June 14, 1966 H. H. PLATT 3,255,707

HYDRAULIC PUMPS AND MOTORS OF THE DISPLACEMENT TYPE

Filed Dec. 2, 1963 3 Sheets-Sheet 1

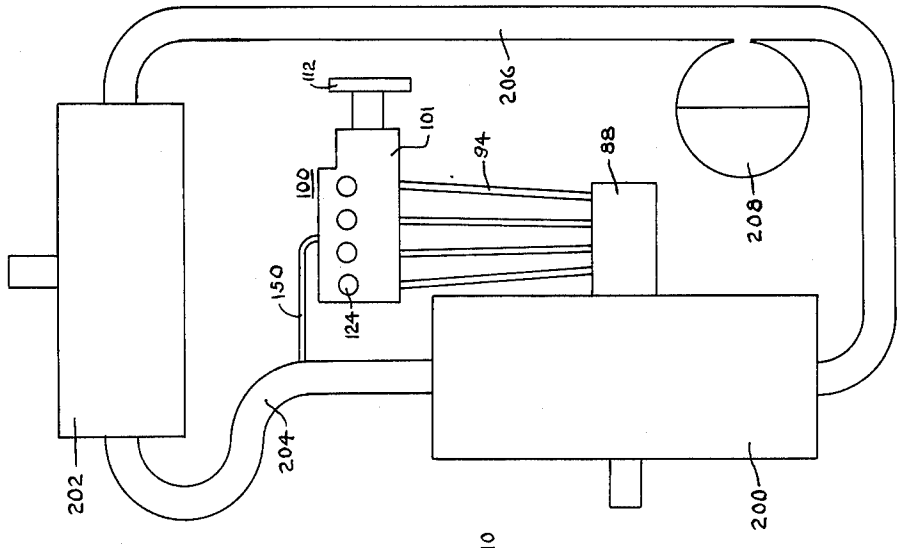


FIG. 6

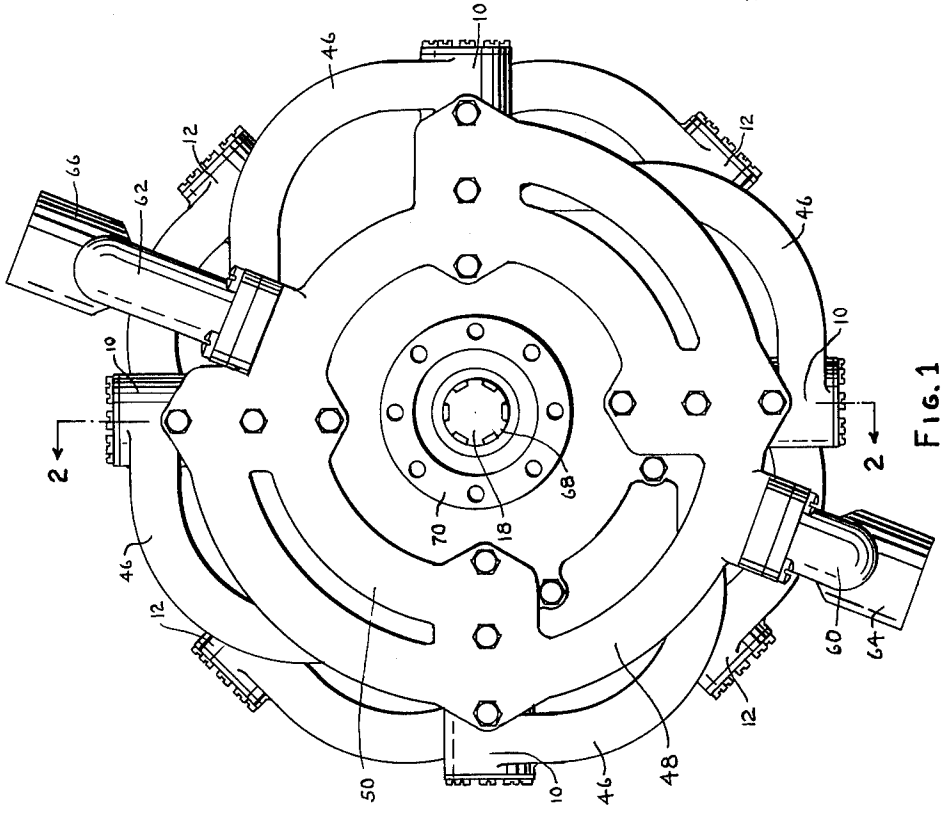


FIG. 1

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HYDRAULIC PUMPS AND MOTORS OF THE DISPLACEMENT TYPE

Filed Dec. 2, 1963 3 Sheets-Sheet 2

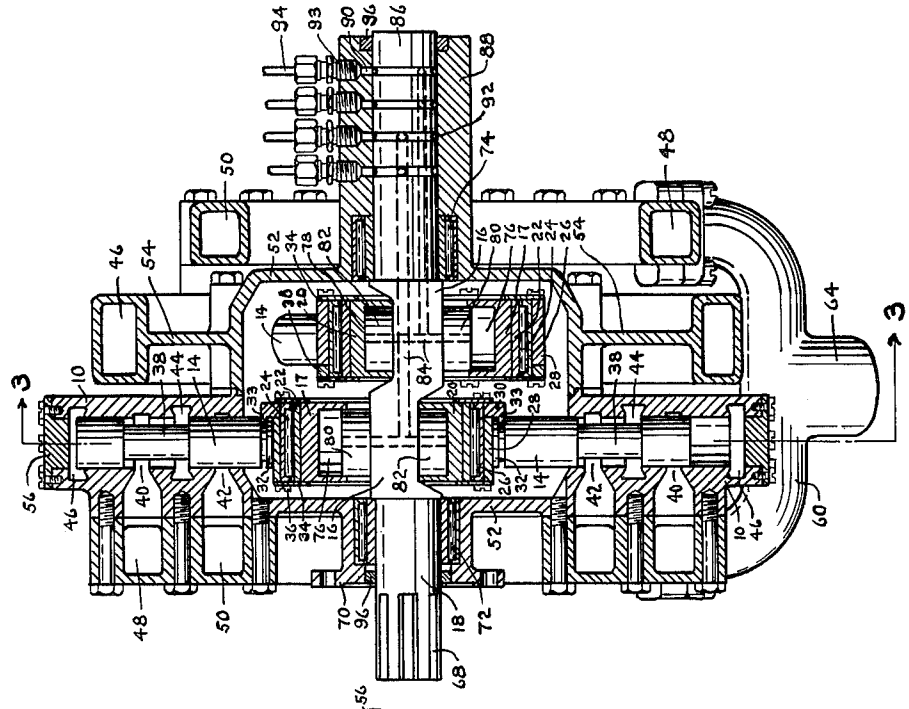


FIG. 2

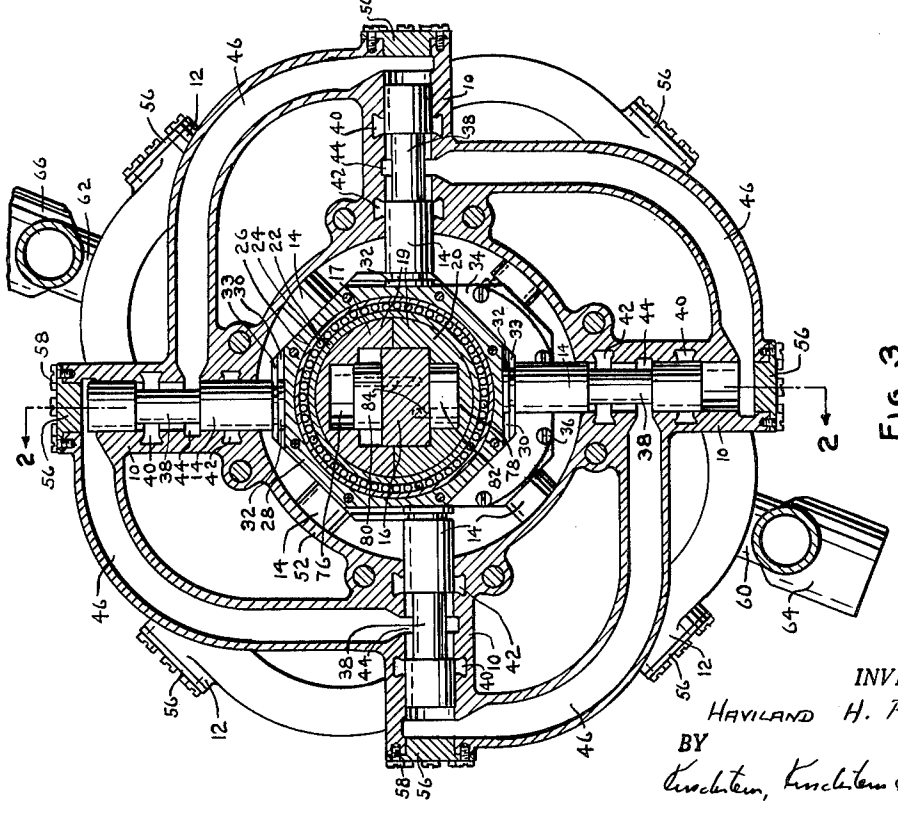


FIG. 3

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HYDRAULIC PUMPS AND MOTORS OF THE DISPLACEMENT TYPE

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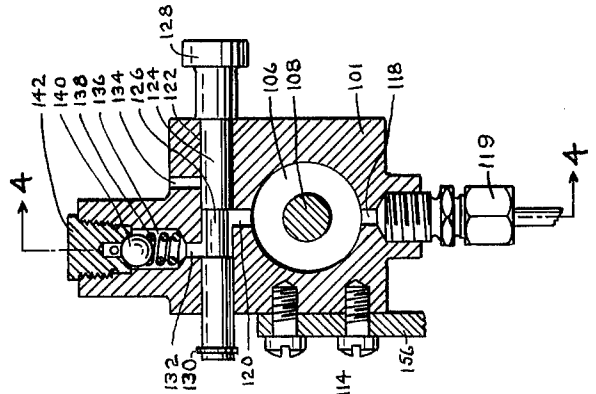


FIG. 5

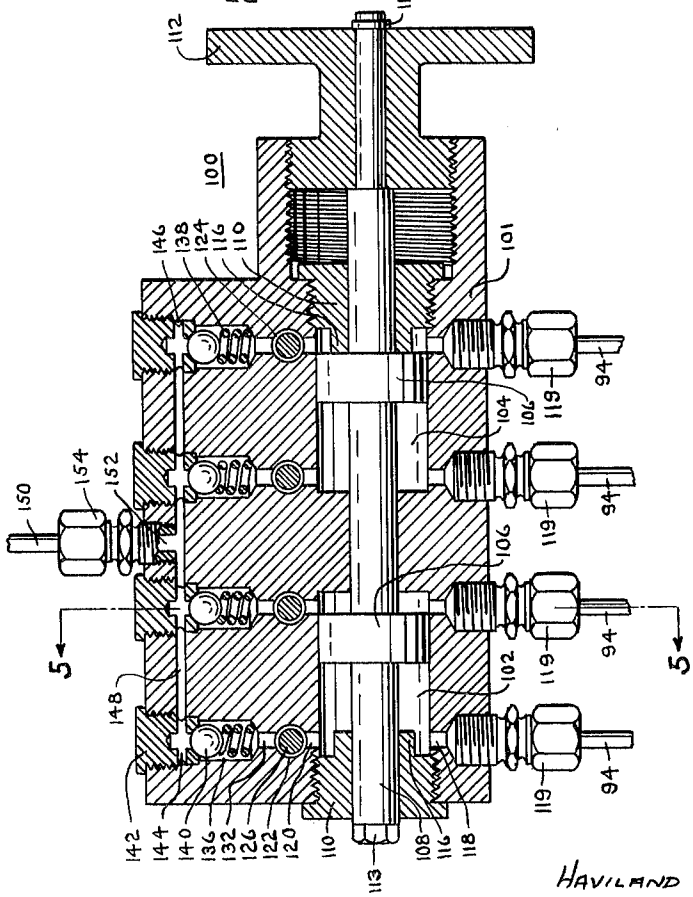


FIG. 4

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3,255,707

HYDRAULIC PUMPS AND MOTORS OF THE DISPLACEMENT TYPE

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13 Claims. (Cl. 103—174)

The present invention relates to hydraulic pumps and motors of the displacement type. More specifically it relates to a power unit in which radially disposed plungers serve either to pump hydraulic fluid at high pressure or to be acted on by hydraulic fluid so as to rotate a power shaft.

One object of my invention is to produce a power unit of the type described which is of less weight and bulk than those which have been used heretofore. This reduction in size and weight is of particular significance in the application to the driving of automobiles and other vehicles. With the use of pumps and motors of my novel construction it appears possible for the first time to achieve the great convenience and performance advantages of a central pump driving individual wheel motors.

Another object is to attain high efficiency in hydraulic pumps and motors through the reduction of mechanical friction and precision in valving.

Still another object is to provide a novel mechanism for changing and controlling the displacement of the plungers which is so compact that it adds very little weight and bulk whereby it has great advantages over mechanisms heretofore known for this purpose.

Further objects and advantages will become apparent from the descriptive material hereinafter.

My invention accordingly consists in the features of construction, combination of elements and arrangements of parts which will be exemplified in the hydraulic power unit hereinafter described and of which the scope of application will be indicated in the appended claims.

In the accompanying drawings in which are shown embodiments of my invention at present favored by me,

FIG. 1 is a front elevational view of one form of power unit embodying my invention;

FIG. 2 is a longitudinal sectional view taken substantially along the line 2—2 of FIG. 1 and illustrating the crank-shaft, eccentric and plunger arrangement of a variable displacement power unit which may serve either as pump or motor;

FIG. 3 is a transverse sectional view taken substantially along the line 3—3 of FIG. 2 illustrating particularly the relative arrangement of the plungers and cylinders;

FIG. 4 is a longitudinal sectional view of a displacement controller adapted for use with the power unit illustrated in FIGS. 2 and 3, taken substantially along the line 4—4 of FIG. 5;

FIG. 5 is a transverse sectional view of the controller taken generally along the line 5—5 of FIG. 4; and

FIG. 6 is a diagrammatic representation of a simple power transmission system employing units of my invention and illustrating particularly the hydraulic connections.

The underlying principle of my invention is the employment of plungers with the dual functions of pressure pistons and of balanced valves. For this purpose the cylinders in which the plungers fit slidably are disposed radially with their adjacent axes forming right angles with each other. The plungers are driven from or drive a crank-shaft depending on whether the unit is functioning as a pump or a motor, that is whether hydraulic fluid is received at low pressure and discharged at high pressure or received at high pressure and discharged at low pressure. In the first instance power is applied at the crank-shaft and in the second instance power is taken off through the crank-shaft.

The basic unit comprises four cylinders, indicated by the numeral 10, as shown in section in FIG. 3. With a single unit however the mechanical balance is inferior and the flow of hydraulic fluid uneven. These disadvantages can be eliminated or minimized by the addition of counterweights on the shaft and of an accumulator in the high pressure line. Such an arrangement is suitable for installations in which space and weight are unimportant. In the embodiment of my invention shown in the drawings, which is intended for more exacting service, two basic units are shown arranged side by side, the cylinders 12 of the second unit being spaced circumferentially intermediate of the cylinders 10 of the first unit, as shown particularly in FIGS. 1 and 3.

The plungers 14, fitting slidably in the cylinders 10 and 12, are operatively connected to the crank 16 of crank-shaft 18. The driving interconnection between crank 16 and plungers 14 is adapted for variable displacement through changing the strokes of the plungers 14. To this end there is mounted transversely slidable on crank 16 an eccentric 19 formed of two halves 17 and 20 which are held together firmly by the ring member 22. The ring 22 serves also as the inner race of a needle roller bearing with rollers 24 and outer race 26. Surrounding the outer race 26 is the sleeve 28 which is formed with four flat portions adapted to bear on the inner ends of the plungers 14 which are formed with mating flat transverse end surfaces. A flanged ring 30 is also provided to hold the plungers 14 in contact with the bearing surfaces of sleeve 28 by engagement of the flanges 32 on plungers 14 with the four tangentially disposed flanges 33 of ring 30. The assembly of ring 22, race 26, sleeve 28 and ring 30 are retained axially on the eccentric 19 by the ring plates 34 and the screws 36.

Each of the plungers 14 is formed with an intermediate portion of reduced diameter 38, the annular space surrounding portion 38 constituting a valve passage cooperating with the ports 40, 42 and 44. Passages 46 are provided to connect the port 44 of each cylinder 10 or 12 with the outer end of the cylinder next in the direction of crank-shaft rotation when the unit is serving as a pump, which is opposite to that for motor operation. Ports 40 are interconnected by a ring segment passage 48 and ports 42 are similarly interconnected by the ring segment passage 50.

Preferably, as shown, the four cylinders 10 and 12 of each unit are formed integrally with passages 46 and crankcase half 52. Stiffening webs 54 may be included also in the integral piece. Cylinder heads 56, secured by the ring of bolts 58, tightly seal the head ends of cylinders 10 and 12. The passages 48 and 50 are formed integrally and bolted to cylinders 10 and 12, as shown particularly in FIG. 1. The manifold 60, bolted to suitable flanges on ring segment 48 interconnects the passages 48 of the two units and manifold 62 similarly interconnects passages 50. Connections from the two manifolds 60 and 62 to external lines are indicated at 64 and 66. To insure accuracy of location, the edges of ports 40, 42 and 44 may be machined by a suitable tool inserted through the cylinder bore. Preferably, as shown, the center lines of the cylinders 10 and 12 are offset slightly in an angular direction opposite to that of pump rotation for the purpose of reducing the side force on the plungers 14.

The forward end of the crank-shaft 18 may be splined, as at 68, for the rotative connection of a motor or other mechanism for driving the unit as a pump or to be driven by the unit as a motor. The flange 70 may be furnished as a support. The crankshaft may be rotationally supported in the roller bearings 72 and 74. The crankcase

halves 52 are concentrically and tightly bolted together, as shown particularly in FIG. 2, to provide a complete enclosure for the working parts.

The provisions for varying the strokes of the plungers and consequently the displacement capacity of the units are described as follows:

The eccentric halves 17 and 20 are bored internally to form the two short cylinders 76 and 78 to fit slidably over the pistons 80 and 82 attached to or formed integrally with cranks 16. This arrangement allows the eccentrics, collectively designated 19, and the parts carried with them, to slide only along the axes of the pistons 80 and 82 through the distance between the face of piston 80 and the end of cylinder 76. When at the upper end of its travel, as shown in FIG. 3, the eccentricity of the composite eccentric 19 with relation to the axis of crankshaft 18 has a maximum value while when at the bottom of its travel the eccentricity is zero. Since the strokes of the plungers 14 are dependent solely on the eccentricity, it follows that by sliding the eccentrics on the pistons 80 and 82 the length of the plunger strokes may be varied from zero to a maximum.

In the operation of the arrangement shown particularly in FIGS. 2 and 3 the positions of the eccentrics 19 are established hydraulically by introducing fluid into the cylinders 76 and 78 and regulating the relative volumes of fluid in those cylinders. Thus, with the maximum amount of fluid above piston 80 and the minimum amount under piston 82 in FIG. 3 the eccentric 19 is in the position of maximum eccentricity and the plunger stroke, as well as the volumetric displacement of the hydraulic unit, is at its greatest value. However, if fluid is forced into cylinder 78 and evacuated from cylinder 76 the eccentric 19 together with everything attached to it will move downward, the eccentricity and displacement becoming zero when cylinder 78 is full and cylinder 76 empty. Thus, by controlled differential insertion of hydraulic fluid in 76 and 78 the plunger displacement of the pump or motor may be controllably varied between the maximum possible and zero.

Hydraulic fluid is led into cylinders 76 and 78 through internal drilled passages (shown by dashed lines) collectively designated 84 in crank-shaft 18 and pistons 80 and 82. There are four passages 84, each leading from one of the four annular grooves 92 in the shaft extension 86 to one of the four faces of pistons 80 and 82. In practice it may be necessary to continue portions of the passages 84 to the end of the shaft extension 86, in which case the outer ends of such passages are plugged tightly. For simplicity in illustration the unutilized portions are omitted from the drawings. Surrounding and fitting shaft extension 86 is an extension 88 of the rear crankcase half 52. In extension 88 are four radial passages 90, each registering with one of the annular grooves 92 and fitted with hydraulic tube connectors 93, of conventional design, establishing operative connections between tubes 94 and passages 90. Thus a free hydraulic connection is provided between each of the tubes 94 and one of the four cylinders 76 and 78. As shown, the two rearmost or outer tubes 94 serve the cylinders 76 and 78 of the rearmost eccentric assembly while the two forward or inner tubes serve the forward eccentric.

In FIG. 2 the shaft extension 86 is shown with its diameter the same as that of the bearing portions of shaft 18. Actually it is advantageous to make this extension of considerably smaller diameter in the interest of reducing friction and leakage, particularly if packings such as O-rings are installed between grooves 92. Since the flow of fluid in and out of cylinders 76 and 78 is of very small volume the passages 84 may be of smaller size than those shown, thus permitting the reduction in shaft extension diameter, the proportions shown in FIG. 2 being used in the drawing merely for the purpose of clarity of exposition.

A form of manual controller is illustrated to an enlarged scale in FIGS. 4 and 5. The controller 100 comprises as its main parts: a body 101 bored out at both ends to form two coaxial cylinders 102 and 104; pistons 106 fitting slidably in cylinders 102 and 104; a piston rod 108 operationally integral with pistons 106; end closure threaded plugs 110 forming heads for cylinders 102 and 104; and a manual control wheel 112 supported on rod 108 and threaded into the body 101.

The piston rod 108 may for assembly purposes be formed in two parts held together by the bolt 113. Control wheel 112 may be free to turn on piston rod 108 but is retained axially by the snap ring 114. Stops 116 limiting piston travel are formed on the cylinder head plugs 110. The bores of cylinders 102 and 104 together with the stroke of pistons 106 are such as to provide for each cylinder a displacement capacity somewhat greater than that of the corresponding eccentric cylinder 76 or 78.

Leading out from the four ends of cylinders 102 and 104 are passages 118 fitted with conventional hydraulic connectors 119 establishing free access to the four tubes 94, the other ends of which are shown in FIG. 2. As convenience dictates, the controller 100 may be remotely located from the pump or motor unit, as on an instrument panel, the connecting tubes 94 being bent to conform to the space requirements.

Extending also from the ends of cylinders 102 and 104 are the four passages 120, each leading to a transverse valve bore 122 in which is slidably fitted a valve plunger 124 having a portion of reduced diameter 126. The travel of valve 124 is restricted by knob 128 and snap ring 130. Leading upward from each bore 122 are two passages 132 and 134, 134 being open to the atmosphere and passage 132 leading to a cavity 136 containing a light spring 138 and a ball check valve 140. Each ball 140 is retained by a screwed-in plug 142 which is drilled axially and transversely, as shown at 144, and provided with an annular groove 146. A passage connection is thus established from each end of each cylinder 102 and 104 to passage 148 traversing the body 101, subject to the restraint of the check valves 140 which allow flow into the cylinders but not out of them when the valves 124 are in the position shown. Passage 148 is connected to a fluid supply tube 150 through passage 152 and conventional fitting 154. Thus, so long as hydraulic pressure exists in tube 150 and valves 124 are in the position shown, both ends of cylinders 102 and 104 as well as all four eccentric cylinders 76 and 78 are pressurized. If however any valve 124 is moved to the other end of its travel passage 132 is shut off and the corresponding end of one of the cylinders 102 and 104 is relieved of pressure through passage 134. Thus, by setting valves 124, any combination of pressurization and venting can be established for the four eccentric cylinders as desired.

A bracket for mounting the controller 100 is indicated at 156. This may be of any form and placed in any location suitable for establishing the controller in a suitable fixed position.

When the valves 124 are in their operative positions, as shown, turning the manual control wheel 112 moves piston rod 108 and pistons 106 axially in the cylinders 102 and 104. Fluid is thus forced from cylinders 102 and 104 into the appropriate eccentric cylinders. Thus, if the left-hand pair of tubes 94 in FIG. 4 serve the eccentric cylinders 76 and 78 of the front power cylinder block and the right-hand pair of tubes 94 those of the rear cylinder block and if the left-hand end of cylinder 102 connects with eccentric cylinder 78, moving the pistons 106 toward the left displaces the left-hand eccentric 19 downward and the right-hand eccentric upward in FIG. 2. The result is a decrease in eccentricity and a diminution of the plunger displacement in both cases. When by this process cylinders 78 have been filled the plunger displacement will have been reduced to zero. Obviously, depending on the connection arrangements of tubes 94 and the various connecting passages, the controller can be ar-

ranged for either direction of displacement response for either direction of motion of piston rod **108**. In any case however the connections are so coordinated that the maximum displacements and the minimum displacements coincide in both cylinder blocks. Mechanical balance is thus assured.

The pressurization of the cylinders **102**, **104**, **76** and **78** assures that they be entirely filled with fluid at all times during operation. The check valves **140** enable this pressurization from the common pressure supply in tube **150** without possibility of back circulation which would impare the positional relationship between pistons **106** and eccentrics **19**.

The functions of valves **124** are two-fold. One of these is that of voiding any air that may be trapped in cylinders **76**, **78**, **102** and **104**, particularly when filling the system after assembly. By actuating first the two valves **124** for the right-hand ends of cylinders **102** and **104** and then the two for the left-hand ends, the eccentrics are caused to reciprocate. The air in both cylinders **76** and **78** can thus be completely voided through passages **134** and replaced with hydraulic fluid. Air in cylinders **102** and **104** is similarly forced out.

The second function of valves **124** is that of enabling the establishment of the correct positional relationship between the pistons **106** and the two eccentrics **19**. Since there is no mechanical connection between these parts some provision must be made to assure the required coordination both initially and during operation. Initial coordination is established preferably by placing the controller in zero displacement position, that is with piston **106** against the left-hand stop **116**, and then venting cylinders **76**. The eccentrics **19** are thus forced by pressure in cylinders **78** into their zero eccentricity positions. The desired coordination having once been established it can be impaired in operation only by differential leakage in the system. Such a differential effect would normally take a long time in developing and can be effectively corrected quickly by repeating the initial procedure described above.

The operation of the hydraulic power unit of my invention is clearly shown by a study of the positions of the plungers shown in FIG. 3. With the disposition shown the shaft is driven in the clock-wise direction for pumping. The eccentric is then causing the plunger **14** in the three o'clock azimuth to move outward in the center of its outward stroke. Fluid is thus being forced out of its cylinder **10** and through passage **46**. At this time the plunger **14** in the twelve o'clock azimuth is at the outermost end of its stroke and consequently its valve portion is in position to connect port **44** with port **40**. Since port **40** connects through duct **48** with the high-pressure manifold **60** the fluid from the three o'clock cylinder discharges readily into the high-pressure line. When the crank-shaft has rotated through a further 90° the plunger of the three o'clock cylinder will have attained the position of that shown at twelve o'clock, that is it will be at the outer end of its stroke and the discharge from the cylinder will have stopped. The valve position of the twelve o'clock plunger will then be that shown in the drawing at nine o'clock with all port connections momentarily shut off. Continuing the rotation through another 90°, the plunger under consideration will have assumed the inwardly moving half-way position depicted at nine o'clock with the volume in the cylinder increasing for the entrance of a fresh charge of fluid. The valve controlling this flow is then in the position shown at six o'clock, that is ports **44** and **42** are connected allowing free flow from low-pressure manifold **62** through passage **50**. Finally, at 270° of rotation the plunger initially at three o'clock is in the position shown at six o'clock which is at the inner end of the stroke. The flow into the cylinder has ceased and discharge is about to begin. The controlling valve will then be in the position shown at three o'clock with all connections closed and the discharge passage about to open. Thus there is attained the ideal valve action in which all port openings and closings occur at times of zero flow. Moreover it will be noted that the pressures on all valve parts are completely balanced so that no thrusts or consequent friction forces occur at any time during the valve motion. Recapitulating from the point of view of a static picture: the three o'clock cylinder is discharging through the twelve o'clock valve; the twelve o'clock cylinder has just completed discharge and the nine o'clock valve is closed; the nine o'clock cylinder is receiving fluid through the six o'clock valve; and the six o'clock cylinder has just completed filling, the three o'clock valve being closed and in transition.

It is noteworthy that, since the valve shift always occurs at the center of the plunger stroke, changing the plunger stroke does not affect valve timing. Therefore the valve action is equally correct for all stroke control positions. It is also true that, for the type of mechanism described, the cylinders of a group must be spaced 90° apart for correct valve timing. Consequently there can be only four cylinders to a group although any multiple of four may be coplanar and operated by the same crank if their connecting passages segregate them into sets of four cylinders.

Driving in the opposite direction, counterclockwise, produces an identical result except for the difference that the manifold pressures are reversed, manifold **60** becoming the low-pressure line and manifold **62** the high-pressure line.

For use as a motor no change is necessary but, as will be observed, the shaft rotation will be opposite to that for pump operation.

In normal operation sufficient pressure will be maintained on the low-pressure line external provision in the normal manner to assure filling of the cylinders without cavitation. Thus it follows that in operation the forces on the plungers **14** are always inward. The retaining flanges **30** therefore are not involved in normal performance but are necessary to prevent plungers from becoming misplaced if the shaft **18** is rotated while the system is unpressurized.

For constant displacement units the variable eccentricity feature is not needed. The eccentrics are then mounted rigidly on the shaft. This allows a slight reduction in diameter reflected in some lessening in size and weight of the unit as a whole. Shaft extension **86** and case extension **88** are also omitted, as well as controller **100**. These changes are however of little importance in weight and are a measure of the remarkably small difference in bulk and weight between the constant and variable types. This is in great contrast to the hitherto usually employed swashplate type of piston actuation which leads to a large size and weight penalty for displacement variability.

The two cranks **16** in the form of my invention illustrated are directly opposed thus assuring a satisfactory balance for high speed operation. The alternate spacing of cylinders **10** and **12** provides a smooth flow of discharged and entering fluid.

An important feature of the arrangement illustrated particularly in FIGS. 2 and 3 is the ease of assembly. It will be noted that no difficulty presents itself in assembling the front unit, that is the block having the cylinders **10**, provided the cylinder heads **56** are secured last. However, if suitable provision were not made, the placing of the rear block, with cylinders **12**, would be impossible. The sequence of operations which makes the assembly easy is as follows: after assembling the forward parts assemble all the eccentric parts of the rear unit on the crankshaft; install bearing **74** in the rear case; push the rear case over the shaft until it is a short distance from mating with the front case; insert the plungers **14** through the outer ends of cylinders **12**, forcing their flat inner faces against the flat faces of sleeve **28**; move the rear case into full engagement and bolt in position, flanges **28** on

plungers **14** sliding into engagement under sleeve flanges **33**; place cylinder heads **56**; and bolt on the part containing passages **48** and **50** and manifolds **60** and **62**.

The power unit described and illustrated herein, while useful for all hydraulic motor and pump applications, is particularly intended for high pressure and high power utilization. A number of its novel features are specifically devised with this end in wiew. Important among these are the flat inner ends of the plungers **14**. If these were conical or spherical and bore directly on a circular eccentric contour the timing of the valves would be seriously impaired. Moreover, the unit bearing pressure at the eccentric contact would be prohibitively high for heavy duty service. Also if there were no antifriction bearing interposed between the eccentric and the plungers the rubbing friction would cause a serious reduction of efficiency. Even with a flat plunger end the unit pressure would be high if a circular outer eccentric member were used. These difficulties are effectively overcome by the construction shown in which the plunger thrust is carried on a joint between two flat surfaces, the unit bearing pressures being thus reduced to an entirely acceptable value. It may be noted that the displacement travel of the faces of plungers **14** relative to the eccentric sleeve **33** per revolution is quite short, the friction loss being therefore small.

A very important feature of the construction shown is that all plungers at all times, regardless of the eccentricity, have true harmonic reciprocation. Any departure from this ideal would result in disturbance of the valve timing. The serious consequences of such an error will be apparent on consideration of a condition in which a valve closes off a cylinder before the precise end of the pressure stroke. The residual fluid in the cylinder would immediately attain a pressure of destructive intensity. Obviously other timing errors would cause loss of efficiency and shock noise. It is on account of the precise valve timing that units constructed in accordance with my invention are very quiet in operation.

For the sake of simplicity no sealing provisions are shown except for the two conventional fluid seals **96** at the shaft ends. It is to be understood that all joints, where necessary, are to be sealed against leakage by the usual provisions such as O-rings, chevron packing, etc., as is well known in the art.

In this connection it is to be noted that moderate leakage is of minor importance in operation, provided the usual means of returning the leaked fluid is utilized, except insofar as leakage from the eccentric control system may disturb the positional relationship between the controller and the eccentrics. An important feature in minimizing this possibility is that of providing greater volume in the controller cylinders **102** and **104** than in the eccentric cylinders **76** and **78**. Thus, cumulative leakage up to the volume difference does not impair performance unless it is differential between the two eccentric units, as noted above.

Also for simplicity there has been omitted from FIG. 6 the usual reservoir for replenishment of fluid and means for injecting it. It is to be understood that these omissions, as well as possible omissions of other features known to the art, in no way impair the operational integrity and practicability of the devices described herein.

FIG. 6 illustrates diagrammatically one type of application of the hydraulic power units of my novel construction. A variable displacement unit operating as a pump **200** is connected in a closed circuit with a constant displacement motor **202** by the high pressure line **204** and the low pressure return line **206**. A conventional accumulator **208** is connected to the low pressure line **206** to assure positive pressure at all times during operation. The displacement controller **100** is shown connected to the extension **88** of the pump **200** by the tubes **94** and to the high pressure line **204** by the tube **150**. This arrangement constitutes a power transmission system having a controllable reduction ratio with the characteristics of stepless infinite variability. The absence of gears and shafts, or other mechanical connections, makes this system especially desirable for remote or difficultly accessible motor locations. Another advantage is that a number of motors can be served by one pump either in series or parallel, this potentiality being particularly applicable to multiple machine drives and automobile wheel propulsion.

The functions may be reversed, if circumstances require, the constant displacement unit being driven as a pump and the variable displacement unit serving as a motor.

While for simplicity manual control of displacement is illustrated variations to provide a number of types of automatic actuation are contemplated. Thus, as an example, in a common type of hydraulic power system a number of motors are served from a single high pressure reservoir which is to be maintained at constant pressure. For this purpose a pressure sensor provides a ready control actuator to reduce discharge of the pump when the pressure rises and vice versa. In another example a combination of speed, throttle and pressure sensors may be used in the application of variable displacement for automatic vehicle drives and the like.

While only one form of the embodiment of the invention has been illustrated and described it will be readily understood that the principles and essential spirit of the invention may take a wide range of other forms in practice and therefore the specific disclosure is illustrative rather than restrictive and the scope of the invention is defined only by the appended claims.

Having thus described the invention, there is claimed as new and desired to be secured by Letters Patent:

1. In a hydraulic power unit, a shaft, an eccentric supported on said shaft, four cylinders radially disposed about said shaft and spaced 90° apart, plungers in said cylinders having reduced diameter valve portions intermediate their ends, three valve ports in each of said cylinders adapted to cooperate with said valve portions of the plungers, a fluid passage connecting the central one of said ports of each cylinder to the head of the cylinder adjacent, an inlet passage leading to another of said ports, a discharge passage leading from the other of said ports, said eccentric being so disposed as to impart by its rotation reciprocating motion to said plungers and said valve portions of the plungers being thereby caused to connect said central port of each cylinder alternately to each adjacent port for alternately admitting fluid to another cylinder during the inward motion of its plunger and allowing discharge during the outward motion of its plunger, said eccentric being constrained to rotate with said shaft but free to slide transversely thereto, opposed pistons operatively integral with said shaft, internal cylinders in said eccentric cooperating with said piston to form opposed chambers of variable but complementary volumes, fluid passages leading from said chambers whereby differential pressures may be applied to cause displacement of said eccentric, and a controller connected to said fluid passages and adapted for varying the differential pressures in said chambers, said controller comprising a cylinder, a piston slidable in said cylinder, two chambers formed by said piston in said cylinder and means adapted to move said piston whereby the relative volumes of said chambers are altered, said passage means connecting each of said cylinder chambers with one of said eccentric chambers, whereby motion of said piston imparts corresponding motion to said eccentric.

2. In a hydraulic power unit as set forth in claim **1**, said controller, said cylinder chambers and said eccentric chambers are connected with a source of pressure, and check valves are interposed between said cylinder chambers and said source of pressure, said check valves being disposed to prevent flow from said chambers to said pressure source.

3. In a hydraulic power unit as set forth in claim 2, valve means interposed between each of said check valves and each controller chamber, said valve means being adapted to close off the source of pressure and to vent said chamber.

4. In a hydraulic power unit, a shaft, an eccentric on said shaft, a cylinder, a plunger in said cylinder adapted for reciprocation by said eccentric, a sleeve freely rotatable on said eccentric, the inner face of said plunger being adapted to bear on said sleeve, the inner face of said plunger being flat and said sleeve having a flat portion adapted to maintain flat-to-flat contact with said plunger, a flange on said plunger, a hold-down flange operatively integral with said sleeve adapted to engage slidably the flange on said plunger, hydraulic means adapted to vary the eccentricity of said eccentric by causing said eccentric to slide transversely of said shaft, said hydraulic means including pistons mounted transversely of said shaft and cylinders in said eccentric fitting slidably over said pistons, opposed chambers in said cylinders closed by said pistons, and control means adapted to introduce fluid under pressure into said chambers differentially thereby causing said eccentric to be moved transversely of said shaft, said control means including a control cylinder, a piston in said cylinder, said piston forming chambers of complementary variable volume, passage means connecting said control cylinder chambers with the cylinders in said eccentric.

5. In a hydraulic power unit as set forth in claim 4, a source of fluid pressure, check valves interposed between said source of pressure and said control cylinder, and said check valves being arranged to allow flow from said source of pressure and to prevent flow toward it.

6. In a hydraulic power unit as set forth in claim 5, valve means interposed between said check valves and said control cylinder adapted to cut off access from said source of pressure to said cylinder and to vent said cylinder.

7. In a hydraulic power unit, a cylinder, a plunger in said cylinder, a shaft, an eccentric supported on said shaft and adapted to reciprocate said plunger in said cylinder, said eccentric being constructed to rotate with said shaft and having freedom to slide transversely thereto, opposed pistons operatively integral with said shaft on opposite sides thereof, said eccentric being split into two portions, each portion having an internally bored cylinder fitting one of said pistons and forming therewith within said eccentric two opposed chambers of variable but complementary volumes, fluid passages leading from said chambers whereby differential pressures may be applied in said chambers to cause displacement of said eccentric, and a controller connected to said passages and adapted to exert differential pressures in said chambers.

8. In a hydraulic power unit as set forth in claim 7, said controller comprising a cylinder, a piston slidable in said cylinder, two chambers formed by said piston

in said cylinder, means adapted to move said piston whereby the relative volumes of said chambers are altered, said passage means connecting each of said cylinder chambers with one of the chambers in said eccentric, whereby motion of said piston imparts corresponding motion to said eccentric.

9. In a hydraulic power unit as set forth in claim 8, said controller chambers have fluid connections with a source of pressure, said connections having check valves arranged to prevent flow from said chambers to said pressure source.

10. In a hydraulic power unit as set forth in claim 9, valve means interposed between each of said check valves and each controller chamber, said valve means being adapted to close off the source of pressure and to vent said chamber.

11. In a hydraulic power unit, a shaft, an eccentric on said shaft slidable transversely thereto, a cylinder, a plunger adapted for reciprocation in said cylinder in response to rotation of said shaft and eccentric, and hydraulic means adapted to move said eccentric in relation to said shaft, said hydraulic means comprising cylinders in said eccentric, pistons in said cylinders, a controller external to said eccentric, a cylinder in said controller, a piston in said cylinder, and passage means connecting said control cylinder with the cylinders in said eccentric.

12. In a hydraulic power unit as set forth in claim 11, a source of pressure, passage means connecting said source of pressure with said control cylinder, and check valves in said passage means arranged to permit flow of fluid from said source of pressure and to prevent flow toward it.

13. In a hydraulic power unit as set forth in claim 12, valve means interposed between said check valves and said control cylinder adapted to close off the source of pressure from the control cylinder and to vent said control cylinder.

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