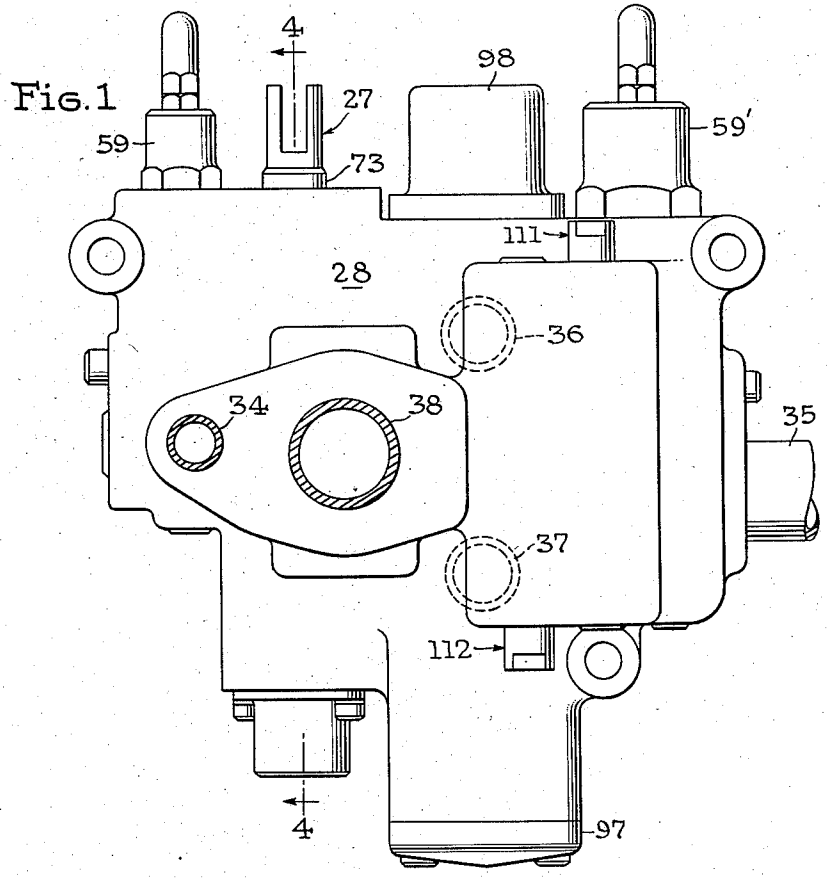
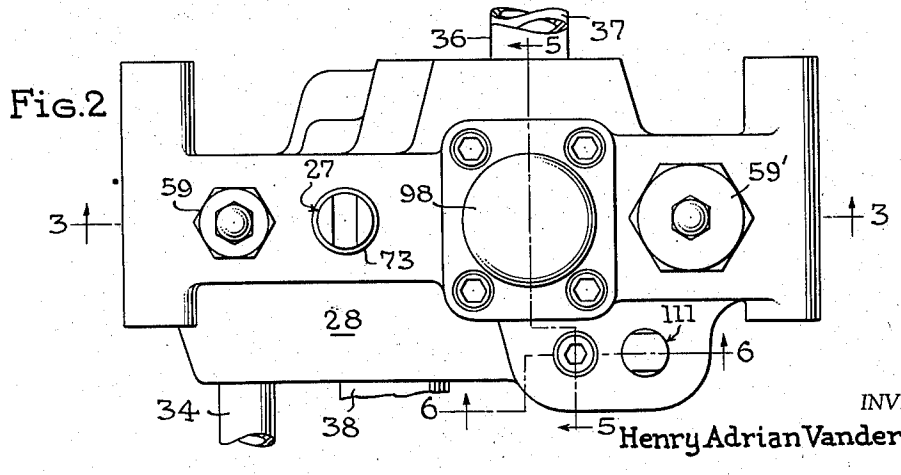


Fig. 2



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BY Dodge and Jones

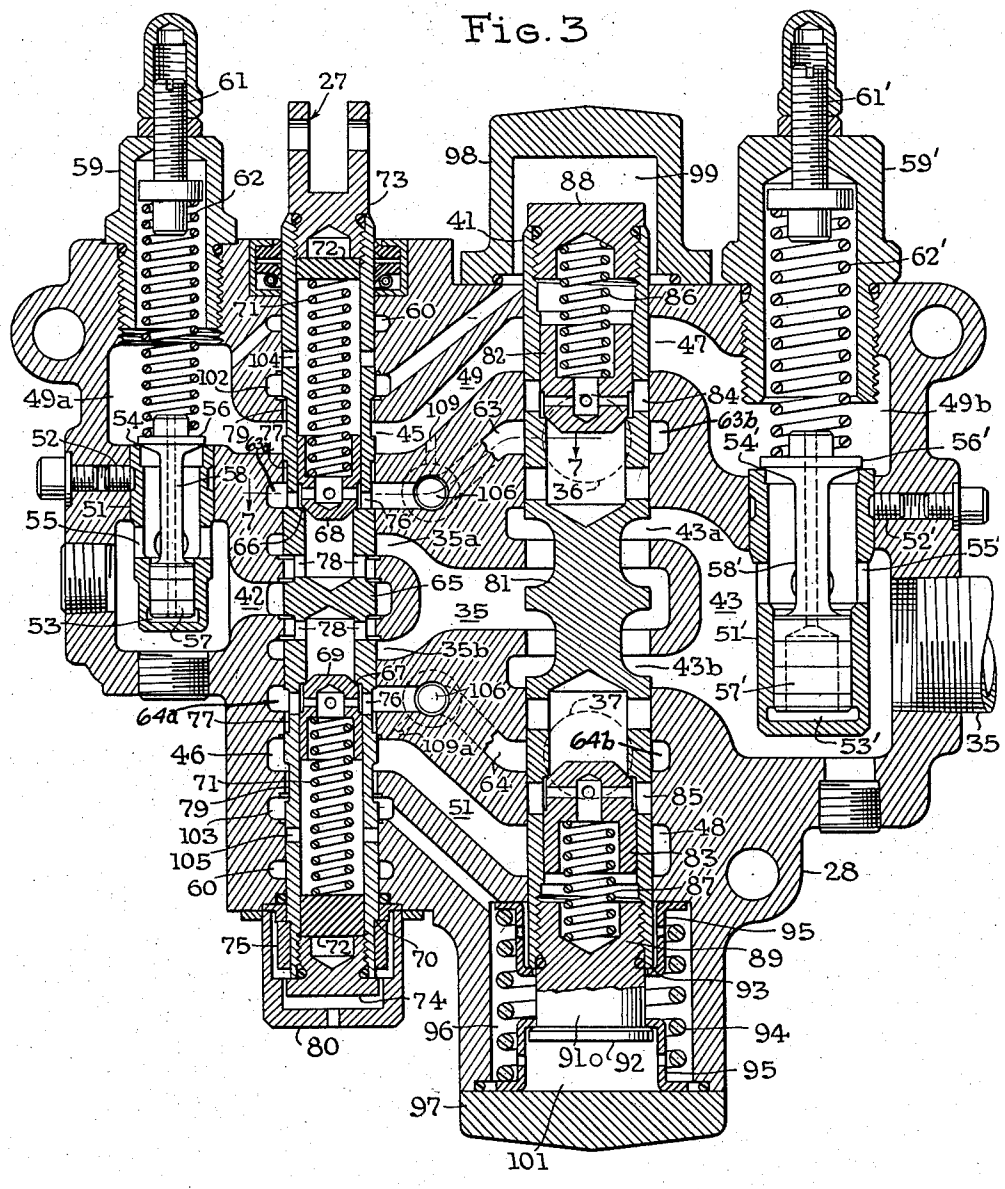
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Fig. 3



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Fig.4

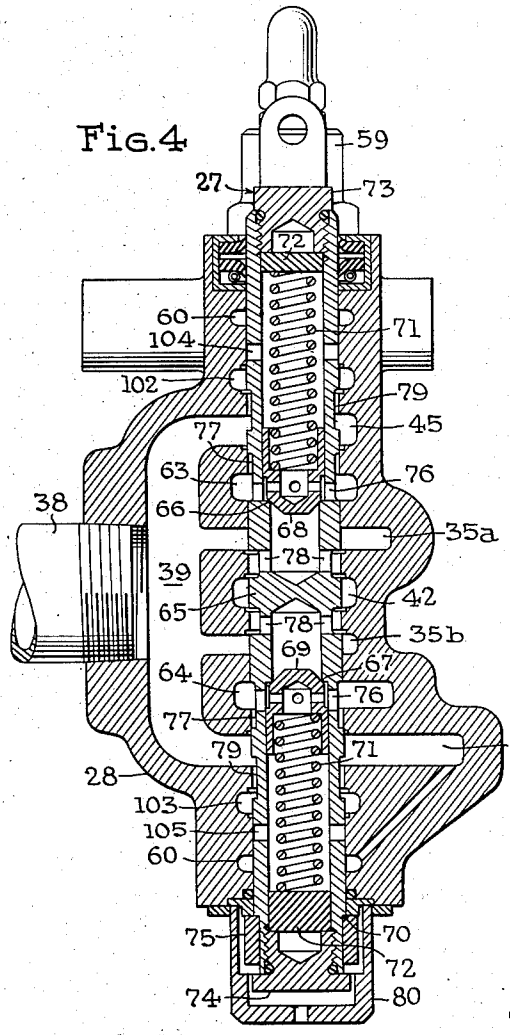


Fig.6

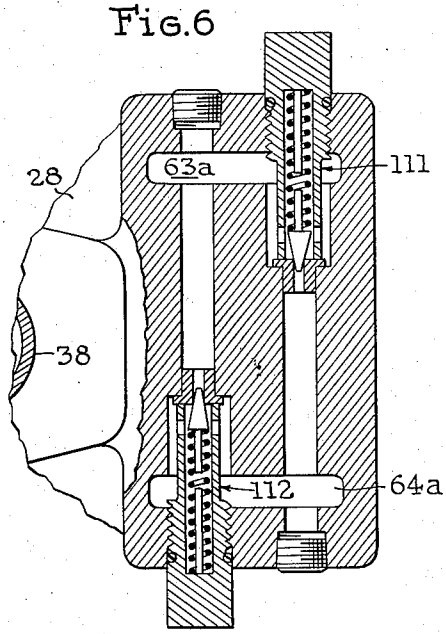


Fig.7

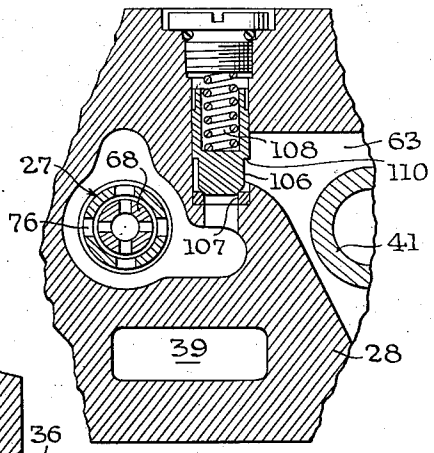
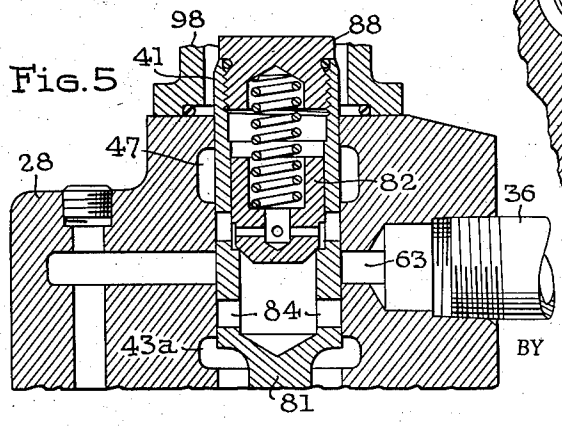


Fig.5



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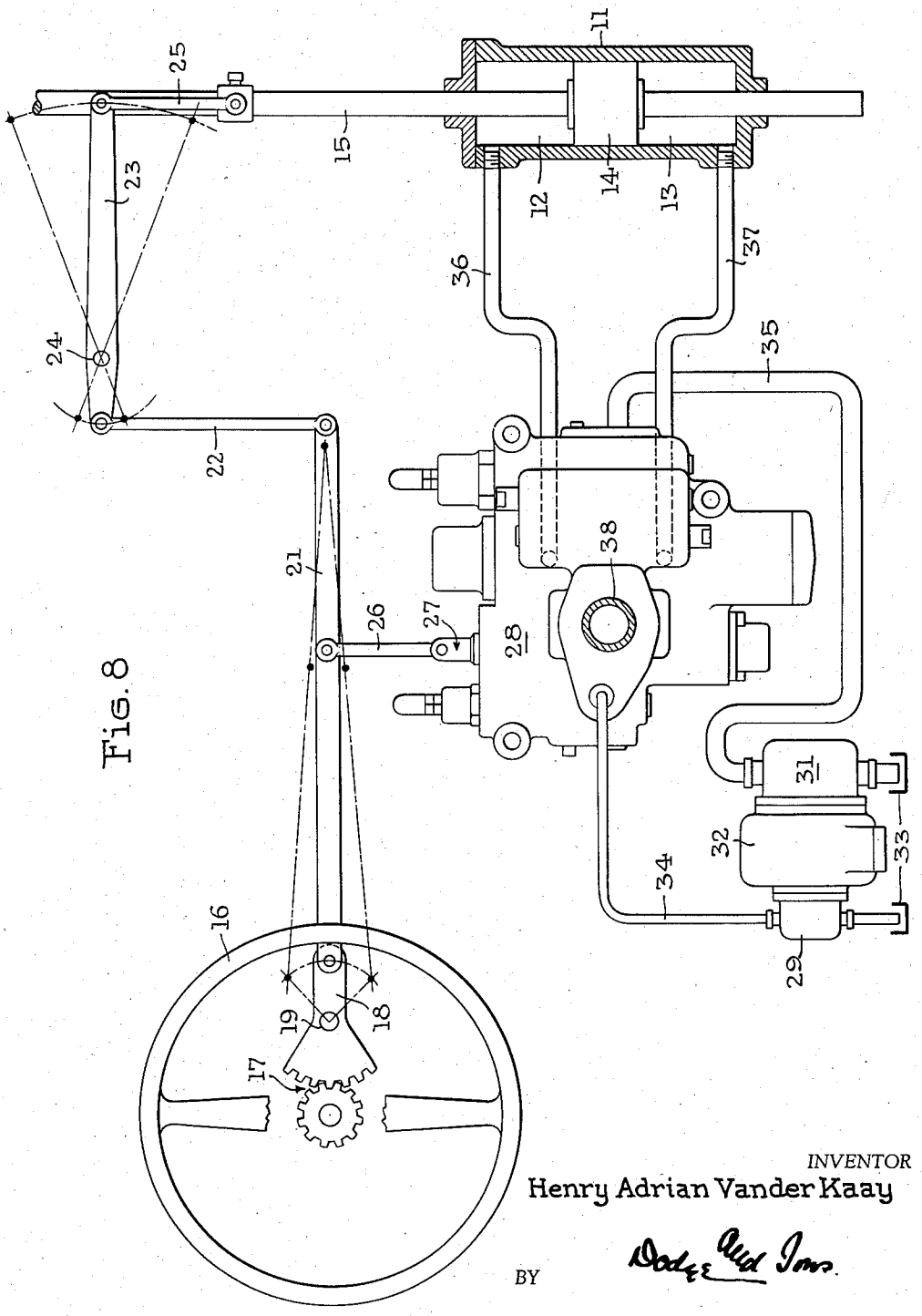


Fig. 8

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United States Patent Office 2,874,682
Patented Feb. 24, 1959

2,874,682

POSITION-RESPONSIVE SERVO-CONTROL UNIT

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Application September 18, 1957, Serial No. 684,745

8 Claims. (Cl. 121—41)

This invention relates to fluid-pressure motors and particularly to servo-motors operated by hydraulic liquid. A common use for such motors is to operate maneuvering gear on vehicles. This may take various forms such as an elevating gear, a steering gear, loading and unloading devices to name a few. Such motors usually are double-acting.

Variable displacement pumps being expensive, heavy and difficult to maintain, and since accumulators are objectionable for this class of service, reliance is often placed on a constant delivery pump supplying an open-center distributing valve. In such a system it is possible to use an additional unloader local to the pump. This minimizes heating at no load, but even so, graduated control depends on throttling of the full output of the pump which wastes energy and results in heating. Graduated control of the output of a large pump is unsatisfactory because the controlled device tends to overtravel.

Another scheme, long used extensively in the hydraulic art, operates two pumps in parallel and includes means to unload both or only one, or neither. The pump or pumps, when not unloaded, are operated at full load. The pumps are commonly of different capacity and usually are run at equal speeds. Where wide range progressive speed graduation of the hydraulic motor is not vital, such a scheme is satisfactory and economical, for throttling is not the basis of control.

The present invention takes advantage of the economy permitted by using two pumps, and adds graduation of speed secured by throttling. The two pumps are of widely different capacity, and flow from the small-capacity pump is controlled directly by a maneuvering valve, preferably of the open-center type. This permits accurate graduation effected by throttling a small flow. The energy losses are small. The larger pump is initially completely unloaded. Its delivery is controlled by another valve, preferably of the open-center type and similar in character to the maneuvering valve, and piloted by the maneuvering valve in such a way that it parallels the flow controlling functions of the maneuvering valve but at a larger flow rate, and in a later phase of control. This later phase starts when the maneuvering valve has passed through the control range for the entire output of the first pump, and moves into an additional range affording control of a piloted motor which actuates the second valve.

The maneuvering valve is positioned by the action of an operator-set element and a follow-up control responsive to motor movement. Hence as the motor approaches the desired position, determined by the operator-set element, the piloted valve is returned to neutral and the final approach is controlled by the graduating action of the maneuvering valve which throttles only the flow from the small pump. This provides accurate control of motor movement to any desired position and at the same time affords the desired high speed characteristic.

A preferred embodiment of the invention will now be

described by reference to the accompanying drawings, in which:

Figs. 1 and 2 are respectively an elevation and a plan of the dual valve unit, these views being used to locate essential external connections and to define planes of section for other views.

Fig. 3 is a section taken on a plane defined by the axes of the maneuvering valve plunger and the piloted-valve plunger. The scale is larger than that of Figs. 1 and 2. The plane of section is indicated by the line 3—3 of Fig. 2. Both valves are shown in their centered or "lap position" in which position ports to the controlled motor are closed.

Fig. 4 is a section on the axis of the maneuvering valve plunger on a plane at 90° to the plane of section of Fig. 3 and at the same scale as Fig. 3. The line 4—4 on Fig. 1 will assist in identifying the plane of section.

Fig. 5 is a fragmentary section on the line 5—5 of Fig. 2 and at the scale of Fig. 3.

Fig. 6 is a fragmentary section on the line 6—6 of Fig. 2 and at the scale of Fig. 3.

Fig. 7 is a fragmentary section on the line 7—7 of Fig. 3 and at the scale of that figure.

Fig. 8 is a diagram of the manual controller, the control valve and the controlled motor with a simplified follow-up linkage. To make the figure clearly legible and keep within the limits of a single sheet, departures from true scale have been found necessary. The linkage per se is of a known type and the view is adequate for purpose of explanation.

Statements of direction, when made without qualification, refer to Fig. 3.

The principal features of novelty reside in a dual valve unit having two reciprocable plungers, one of which, called the primary distributing valve, is subject to manual control with follow-up effected by motion response of the controlled motor, and the other of which, called the secondary distributing valve, is shifted by a pressure motor piloted by the primary distributing valve when the latter is shifted beyond its initial range of motion in either direction.

Before describing the dual valve, a system to which it is suited will be described by reference to Fig. 8.

The maneuvering motor is there shown as a piston motor 11 having working spaces 12 and 13 separated by a piston 14 whose rod 15 is connected to a device to be maneuvered, for example a vehicle steering gear (not shown).

Control is effected by a wheel 16 which through a pinion and sector unit 17 swings lever 18 about a fixed fulcrum 19, between limits indicated on the drawing. A motion-combining link 21 is hinged to lever 18 and to suspender link 22 which is carried by the short end of rock lever 23. This lever 23 is fulcrumed at 24 and connected to rod 15 by link 25.

Link 21 is connected by link 26 to the end of plunger 27 in valve housing 28. Plunger 27 is the primary distributing valve, above mentioned.

The system has two sources of pressure fluid (ordinarily an hydraulic oil), a small pump 29 and a larger pump 31 which are shown as driven by a single continuously running motor 32. The pumps draw liquid from a sump conventionally diagrammed at 33. The pressure discharge of the small pump 29 is conducted by line 34 to a port in housing 28. A line 35 performs a similar function for the larger pump 31.

The connections to the working spaces of motor 11 from housing 28 are indicated at 36 and 37 in Figs. 1–3, inclusive, and 36 appears also in Fig. 5. The locations of these two connections are indicated in broken lines in Fig. 3. The main discharge connection from housing 28

is shown at **38** and leads from a manifold passage **39** (Fig. 4) to sump **33**.

The dual valve mechanism can now be described chiefly by reference to Figs. 3, 4 and 5.

Though not strictly necessary, it is convenient to use the illustrated arrangement in which the primary distributing valve plunger **27** and the larger secondary distributing valve plunger **41** are reciprocable in parallel seat bores. Each valve is of the open center type so that the discharge porting is similar. This porting differs little except in size and spacing as between the seats for the two valves, and the illustrated scheme of connecting together similar discharge ports of the two bores and then using one sump connection for all, affords economy in the use of external connections. The same scheme is used for the two motor ports, to the extent that they are connected between the two bores.

The plungers are of the type disclosed in the Stephens Patent 2,362,944, Nov. 14, 1944, that is to say they are bored and counterbored from each end to afford separated flow passages each with a valve seat and coacting spring seated check valve effective to prevent undesired back-flow through the motor ports in the plunger in the event of failure of the motive fluid supply.

An open center valve is one in which the main supply port communicates with the main discharge passage when the valve is in its neutral or mid-position (sometimes called "lap position" or "hold position"). For a double-acting motor these ports are commonly at mid-length of the valve seat and plunger, and such an arrangement is shown as to each of plungers **27** and **41**.

The main supply port **42** for the primary distributing valve **27** is an annular groove encircling the primary valve seat bore and is fed by connection **34**. It unloads the system into the bifurcated discharge port **35** whose bifurcations **35***a*, **35***b* straddle it. Conversely the main supply port **43** for the secondary distributing valve (which is fed by connection **35**) has bifurcations **43***a*, **43***b* which straddle port **35**. The bifurcations **35***a* and **35***b* communicate directly with manifold **39** (see Fig. 4). As a result the open center porting is simple and compact and can be formed by casting procedures.

There must be two exhaust passages for discharging working spaces of motor **11**. Encircling the seat bore for plunger **27** are annular groove ports **45** and **46**, and encircling the seat bore for plunger **41** are annular groove ports **47** and **48**. As clearly shown in Fig. 3, **45** and **47** are connected together by passage **49**, as are **46** and **48** by passage **51**. As shown in Fig. 4, **45** and **46** are extensions from manifold **39**.

Overload relief valves are arranged to open into extensions of passage **49**. These extensions are designated as **49***a* and **49***b*. The relief valves differ only in size and are of known form. The one which relieves supply port **42** into space **49***a* comprises an insert **51** clamped in a bored seat by set screw **52**. The insert has at its lower end a dashpot **53** and at its upper end a seat **54** for a poppet valve. Between these parts are side ports **55**. The valve unit proper comprises a valve head **56** and a piston **57** connected by an axially ported stem **58**, the piston **57** working in the dashpot **53**. A bonnet **59** threaded into housing **28** carries an adjusting screw **61** for the valve-loading spring **62**. A check nut and acorn nut clearly shown in section in Fig. 3 complete the assembly.

A relief valve assembly identical as to form but larger relieves port **43** into extension **49***b*. Its components are identified by primed numerals (**51'** to **62'**). In a prototype control unit each relief valve was set for appoximately 1350 p. s. i.

Leakage collecting annular grooves **60** encircle the plunger **27** near each end. The upper one is drained to space **49***a* (see Fig. 3) and the lower to port **46** (see Fig. 4).

Between port **35** and port **49** is the motor-cylinder port **63** (connected with working space **12**), and between port **35** and port **51** is the motor-cylinder port **64** (connected with working space **13**). Each of these ports terminates at each of its ends in a corresponding annular groove port formed in body **28**, the annular grooves **63***a* and **64***a* encircling the plunger **27** and the annular grooves **63***b* and **64***b* encircling the plunger **41**. Just as ports **47** and **48** are more widely spaced than are **45** and **46**, the annular grooves **63***b* and **64***b* are more widely spaced than are the annular grooves **63***a* and **64***a*. The purpose is to increase the functional ranges of motion of plunger **41** as compared with plunger **27**. Plunger **41** is actuated by a double-acting piston motor of rather small displacement. The functional travel of plunger **41** is large as compared to that of plunger **27** as its travel controls the output of the large pump **31**.

As has been stated, plunger **27** is bored and counterbored from each end. This affords a partition **65** between the bores and two seats **66** and **67** for spring loaded poppet valves **68**, **69**. Each poppet valve is crossbored, as shown. The valve loading springs **71** are confined between respective valves **68** or **69** and spring seats **72**. The upper seat **72** is supported by a gasketed member **73** which is threaded into the upper end of the plunger **27** and serves both as a sealed plug and as a shackle for the attachment of link **26** (see Fig. 1). The lower seat **72** is supported by a flanged member **74** which is threaded into the lower end of plunger **27** and serves both as a sealed plug and as a stop coacting with the slidable ring **75** to limit upward motion of the plunger. Ring **75** also coacts with shoulder **70** on plunger **27** and cap **80** to limit downward movement of the plunger. Each poppet valve **68**, **69** controls outflow through side ports **76** which lead to respective shallow grooves **77** surrounding the plunger. There are also side ports **78** through the plunger immediately above and below partition **65**. There are also shallow grooves **79** encircling plunger **27** in position to coact with port **45** and port **46**, respectively.

In mid-position of plunger **27**, as shown, port **42** unloads into port **35** and ports **63** and **64** are lapped. In an initial narrow range of positions above and below mid-position one of ports **64** or **63** is charged and the other vented to admit pressure liquid to one or the other working space **12**, **13** from pump **29**.

The plunger **41** is also an open center valve coacting with ports **47**, **48**, **63**, **64**, **43***a*, **43***b* and **35** to parallel on a larger flow-rate basis, the valve functions of plunger **27**. The plunger **41** is bored from its ends and counterbored to afford a middle partition **81** and seats for two poppet valves **82** and **83**. These have cross ports and seat respectively adjacent side ports **84**, **85** in the tubular portions of plunger **41**. The poppet valves are seated by springs **86**, **87** and are sustained by combined spring seats and sealed plugs **88**, **89** as shown.

The lower plug **89** sustains and guides a centering spring unit. The plug **89** affords a cylindrical guide **91** with flange **92** at its lower end. The end of plunger **41** provides a shoulder **93** at the upper end of the guide **91**. A helical spring **94** encircles guide **91** and reacts oppositely on two flanged, annular spring seats **95** which are slidable on guide **91**. This assembly is confined in a cylindrical recess **96** formed in body **28** and closed by gasketed cap **97**. Fig. 3 shows the construction and the proportions which assure centering of valve plunger **41**. A gasketed cap **98** encloses the upper end of plunger **41**.

From the construction just described, the plunger **41** functions as a movable abutment between two working spaces, **99** at the upper end and **101** at the lower end. Space **99** is connected to a groove port **102** encircling plunger **27**, and space **101** is connected to groove port **103** also encircling that plunger. In mid-position of plunger **27** these ports **102**, **103** are connected respectively to exhaust ports **45** and **46** by the grooves **79**, **79** which

encircle plunger 27. Thus when plunger 27 moves to its mid-position plunger 41 is freed to move to its centered position under the urge of spring 94.

In plunger 27 are side ports 104, 105 so that upon motion of that plunger, beyond the initial narrow range, pressure liquid continues to flow to the selected port 63 or 64 and also will flow from port 42 (the supply port from the smaller pump) through the cross bores in check valve 68 to port 102 to space 99 if plunger 27 is moved down, or through the cross bores in check valve 69 to port 103 if plunger 27 is moved up. This causes the plunger 41 to be shifted in the same direction as plunger 27, to connect the large capacity pump to motor 11.

It should be noted that during movement of the wheel 16 to shift plunger 27 in either direction, the follow-up mechanism acts in response to motor movement to restore the plunger 27 to its mid-position. Therefore, shifting of valve 41 occurs when the rate of wheel movement is greater than the rate of motor movement by an amount such that plunger 27, which is controlled by both the wheel and the follow-up linkage, shifts beyond the initial narrow range. Thus, plunger 41 remains in mid-position unless a predetermined turning rate is exceeded.

If plunger 27 be moved quickly beyond the initial range from mid-position, port 63 or port 64 (according to the direction of shift) would still be opened, so that port 42 would supply motor 11 with motive liquid. Consequently port 42 might not have sufficient flow of pressure liquid also to charge space 99 or 101 quickly if, for example, the motor 11 were lightly loaded. In such a maneuver, quick shift of plunger 41 is important because it supplies liquid from the large pump 31 to motor 11. To cover this contingency, the back pressure valve 106 shown in Fig. 7 is used. There is one such valve in each of ports 63 and 64 as indicated in Fig. 3. This valve 106 controls a seat 107 in port 63, the motor port leading to space 12 of motor 11. The valve is seated by a spring 108 and the space behind the valve is vented to exhaust port 49 at 109 (see Fig. 3). The annular shoulder 110 causes the valve 106 to open in response to reverse or exhaust flow through port 63. An identical valve assembly is used in port 64, the vent passage 109*a* leading to port 51, all as shown in Fig. 3.

The springs 108 of the two valves in the prototype valve were each adjusted so that its valve 106 produces a pressure drop of approximately 110 p. s. i. This proved satisfactory. The effect of the valves 106 is to ensure a back pressure at all times which will be sufficient to shift plunger 41.

It will be observed that whenever the valve plungers 27 and 41 are both in mid-position the piston 14 is hydraulically locked. To relieve shocks it is desirable to permit throttled cross flow above chosen pressures (such as are generated only by shock loading) between working spaces 12 and 13 of motor 11. The structure shown in Fig. 6 (not the invention of this applicant) is one way of accomplishing the result. Passages 63*a* and 64*a* are respectively branches of ports 63 and 64. Two spring loaded relief valves 111, 112 are reversely set so as to permit throttled flows in reverse directions between passages 63*a* and 64*a*. In the prototype they opened at approximately 1600 p. s. i., a value above ordinary operating pressure of motor 11.

It will be apparent that the described dual valve affords two rates of motor movement. The lower rate of motor movement is controlled by shifting of the plunger 27 within initial narrow ranges in either direction from mid-position. Within this range only pump 29 is loaded. A higher steering rate is afforded by shifting plunger 27 beyond the initial narrow range which causes plunger 41 to be shifted by hydraulic fluid admitted to motor space 99 or 101. Shifting of plunger 41 in either direction from mid-position causes loading of pump 31, pump 29 remaining loaded. This causes more rapid movement of the controlled motor. The follow-up mechanism acting on plunger 27 causes first plunger 27 and then plunger 41 to be returned to mid-position as the motor 11 reaches its desired position.

The inventive concept is not limited to use of the described preferred embodiment and no limitation to this structure should be implied except as is expressed in the appended claims.

What is claimed is:

1. The combination of a main double-acting fluid pressure motor whose action is to be controlled; two sources of pressure fluid, namely, a first source and a second source; a shiftable control member, a primary distributing valve having a mid-position and ranges of actuating positions which are reached by shifts of initial extent in opposite directions from mid-position and in which ranges the valve establishes admission-and-exhaust flows from said first source which respectively cause the motor to move in relatively opposite directions; an actuating and follow-up mechanism interconnecting said primary distributing valve, said control member and said main motor, whereby the position of the main motor is governed by the position of the control member; a secondary distributing valve having a mid-position and ranges of actuating positions respectively on opposite sides thereof, in which it establishes admission-and-exhaust flow paths from said second source, which respectively cause the motor to move in relatively opposite directions; means serving to bias the secondary distributing valve to its mid-position; a double-acting piloted pressure motor for overpowering said biasing means and shifting said secondary distributing valve into one or the other of its ranges, and means for controlling said piloted pressure motor and comprising exhaust and distributing ports controlled by the primary distributing valve, the distributing ports being rendered effective by relatively opposite shifts greater than the initial relatively opposite shifts of the primary distributing valve to energize the piloted pressure motor in opposite directions, whereby the admission path established by the second valve communicates with the working space of the main motor then receiving fluid through the primary distributing valve, and the exhaust port serving upon return of said primary distributing valve to a position within said range of actuating positions to de-energize the piloted motor.

2. The combination of a maneuvering motor of the double-acting expansible chamber type; means for supplying pressure fluid at two different flow rates; a maneuvering valve of the admission-and-exhaust type having a mid-position, said valve serving to control said motor by admitting and exhausting fluid from the low rate supply means to and from selected opposite working spaces of said motor in response to opposite initial range shifts from mid-position; valve controlling means having an operator-set component, a motor-moved component, and a follow-up connection therebetween, said connection serving to move said valve to said mid-position by response of the motor to change of set of the operator-set component; a second valve of the admission-and-exhaust type supplied with motive fluid by the high-rate pressure-supply means and shiftable in opposite directions from a mid-position in which it closes communication with said motor to positions in which it admits pressure fluid to one or the other working space of the motor, while exhausting the working space not so selected; a double-acting motor for shifting said second valve; and means for controlling the last-named motor, comprising ports in said maneuvering valve rendered effective only by range shifts greater than the initial range shifts from mid-position of the maneuvering valve, whereby the admission path established by shifting of the second valve communicates with the maneuvering motor working space receiving pressure fluid from said low-rate supply means.

3. The combination of a main double-acting fluid pressure motor whose action is to be controlled; two sources

of pressure fluid, namely, a first source and a second source; a shiftable control member; a primary distributing valve of the open center type having a mid-position in which it unloads the first source, and ranges of actuating positions which are reached by shifts of initial extent in opposite directions from mid-position, and in which ranges the valve loads the first source and establishes admission-and-exhaust flows which respectively cause the motor to move in relatively opposite directions; an actuating and follow-up mechanism interconnecting said primary distributing valve, said control member and said main motor, whereby the position of the main motor is governed by the position of the control member; a secondary distributing valve of the open center type, having a mid-position in which it unloads the second source, and ranges of actuating positions, respectively on opposite sides thereof, in which it loads the second source and establishes admission-and-exhaust flow paths which respectively cause the main motor to move in relatively opposite directions; means serving to bias the secondary distributing valve to its mid-position; a double-acting piloted pressure motor for overpowering said biasing means and shifting said secondary distributing valve into one or the other of its ranges aforesaid; and means for controlling said piloted pressure motor comprising exhaust and distributing ports controlled by the primary distributing valve, the distributing ports being rendered effective by relatively opposite shifts greater than the initial relatively opposite shifts of the primary distributing valve to energize the piloted pressure motor in opposite directions, whereby the admission path established by the second valve communicates with the working space of the main motor then receiving fluid through the primary distributing valve, and the exhaust port serving upon return of said primary distributing valve to a position within its range of actuating positions to deenergize the piloted motor.

4. The combination defined in claim 1 in which the first source is characterized by a lower flow rate than the second source.

5. The combination defined in claim 1 in which the two sources have different flow rates, and in which the motive pressure fluid for the piloted pressure motor is derived from the source having the lower rate.

6. The combination of a maneuvering motor of the double-acting expansible chamber type; means for supplying pressure fluid at two different flow rates; a maneuvering valve of the admission-and-exhaust type, having a mid-position in which it closes communication with said motor and unloads the low-rate supply means, said valve serving to control said motor by admitting and exhausting fluid from the low-rate supply means to and from selected opposite working spaces of said motor in response to opposite initial range shifts from mid-position; valve-controlling means having an operator-set component, a motor-moved component, and a follow-up connection therebetween, said connection serving to move said valve to said mid-position by response of the motor to a change of set of the operator-set component; a second valve of the admission-and-exhaust type, supplied with motive fluid by the high rate pressure-supply means, and shiftable in opposite directions from a mid-position in which it closes communication with said motor and unloads the high-rate pressure supply means, to positions in which it admits pressure fluid to one or the other working space of the motor, while exhausting the working space not so selected; a double-acting motor for shifting said second valve; and controlling means for the last-named motor, comprising ports in said maneuvering valve rendered effective only by range shifts greater than the initial range shifts from mid-position of the maneuvering valve, whereby the admission path established by shifting of the second valve communicates with the maneuvering motor working space receiving pressure fluid from said low rate supply means.

7. The combination of a maneuvering motor of the double-acting expansible chamber type; means for supplying pressure fluid at two different flow rates; a maneuvering valve of the admission-and-exhaust type having a mid-position, said valve serving to control said motor by admitting and exhausting fluid from the low rate supply means to and from selected opposite working spaces of said motor in response to opposite initial range shifts from mid-position; valve controlling means having an operator-set component, a motor-moved component, and a follow-up connection therebetween, said connection serving to move said valve to said mid-position by response of the motor to change of set of the operator-set component; a second valve supplied with motive fluid by the high rate pressure supply means and shiftable in opposite directions from a mid-position in which it closes communication with said motor to positions in which it admits pressure fluid to one or the other working space of the motor; a double-acting motor for shifting said second valve; and means for controlling the last-named motor, comprising ports in said maneuvering valve rendered effective only by range shifts greater than the initial range shifts from mid-position of the maneuvering valve, whereby the admission path established by shifting of the second valve communicates with the maneuvering motor working space receiving pressure fluid from said low rate supply means.

8. The combination of a maneuvering motor of the double-acting expansible chamber type; means for supplying pressure fluid at two different flow rates; a maneuvering valve of the admission-and-exhaust type, having a mid-position in which it closes communication with said motor and unloads the low-rate supply means, said valve serving to control said motor by admitting and exhausting fluid from the low-rate supply means to and from selected opposite working spaces of said motor in response to opposite initial range shifts from mid-position; valve-controlling means having an operator-set component, a motor-moved component, and a follow-up connection therebetween, said connection serving to move said valve to said mid-position by response of the motor to a change of set of the operator-set component; a second valve supplied with motive fluid by the high rate pressure-supply means, and shiftable in opposite directions from a mid-position in which it closes communication with said motor and unloads the high-rate pressure supply means, to positions in which it admits pressure fluid to one or the other working space of the motor; a double-acting motor for shifting said second valve; and controlling means for the last-named motor, comprising ports in said maneuvering valve rendered effective only by range shifts greater than the initial range shifts from mid-position of the maneuvering valve, whereby the admission path established by shifting of the second valve communicates with the maneuvering motor working space receiving pressure fluid from said low rate supply means.

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